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THE
EFFICIENT USE OF FUEL

A TEXT-BOOK ON FUELS AND THEIR EFFICIENT UTILISATION FOR THE USE OF STUDENTS AND TECHNICAL MEN IN INDUSTRY

Prepared under the Direction of the Education Sub-Committee of the Fuel Efficiency Committee of the Ministry of Fuel and Power

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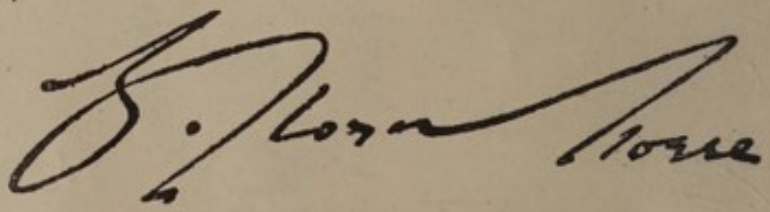
FOREWORD

FROM THE MINISTER OF FUEL AND POWER

THE effective use of fuels of every kind is of vital importance now, and will continue to be essential to the country's well-being so long as we are dependent on coal for our heat and power. The promotion of the efficient use of fuel has been a major preoccupation of the Ministry of Fuel and Power since its formation, and I trust that this book, in which has been assembled a very great deal of information not readily available, will not only serve the present urgent need, but will be of permanent value to all those concerned with the industrial use of fuel.



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PREFACE

MANY and hard are the lessons taught by the stress and strain of war. Not the least amongst those that emerge from the present conflict has been the realisation by the British people of the value of their coal resources. There is a danger that the supply of fuel will be inadequate to maintain industrial activity to its fullest extent, and, as a consequence, the vital supply of munitions will be interrupted. To meet this situation the Fuel Efficiency Campaign has been organised by the Ministry of Fuel and Power ; and amongst the instruments of that movement were the emergency training schemes undertaken jointly by the Board of Education and the Ministry to spread the knowledge of the efficient methods of using fuel.

It was soon realised that no single existing text-book covered both the fundamental principles and the essential operative features necessary to attain immediate results. The help of leading fuel technologists was therefore enlisted by the Education Sub-Committee of the Fuel Efficiency Committee, and a syllabus for the guidance of lecturers and a series of lecture notes that could be given to students were drawn up. In the course of preparing this material, however, it was soon realised that only the barest information could be given and the urgent need for a more comprehensive text-book on Fuel Efficiency was disclosed. Moreover, the favourable reception accorded to the Sub-Committee's earlier effort was a further incentive to the preparation of an extended and illustrated text-book, embracing the use of all fuels for industrial purposes.

The present work is the outcome of this effort. Its preparation has been entrusted by the Sub-Committee to the General Editorial supervision of Dr. G. E. Foxwell. Many leading authorities have generously assisted in its compilation and Dr. Foxwell has himself contributed substantially to the preparation of the subject matter. Each individual chapter has been submitted for approval to panels of experts on the particular subjects under review, and they have in many cases subjected them to certain criticism and revision. It is thus apparent that every means has been taken to ensure that the book is as authoritative as possible, free from personal bias and appreciative of the practical outlook.

It is not intended as a book of reference to be placed on the shelves of libraries and to which occasional reference is made. Each chapter is meant to be read by the student, and though it may be that one individual will not read them all, he will select those in which he is interested, and possibly become interested in some of the others.

The book was originally intended to emphasise the " application of knowledge," and arising out of this attempt to be practical, the preparation of certain chapters has indicated where information is lacking, and the Committee hopes to explore these untrodden paths in the near future.

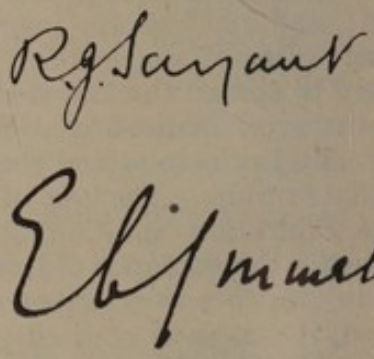
In spite of the efforts made to present an effective work, the book has been prepared under emergency conditions and the need for haste has been uppermost. It has been done by busy men and many imperfections may accordingly be apparent. For these indulgence is asked, and it is requested that they may be pointed out so that they may be removed or corrected in future editions.

Further, it is hoped that this work may be regarded as part of a live and progressive movement, directed not only to the betterment of the present

supply situation, but to an improvement in the efficient use of our greatest mineral asset. It may not be possible to say how long our coal resources will last, but it is possible to ensure that the most may be made of their potential value.

The Education Sub-Committee cannot refer too highly to the skill and enthusiasm shown by Dr. Foxwell in the editing of this work. Their cordial thanks are offered to Mrs. Fuller for her share in piloting the work through its preliminary drafts, revision and proof reading.

Grateful acknowledgement is accorded to those mentioned below, for having taken part in the production of this work, and for their generous and able assistance, always given so freely and promptly. Thanks are extended, in no less a degree, to all those who have assisted but remain anonymous.



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CONTENTS

CHAPTER PAGE

FOREWORD ii

PREFACE iii

I. INDUSTRIAL FUELS: COAL 1

Fuel conservation—Origin and nature of coal—Properties of coals—Grouping of coals—Main uses—Industrial characteristics—Sulphur—Coal preparation and cleaning—The size of coals.

II. INDUSTRIAL FUELS: FUELS DERIVED FROM COAL 23

Carbonisation—Coke—Coal gas (Town gas)—Producer gas—Blast furnace gas—Water gas—Coal tar fuel oils—Electricity.

III. THE STORAGE OF COAL 39

Oxidation of coal—Chemistry and physics of oxidation of coal—Practical principles of storage—Deterioration of coal in store (coking power, C.V., size, friability)—Watching temperatures—Dealing with fires—Storage of open-cast coal.

IV. THE CHEMICAL PRINCIPLES OF COMBUSTION AND GASIFICATION 49

Combustion and gasification—Chemistry of combustion—Calculation of analysis of flue gas from solid and gaseous fuels and of air requirements—Chemistry of gasification.

V. HEATING EFFECTS IN COMBUSTION AND GASIFICATION 60

The measurement of heat—Specific heat—Excess air for combustion and its effects—Heat losses in flue gases—Thermal effects in gasification—Ignition temperatures—Ignition limits—Flame speed—Flame temperatures.

VI. COMBUSTION IN PRACTICE 77

The art of securing proper combustion—Draught—The calculation of natural draught—Dimensions of the chimney—Primary and secondary air—Flow of air through fuel beds—Excess air—Flue gas losses—Time, temperature and turbulence as factors in combustion technique—Physical characteristics of the fuel bed—Combustion of various fuels and industrial combustion processes—Clinker formation—Firebed cooling—The utilisation of draught.

VII. PROPERTIES AND PRINCIPLES OF UTILISATION OF STEAM .. 104

Pressure—Formation of steam—Work done by steam—Selection of boiler pressures—Superheated steam for power production and for process work—Use of steam in process work—Pipe-lines—Return of condensate.

VIII. HEAT TRANSMISSION 119

Methods of heat transfer—Transmission of heat through a boiler plate—Conduction—Overall conduction coefficients—Conduction through composite walls and pipe coverings—Convection—Natural convection—Forced convection—Convection effects in banks of tubes and in single tubes—Heat transfer from superheated steam—Convection in liquids—Heat transfer from condensing vapours—Boiler circulation—Radiation—The laws of radiation—Radiation from a coke fire—Gas radiation, including methods of calculation—Radiation from luminous flames—Heat exchangers, regenerators and recuperators—Heat losses from external surfaces—Heat loss through a furnace wall—Practical aspects of heat transmission in boilers.

CHAPTER PAGE

IX. THE FLOW OF FLUIDS 166

The mechanics of fluid flow—Bernouilli's theorem—Streamline and turbulent flow—Viscosity of fluids—Equations of fluid flow—Enlargement and contraction losses—Effect of bends, etc.—Special formulæ used for gas transmission—Principles of the measurement of fluid flow—The flow of gases in furnaces—Examples of fluid flow calculation.

X. THE MEASUREMENT OF SOLID FUEL, STEAM, WATER, GAS AND AIR 202

Solid fuel; methods involving measurement and direct weighing—Measurement of fluid flow by orifice plate—Design data—Orifice location—Pressure losses—Formulæ—Pulsating flow—Correction factors—Measurement of water by V-notch—Examples of calculation.

XI. STOKING AND BOILER OPERATION 229

The air supply—The management of draught—Passage of air through a fuel bed—Methods and practice of hand firing—Methods and practice of mechanical firing—Water level—Blow-down—Cleaning fires—Banking fires—Cleaning heating surfaces—Carbon losses—Clinker formation.

XII. PULVERISED FUEL 251

Equipment and application : Pulverisation—Extraction fans—Distribution systems—Burners—Application to boilers—Cement—Metallurgical applications in the ferrous and non-ferrous industries—Choice of fuel—Furnace design.
Efficient use of pulverised fuel in boilers : Differences between operation on pulverised fuel and firing on grates—Bogey results—How to survey a pulverised fuel boiler installation.

XIII. STEAM BOILERS 273

Boiler design—Vertical boilers—Stationary locomotive boilers—Cornish boilers—Lancashire boilers—Externally fired boilers—Economic boilers—Waste heat boilers—Water-tube boilers—Effect of prime mover on boiler design—Feed water temperature—Furnace design—Water circulation.

XIV. STEAM BOILER AUXILIARY PLANT 297

Boiler mountings and fittings—Soot removal—Superheaters—Economisers—Feed water heaters—Air preheaters—Auxiliary power and steam utilisation—Boiler feed pumps—Pumping hot water—Maintenance of feed pumps—Draught production by fans.

XV. BOILER FEED WATER PURIFICATION 325

Mineral salts in water—Effects of scale—Water treatment—The lime soda process—The zeolite process—Demineralisation—Internal treatment and water conditioning—Feed water for high pressure boilers—Corrosion—De-aeration—Caustic embrittlement—Blow-down, its control, technique and disposal—Control of water treatment—Sampling and testing.

XVI. BOILER EFFICIENCY AND TESTING 340

The methods of testing steam boilers to secure essential information regarding operating results—Examples of operating results for efficient and inefficient plants—The more detailed examination of boiler plants.

XVII. GAS PRODUCERS 355

Gas producer plant—Nature and characteristics of fuel—Blast and the admission of steam—Clinker—Channelling—Producer efficiency and heat balances—Operation of producer—Purification of producer gas—Utilisation of producer gas.

CHAPTER PAGE

XVIII. FURNACES 376

The selection of furnaces—Furnace atmospheres—Combustion and heat transmission—Sources of heat loss—Thermal efficiency—Recovery of waste heat—Insulation—The heat losses associated with continuous and intermittent operation—Radiation losses through openings—Other losses—Scientific control—Instruments—Records—Practical hints to operators of furnaces fired by coal, coke and gas, pulverised coal and oil.

XIX. SPECIAL FURNACES 407

Structural features—Auxiliary plant—Furnace testing—Melting furnaces ; hearth, crucible, shaft and cupola furnaces—Sintering furnaces —Kilns and ovens ; intermittent and continuous—Furnaces for the glass industry—Reheating furnaces ; soaking pits—Heat-treatment furnaces : single-purpose furnaces ; conveyor furnaces ; " top-hat " furnaces ; lead pots and salt baths—Furnace control—Variable heat flow in furnaces.

XX. WASTE HEAT BOILERS 486

The availability of waste heat—Principles of heat recovery in waste heat boilers—Draught—Air inleakages—Feed water treatment—Tube cleaning—Ancillary plant—Efficiency—Economics—Boiler selection.

XXI. ELECTRIC FURNACES 499

Electrical terms—Conversion of electricity to heat—Arc heating—Resistance heating—Types of electric furnace—Load characteristics of electric furnaces—General operation of the various types—Electric supply and furnace operation.

XXII. REFRACTORY AND INSULATING MATERIALS 515

Refractories : Influence of refractories on fuel economy—Types of refractories—Raw materials—Manufacture—Constitution of refractories—The chemical and physical properties of refractories—The action of slag, gases and vapours—Jointing cements—Selection and testing.
Insulating materials : Classification—Conductivity—Insulating firebricks—Lagging and lagging materials.
Construction : Methods of building refractory structures and insulating walls.

XXIII. UTILISATION OF STEAM FOR INDUSTRIAL PROCESSES .. 552

Steam heating—Peak loads—Advantages of steam—Effect of heat gradient—Pressure and heat losses—Heat extraction from steam—Superheated steam—Heat transference—Heating systems—Steam circulating systems—Desuperheating steam—Methods of handling condensate—Multiple effect evaporation—Pressure hot water for heating processes—Fuel economy.

XXIV. CENTRAL HEATING 606

Space heating—Methods employed for central heating—Types of boilers used—Boiler firing—The correct management of central heating and hot water supply.

XXV. DRYING 631

Vapour pressure and the principles of drying—The practice of drying—Types of drying plant—Dust collection—Control of conditions—Thermal efficiency—Technical data—Heat losses—Methods of heating, furnaces and fuels.

CHAPTER PAGE

XXVI. THERMOSTATIC CONTROL IN STEAM AND HOT WATER PRACTICE 661

The importance of automatic control—Control in central heating—Hot water supplies—Process heating—Recovery of waste heat from internal combustion engines—Steam boilers—Control equipment—General technique.

XXVII. AUXILIARY PLANT FOR THE EFFICIENT UTILISATION OF GAS 672

Nomenclature—Industrial gas fitting—Governors—Burners—Flues—Cocks—Temperature control—Thermostats and pressurestats—Pilots, cut-offs and back pressure valves—Meters.

XXVIII. OIL AND ITS UTILISATION 696

Nature of petroleum—Distillation and cracking—Fuel oils—The utilisation of fuel oils; pretreatment; burning technique; burner maintenance—Criteria for fuel oils; testing.

XXIX. GRAPHICAL HEAT BALANCES 709

General principles—The construction of graphical heat balances—Heat flow diagrams—The heat requirements of different systems as shown by heat flow diagrams.

XXX. INSTRUMENTS 719

Measurement of composition of flue gases and the instruments used—Measurement of draught—Temperature measurements; thermometers, thermocouples and pyrometers—The accurate measurement of flue gas temperature: fine wire thermocouples, screened thermocouples, suction pyrometers—Measurement of surface temperature of external walls—Heat flow gauges—Measurement of smoke—Boiler feed water measurement—Maintenance of instruments.

XXXI. SAMPLING SOLID FUELS 747

Principles of sampling and standards of accuracy—Sampling (*a*) large and (*b*) small coal for all purposes other than screen analysis—Preparation of laboratory sample for chemical analysis—Sampling waggons—Sampling for screen analysis—Sampling coke for moisture, analysis and shatter test.

XXXII. THE SAMPLING OF GASES 759

The problems and precautions in sampling gases—Selection of position for sampling—Suspended matter—Temperature effects and their control—Other factors vitiating the true composition of the sample; effect of materials and air leaks—Rate of sampling—Transport—Contamination of samples; solutions for use in sampling.

XXXIII. ANALYSIS OF SOLID FUEL 776

XXXIV. THE SELECTION OF FUELS FOR INDUSTRIAL PURPOSES .. 779

Cost and operational factors on which selection must be based—The characteristics of the fuels available.

APPENDIX I. THE SURVEY OF BRITISH COALS 788

APPENDIX II. STEAM TABLES 790

APPENDIX III. ENTROPY 797

The notation used in many of the Figures differs from that used in the text (notably "lbs." in the Figures and "lb." in the text as the abbreviation for "pounds"). This has arisen from the necessity of using existing blocks for the illustrations.

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CHAPTER I

INDUSTRIAL FUELS: COAL

Fuel conservation—Origin and nature of coal—Properties of coals—Grouping of coals—Main uses—Industrial characteristics—Sulphur—Coal preparation and cleaning—The size of coals.

WITH the exception of petroleum oils, substantially all fuel used in this country either consists of raw coal or is manufactured from coal; coal is therefore the basis of fuel and power in Great Britain.

FUEL CONSERVATION

Fuel conservation has two main aspects; (1) As a long-term policy affecting a national asset of irreplaceable raw material. (2) As a short-term policy to meet the urgent need for reducing consumption during the war.

The importance of the long-term policy to posterity requires a little attention. According to Bone and Himus,* the net coal which will be available at the surface will be about 170,000 million tons. This estimate is that of Stanley Jevons, in 1915, of reserves within 4,000 feet of the surface, taking 1 foot as the minimum workable thickness of seam.

In 1913, 287 million tons of coal were mined, and in 1938, 227 million tons. It might therefore be supposed that coal will last from 580 to 749 years at these rates of production. This gives an agreeable sense of security but, as Bone and Himus point out, " the most important aspect of the coal question for Great Britain is not so much how long can our reserves last, but how long can we continue to get the available coal at a cost which shall not place us as a nation at a disadvantage relative to our nearest competitors." The answer is not something of the order of 500 to 800 years, but possibly some 50 years or less. A more recent analysis of the position in 1940 by the then President of the Institution of Mining Engineers, Mr. Forster Brown (Presidential Address, 1940) fully confirms this estimate of the position and emphasises the probable increase in coal-getting costs. A shortage of certain special types of coal, such as coking coals, may well be experienced within the next two generations. Early consideration will have to be given either to forbidding the use of these coals, except for special purposes for which they are peculiarly adapted, or to finding a means of making other types of coal suitable for coke manufacture.

From the war aspect—the short-term policy—the maximum effort must be made now to use fuel with the greatest efficiency. The object of this handbook is to assist in attaining this goal; at the same time the necessity for reorganisation of the methods of coal preparation and utilisation after the war is clear.

THE ORIGIN AND NATURE OF COAL

The term " coal " is applied to those rocks in the earth's crust produced by the decay of plant materials, accumulated millions of years ago. Shales impregnated with carbonaceous substances are excluded. Scientists have identified in coal, traces of the structure of known plants and even portions of the trees themselves, which have suffered only a slight change of form.†

* " Coal, Its Constitution and Uses," p. 17, Longman, Green & Co., London, 1936.

† J. G. King: " The Preparation, Selection and Distribution of Coal," Chapter I (The Coal Trade Luncheon Club and the Institute of Fuel, 1931).

Two theories have been suggested to explain the manner in which these accumulations of plant remains have collected in sufficient quantity to form coal seams. The first—the growth in place or " in situ " theory—holds that the coal seam marks the area in which the original material grew and accumulated. The second—the " drift " theory—holds that the material was transferred by rivers from the area in which it grew, to a lake or estuary in which it was deposited. It is probable that both methods of accumulation have been operative.

The best example of a thickly forested fresh-water swamp, illustrating the " in situ " theory, is the Great Dismal Swamp of Virginia, North Carolina (U.S.A.) The water is shallow and the soil below consists of 10–15 feet of partially decayed trunks, branches, leaves, roots and seeds of the continually falling trees ; sand and other impurities are absent. Coastal swamps of a similar nature are abundant in the tropics.

As an example of the " drift " method may be cited the large quantities of material consisting of trunks, branches, etc. carried by the Mississippi—particularly at flood time—to the mouth of the river where the water-logged condition of such material causes it to be deposited. As it may happen that at the mouth of the river there is already growing an estuarine swamp, it will be seen that both methods may contribute to the same area of accumulation.

On dry ground, fallen trees and other dead plants are attacked by the oxygen of the air, and the cellulose, their chief constituent, is converted into water and various gases so that in a comparatively short time they have rotted away, leaving practically no trace. Under water, however, when air is excluded, plant debris is decomposed by the action of bacteria. This decomposition continues until the substances produced become so concentrated that the bacteria can no longer live ; the extent of the decomposition, therefore, depends on whether or not the products can drain away. Certain parts of the plant are always more resistant to decay than others, and fragments of plants only slightly altered, particularly spore exines (the tough outer jackets of spores), bark, or its cuticle (outside skin) can be identified in coal.

As a result of geological agencies which caused the surface of the earth to fall and rise alternately, the accumulations of partly decayed plant material have been covered by layers of sedimentary rocks, and then further plant remains have been deposited. In this way a sequence of seams has been built up.

The changes that took place after the cessation of bacterial decay may have been even more profound. The influences then operating were heat and pressure, the pressure being exerted not only vertically by the weight of the overlying strata, but also sideways by the powerful earth movements which have resulted in the folding and compression of the seams. The chemical action which took place was the elimination of water and oxides of carbon, resulting in a reduction of the amount of material that can be obtained from the coal in the form of volatile vapours and gases (the " volatile matter ") by the action of heat. There is some doubt how far these subsequent changes in the coal seams have been due to heat. Normally, heat would be indirectly connected with the pressure, but the temperature of parts of a few seams has been raised by molten igneous intrusions from below.

VARIATIONS IN COAL

The result of the action of natural agencies upon the plant remains has not always been the same. These variations may be considered under the following headings :—

(1) The nature of the original vegetation and its separation by the grading action of water.
(2) The extent of bacterial decay.
(3) The extent of the removal of volatile matter by heat and pressure.

(1) Continuous seams of coal are known, for example in South Wales, in which the character of the coal changes steadily in a lateral direction, from bituminous to anthracite (see later) ; microscopical examination shows that the same types of plant were present over the whole area. The difference cannot, therefore, be accounted for by changes in vegetation. The segregation of different parts of similar vegetation can, however, produce a marked difference in composition. For example, the dull bands (durain) which occur in many seams contain more plant remains than the bright bands adjacent to them. Other plant fragments also bear evidence of having been water borne. This suggests that some segregation of the parts of the plant by the action of water occurred and produced a difference in the final coal.

(2) The extent to which bacterial action proceeded was probably influenced by drainage, since the decomposition normally continues until the products accumulate to such an extent as to arrest decay.

(3) The heat and pressure to which a coal has been subjected would depend upon the depth of the superimposed strata and the degree of earth movement. In areas where the seams have remained relatively undisturbed, the oldest are frequently lowest in volatile matter, the percentage tending to decrease with depth. This was stated by Hilt to be of fairly general application, but several noted exceptions occur. In the South Wales seams mentioned above, the anthracitisation occurs in the north-west of the coal-field where there has been most disturbance. It is, therefore, possible that pressure and heat are responsible for the change. In Scotland, parts of certain seams are changed to a pseudo-anthracite by an entirely different cause, viz., by the heat derived from the intrusion of molten magma.

GLOSSARY OF TERMS—SOME CONSTITUENTS AND CHARACTERISTICS OF COALS

Before proceeding to discuss the types of coal available it is necessary to explain certain terms which are used when describing coals.

Moisture. The amount of moisture retained by a coal may vary between fairly wide limits. By exposing coal in a thin layer in a dry, well-ventilated place, the coal loses " free " or " surface " moisture, and dries to an " inherent " or air-dried moisture content in equilibrium with the moisture in the atmosphere. At a given humidity, the inherent moisture content of coal, determined by heating the air-dried coal to 110° C., is characteristic of that particular type of coal and varies, with different British coals, from below 1 per cent. to about 15 per cent. The free or surface moisture is mechanically adhering moisture which may come from the operation of wet cleaning processes or which may be due to exposure to rain.

Ash. The ash in industrial coal is determined by the complete combustion of a weighed sample. It includes the " inherent " ash intimately associated with the coal and partly or mainly derived from the original coal plants, and adventitious ash derived from shale, clay, pyrites, ankerites (white partings) or dirt from bands in the coal seam, and fragments of the roof or floor which have become included in mining the coal. The adventitious ash-forming constituents can be removed by physical means, but the inherent ash-forming constituents cannot be removed in this way. The ash in British coals cleaned by floating in a solution of specific gravity of 1·6 amounts, on the average, to between 4 and 4·5 per cent., ranging from 1 to 8 per cent. for individual coals. The ash in British coals cleaned by floating in a solution of specific gravity of 1·35 amounts, on the average, to about 2·8 per cent., ranging from 1 per cent. to about 5 per cent. for individual coals.

The amount of ash as determined is less than the mineral matter originally present by the amount of water and carbon dioxide driven off from clays

during combustion, and due to other minor changes, as, for example, the oxidation of pyrites. Generally it can be said that the weight of ash is about 10 per cent. less than the weight of original mineral matter, i.e. if the amount of "ash as determined" is 20 per cent. the amount of inert mineral matter frequently referred to as "ash" is really 22 per cent.

The amount of ash and its fusion temperature have an important bearing upon the value of coal in industry (see Chapter VI).

Volatile Matter. The volatile matter which, it should be made clear, does not include the moisture in the coal, is defined as the percentage loss in weight when 1 gm. of coal is heated under carefully controlled conditions for seven minutes at 925° C. in a crucible from which air is excluded (cf. Chapter XXXIII).

The loss consists of water (derived from the decomposition of the coal and not from surface, or inherent, moisture), gas and tar; the percentage of volatile matter (on the dry coal) when expressed on the dry ash-free basis, supplies considerable information as to the probable behaviour of the coal in use.

Dry Ash-Free or Dry Mineral-Matter-Free Coal. For the purpose of comparing certain characteristics of coal, and for grouping coals, it is customary to use figures based on dry ash-free coal, the comparison then being on the actual coal substance irrespective of adventitious material; for instance, if a coal contains 5 per cent. moisture and 5 per cent. ash—a total of 10 per cent. of material which can be termed "inert"—and has a calorific value of 12,000 B.Th.U., this value on the dry ash-free basis will be :—

$$12{,}000 \times \frac{100}{90} = 13{,}333 \text{ B.Th.U.}$$

Similarly, if the determined amount of volatile matter (less moisture) be 36 per cent., it would be $\frac{36}{0{\cdot}9} = 40$ per cent. on the dry ash-free basis. To represent approximately the mineral matter, the weight of ash should be increased by 10 per cent., and the calculation will then give results on a dry mineral-matter-free basis.

For exact scientific work the dry mineral-matter-free basis should be used. One formula that can be used for this purpose is due to King, Maries and Crossley, which states :—

Mineral matter = 1·09 × per cent. ash + 0·5 × per cent. pyritic sulphur + 0·8 × per cent. CO_2 evolved on treatment with acid − 1·1 × per cent. SO_3 in ash + per cent. SO_3 in coal + 0·5 × per cent. chlorine in coal.

A simpler method is due to Parr :—

Mineral matter = 1·08 × per cent. ash + per cent. moisture + 0·55 × per cent. sulphur.

The King-Maries-Crossley formula assumes the coal to be dry. The Parr formula takes the moisture into account and also assumes that all sulphur is present in the form of pyrites.

Fixed Carbon. The residue left after heating the coal in a closed vessel to drive off the volatile matter is the "coke." This residue contains all the inorganic constituents present in the original coal that go to form the ash.

The percentage of solid residue minus the ash = "fixed carbon."

The solid residue also contains hydrogen, sulphur, nitrogen and oxygen in addition to carbon. It may vary from a black powder to a highly porous button, and its appearance, strength and cellular structure are significant characteristics.

Caking and Swelling Power. The appearance of the residue from the volatile matter determination is a guide to the property which a coal possesses of

forming a coherent coke. A more reliable method of assessing this property is to use an assay test, one example of which is the Gray-King.* In this assay, 20 gm. of the coal are heated in a silica tube in a special furnace to 600° C. (1,112° F.). The appearance of the resulting residue (shown in Fig. 1) is then compared with a series of standard cokes and its position determined in the series, which ranges from non-caking up to highly-caking coals. The letters A to G are used to designate cokes which range from those given by a non-caking coal (Type A) to those which are hard and strong and are of the same volume as the original coal (Type G). For coals more strongly swelling than G, subscripts are added, e.g. G_1, G_2, G_3, etc. up to G_{10}, in order to indicate the degree of swelling.

The swelling properties may be determined by the Woodall-Duckham crucible swelling test described in British Standard Specification 1016 of 1942, page 64. The remarkable differences in the swelling power of coals are illustrated in Figs. 2 and 3 of coke buttons obtained in the Woodall-Duckham swelling test and in the Lessing method of determination of volatile matter.

The relationship between the results of these methods is as follows :—

W–D crucible swelling test	Gray-King	Lessing
1	A–B	1
1½	C–D	1½
2 –2½	D–E	2
3	D–G	2½
4 –4½	G_2	3
5 –5½	G_3	3½
6½–7	G_5	4
9	G_8–G_{10}	5

Calorific Value. The calorific value of a coal is a measure of its total heating power. It may be expressed as the number of British Thermal Units evolved when one pound of the coal is completely burned (or the number of calories per gram). The calorific value of fuels will be discussed in Chapter V.

Ultimate Composition. In addition to the determinations of moisture, ash, volatile matter and fixed carbon, which are grouped together under the term " proximate " analysis, the " ultimate " composition may be determined.

This involves determinations of the major elements present in coal. Carbon, hydrogen, nitrogen and sulphur are determined by analysis, oxygen being ascertained by difference. Phosphorus and chlorine may also be determined. These determinations, unlike those of the " proximate " analysis, call for considerable chemical skill.

TYPES OF COALS

In considering the various types of coal, peat must be mentioned since it may bear a resemblance to the progenitor of coals.

Peat. Peat in the bog or swamp may contain more than 90 per cent. of water. At the top it is light in colour, but at lower depths the peat becomes darker and finally nearly black, by which time it is not so obviously of vegetable origin. It is frequently low in ash, but for use as a fuel the water content presents a serious problem, and much money has been spent endeavouring to dry peat on a commercial scale. To meet local fuel requirements, hand cutting and drying in stacks has been practised for many centuries. There are extensive deposits in England and Scotland besides the more famous ones in Eire.

* Fuel Research Survey Paper No. 44.

Lignites and Brown Coal. Lignites are intermediate in character between peat and normal coals. They are of great importance in many parts of the world, but occur only in small pockets in Great Britain, of which the best known is at Bovey Tracey. A number of varieties are recognised, namely, (1) woody or fibrous brown coal, (2) earthy brown lignite, compact but friable, (3) brown coal showing a cleavage and a slight woody structure, and (4) black lignite resembling true coal in appearance. Although given in this order it is not implied necessarily that these substances represent a sequence in the transition of peat-like substances to coal. They are all characterised by high moisture contents (30–50 per cent.) and break down readily to slack on weathering.

It would appear that no real attempt has been made to distinguish between lignite and brown coal. The dictionary defines lignite as " mineral coal retaining the texture of the wood from which it was formed." It would therefore be reasonable to define lignite as a solid fuel more mature than peat, which shows distinct woody structure when examined by the naked eye or under a pocket lens, and to consider brown coal as being a still more mature fuel, and devoid of obvious woody structure. The lignite would have a higher moisture content (50 per cent.) and lower carbon content than the brown coals (moisture 30 per cent.).

Bituminous Coals. The term " bituminous coal " has been applied to coals varying, on ash-free dry material, from about 75 per cent. carbon to about 91 per cent. carbon.

As the carbon increases, the oxygen of the coal decreases. The volatile matter varies on dry ash-free coal from about 45 per cent. for a coal containing 75 per cent. of carbon down to about 23 per cent. for a coal containing 91 per cent. of carbon. The calorific value of the dry ash-free material increases with the carbon content.

All bituminous coals show a banded structure due to bright and dull coal arranged parallel to the bedding planes. These bands have been classified by Stopes into four types—vitrain, clarain, durain and fusain. Without entering into further details here it may be said that vitrain and clarain together constitute the bright parts of the coal, and durain the dull and harder coal, known as Hards or Splints. Durain, as a rule, has a higher ash content than bright coal from the same seam, but the ash usually fuses at a higher temperature. Fusain is the dull and friable material usually found in thin layers along the bedding planes, although occasionally bands are found. It is the material known to miners as " Mother-of-Coal." The division of a seam into bright and dull coal, when the bands are thick enough to separate, is often of great importance industrially, since the two constituents may have appreciably different properties. For example, the dull coal (Hards) of the East Midlands coal-field is much favoured for locomotive firing. This is not infrequently due to the fact that the ash has a high fusion temperature.

Semi-Bituminous Coals. Coals intermediate between bituminous coals and anthracite vary in carbon content from 91 to 93 per cent., and in volatile matter from about 23 per cent. to 10 per cent. on dry ash-free coal.

Anthracite. Coals of more than 93 per cent. carbon and 10 per cent. or below of volatile matter, are termed anthracites. They have a high lustre and are sometimes graphitic in appearance, but evidence of banded structure can still be found by careful examination. Most anthracites occur in the older geological formations, although not all the older coals are anthracitic.

Other Types. There are several other coals which are regarded as distinct types. One of the more important of these is cannel. It is hard and particularly rich in plant remains so that the volatile matter may be as high as 56 per cent.

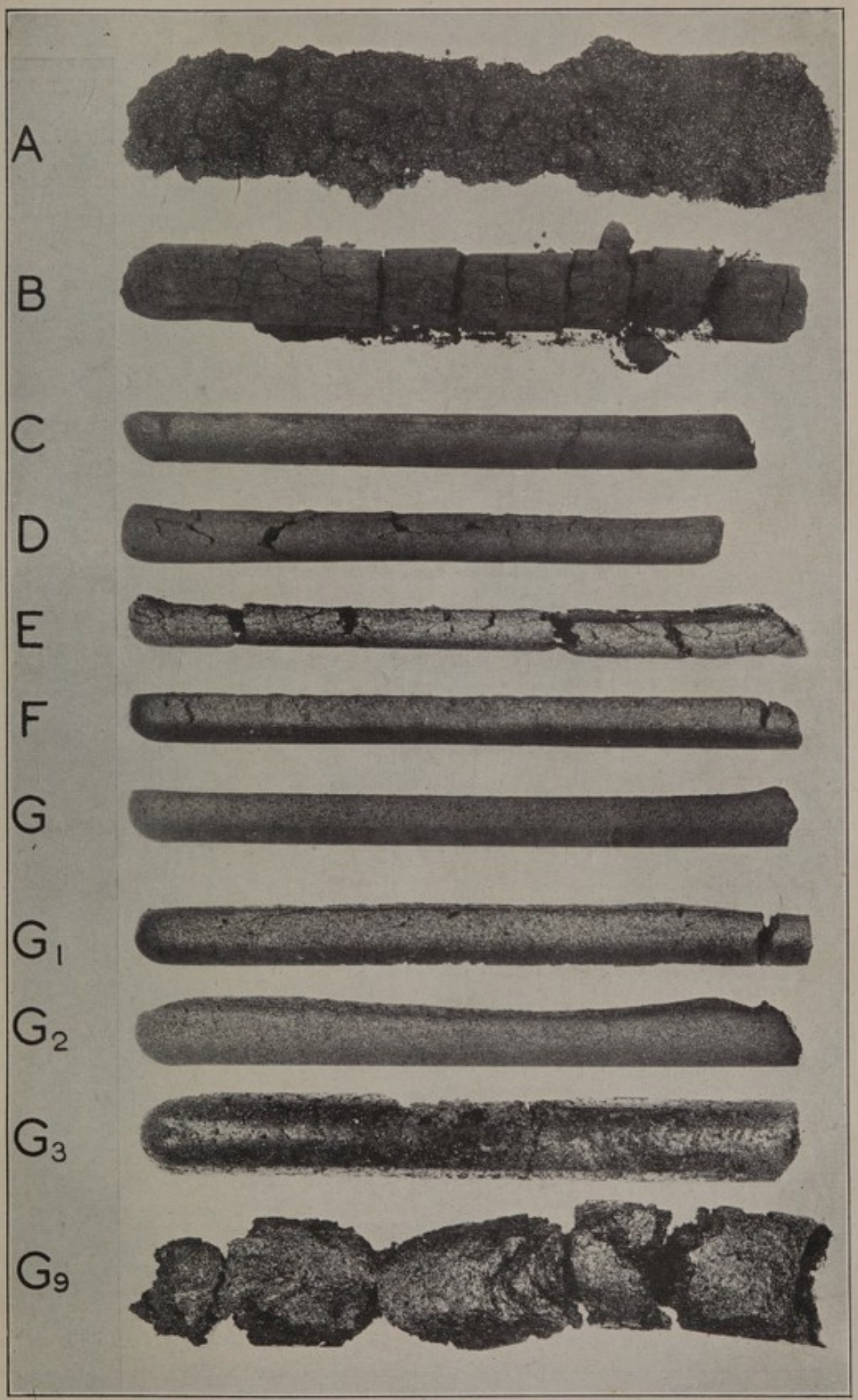

FIG. 1. Residual cokes from the Gray-King assay.

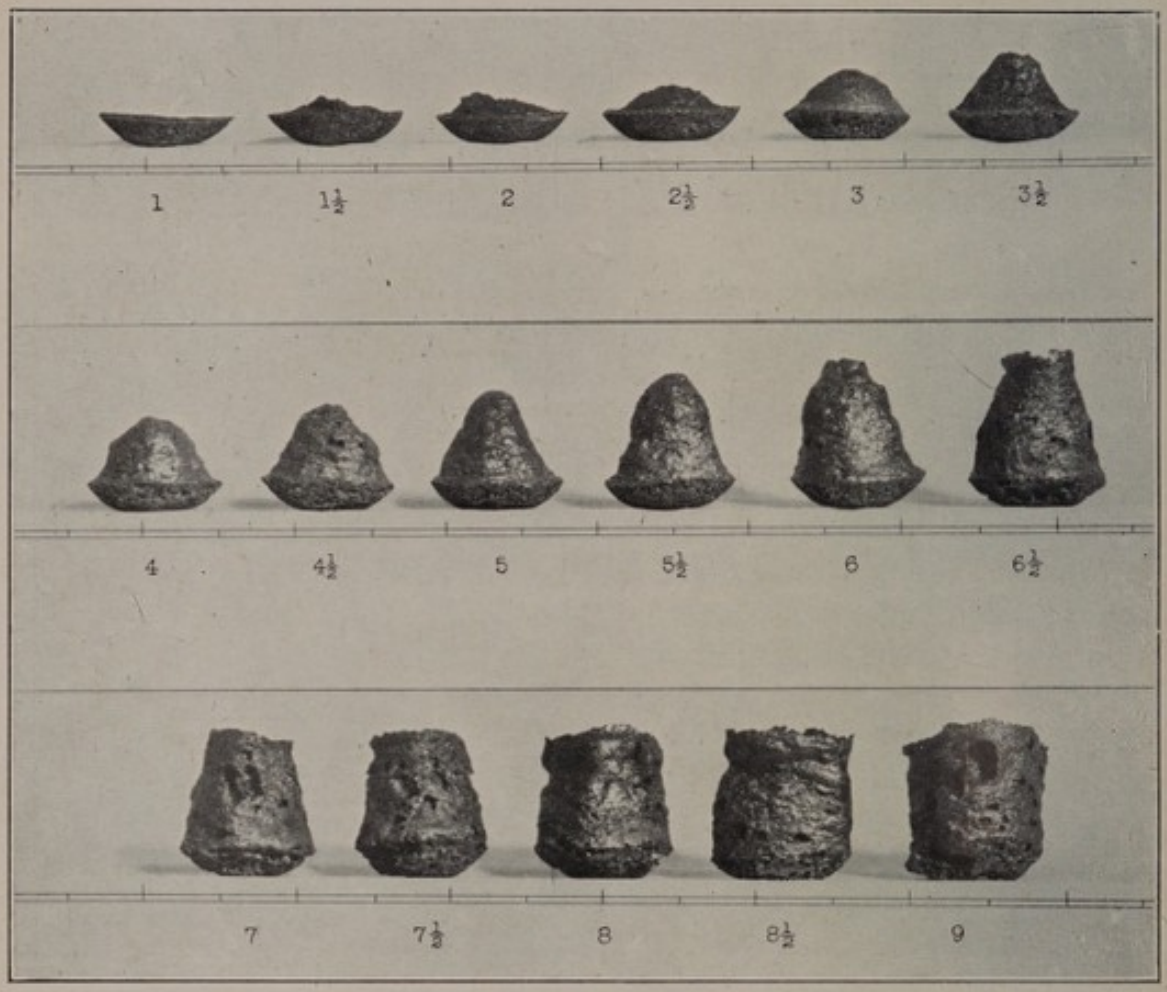

FIG. 2. Coke buttons from the Woodall-Duckham swelling test (British Standards 804 and 1016).

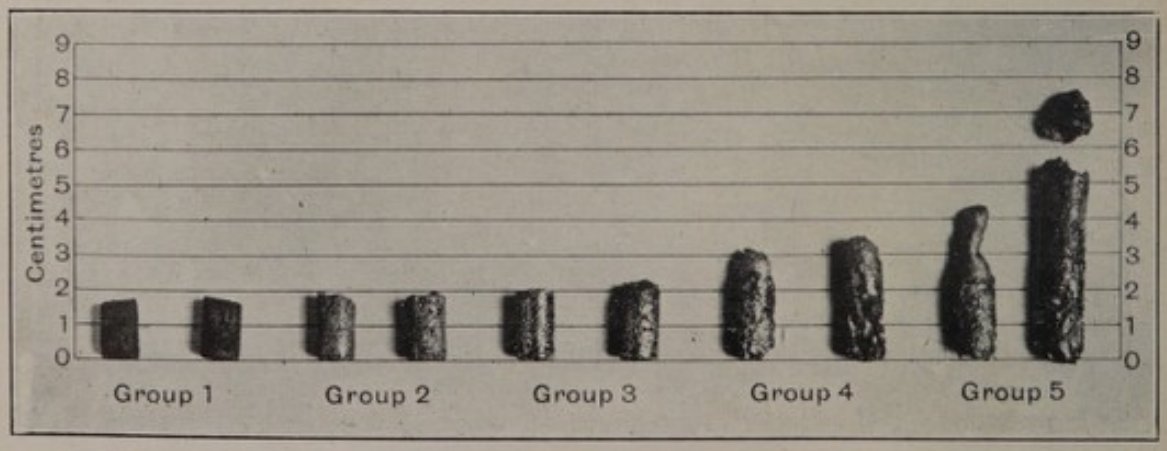

FIG. 3. Residual cokes from the Lessing test.

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GROUPING OF COALS

The above description of coal types constitutes a rough grouping. Many attempts have been made to devise comprehensive classifications of coals in order that the properties and uses of a coal may be predicted from the analysis of a sample. In this country the classification due to Seyler has probably had the biggest vogue, but with the development of fuel technology, all classifications have a decreasing value, since wider ranges of coal can now be utilised than would have been possible a few years ago. For example, the varieties of mechanical stoker now employed in boiler practice enable many coals to be used for steam raising which were not so suitable previously. A coal can be obtained to suit a particular form of stoker, or a stoker installed to burn a particular type of coal. Developments in the gas-making industry, such as the use of blending and the development of vertical retorts, have added to the range of coals suitable for this purpose. Blending is also improving the value of many coals for coke manufacture. There is, therefore, an increasing tendency to rely upon a simplified classification, especially for bituminous coals. A classification of this nature is given in Table 1.

TABLE 1. CLASSIFICATION OF BITUMINOUS COALS

Description	Group 1	Group 2	Group 3	Group 4
	Freeburning or non-caking	Slightly caking	Medium caking	Strongly caking
Analysis of Dry Ash-Free Coal :—				
Carbon, per cent.	78–81	81–82·5	82·5–84	84–89
Hydrogen, ,, ,, ...	5·1–5·6	5·2–5·6	5·2–5·6	4·5–5·5
Volatiles, ,, ,, ...	45–41	42–38	39–35	37–25
Calorific value				
B.Th.U./lb.	13860–14400	14400–14670	14670–15030	15030–15660
cals./gm.	7700–8000	8000–8150	8150–8350	8350–8700
Inherent moisture, per cent...	16–10	10–7	7–4	4–2
Type of coke, Gray-King Assay	A B	C D E	F to G_8	G_2 to G_9
W–D Swelling test	1	1½–2½	3–4½	5–9

It may prove desirable to divide group 4 into two groups, as follows :—

	Group 4	Group 5
Carbon, per cent. ..	84–86·5	86·5–89
Volatiles, ,, ,, ..	37–32	32–25
Calorific value :		
B.Th.U./lb. ..	15030–15550	15550–15660
cals./gm.	8350–8650	8650–8700

Group 5 would contain coals used for making coke with a high shatter test and generally non-reactive.

It will be noted that the calorific value and the percentage of carbon of the dry ash-free coal, as well as the caking and swelling properties, increase in passing from Group 1 to Group 4. On the other hand the percentage of volatile matter and of inherent moisture decrease in the same order.

The majority of industrial bituminous coals fall into one or other of the groups. There are exceptions, as, for example, " Hards " which, in general, do not cake ; there are also marginal cases. The main purpose of the grouping is to indicate certain properties which have an important bearing on industrial uses.

The distribution of these coals in the various coal-fields of England and Scotland is indicated in the following table, which should be considered as a preliminary guide pending more complete examination.

GROUP 1—COALS are almost devoid of caking properties.

Warwickshire, Leicester and South Derbyshire, Cannock Chase and South Staffordshire	All coals.
Derbyshire	Pits in extreme south.
Nottinghamshire	Pits in southern part of eastern area.
Lancashire	Top seams in a very few pits.
Northumberland	Upper seams in extreme north.
Scotland	Many coals in Fifeshire and the Lothians.

GROUP 2—COALS are often described as free-burning, but they have slight caking properties and are slightly swelling.

Derby and Notts	A large number of coals.
Yorkshire	The Shafton seam, and the Barnsley and High Hazel seam in the Doncaster area.
North Staffordshire	The Great Row seam, Woodhead seam, and many coals on the east side.
Lancashire	The upper seams such as Florida and Crombouke.
Northumberland	Many coals north of Backworth, especially upper seams.
Scotland	Many in Fifeshire, the Lothians and Ayrshire—also in Stirling and Lanark.

GROUP 3—COALS are decidedly caking, and swell moderately, and they include a number of coals which are used for gas-making and some coals used in coke ovens. If used in coke ovens they make what is known as " reactive " coke.

Derbyshire	Especially north-west area.
Nottinghamshire	Very few.
Yorkshire	Most seams in South Yorkshire except the Sheffield-Barnsley and Doncaster areas. Coals in West Yorkshire.
North Staffordshire	Most of the coals in the west.
Lancashire	Trencherbone downwards.
Northumberland	South of Backworth—also lower seams toward north.
Scotland	Certain seams in central area.

GROUP 4—COALS are strongly caking and highly swelling and are chiefly valued for their use in coke ovens to produce a hard and generally unreactive metallurgical coke. The lower end of the group includes the Durham gas coals.

Yorkshire	In the Sheffield-Barnsley area—also some in West Yorkshire.
North Staffordshire	A few seams in the west.
Lancashire	Lower Mountain Mine, and bottom seams such as the Arley.
Durham	Practically all coals.
Northumberland	Extreme south, lower seams.
Scotland	North Central area—Dumbarton, Lanark, Stirling.

SOUTH WALES COALS

The scheme of classification put forward for bituminous coals does not apply to the South Wales coals since their volatile matter content and coking properties are different from those of coals mined in other parts of the country. Coals generally are found to fall on a band, known as the " coal band " when their hydrogen content is plotted against their carbon content, both being taken on the dry ash-free (or dry mineral-matter-free) basis. When the South Wales coals are thus treated the result is as given in Fig. 4. It will be seen that the carbon content of the coals on the dry mineral-matter-free basis extends from 84 per cent. to 94·5 per cent., and the hydrogen content on the same basis is from 5·5 per cent. to 2·5 per cent.

It must always be remembered that a classification based on the inherent chemical properties gives a general picture of the types of coal, but it ignores such factors as size and ash content which, frequently, are very important in the selection of coals for specific purposes.

A general classification of South Wales coals similar to that of Table 1 is given in Table 2.

The volatile matter content, corrected for CO_2 in the dry ash-free coal, is a reliable first guide to the general properties of most of the commercial grades of South Wales coals. It can be made the basis of a broad classification if due

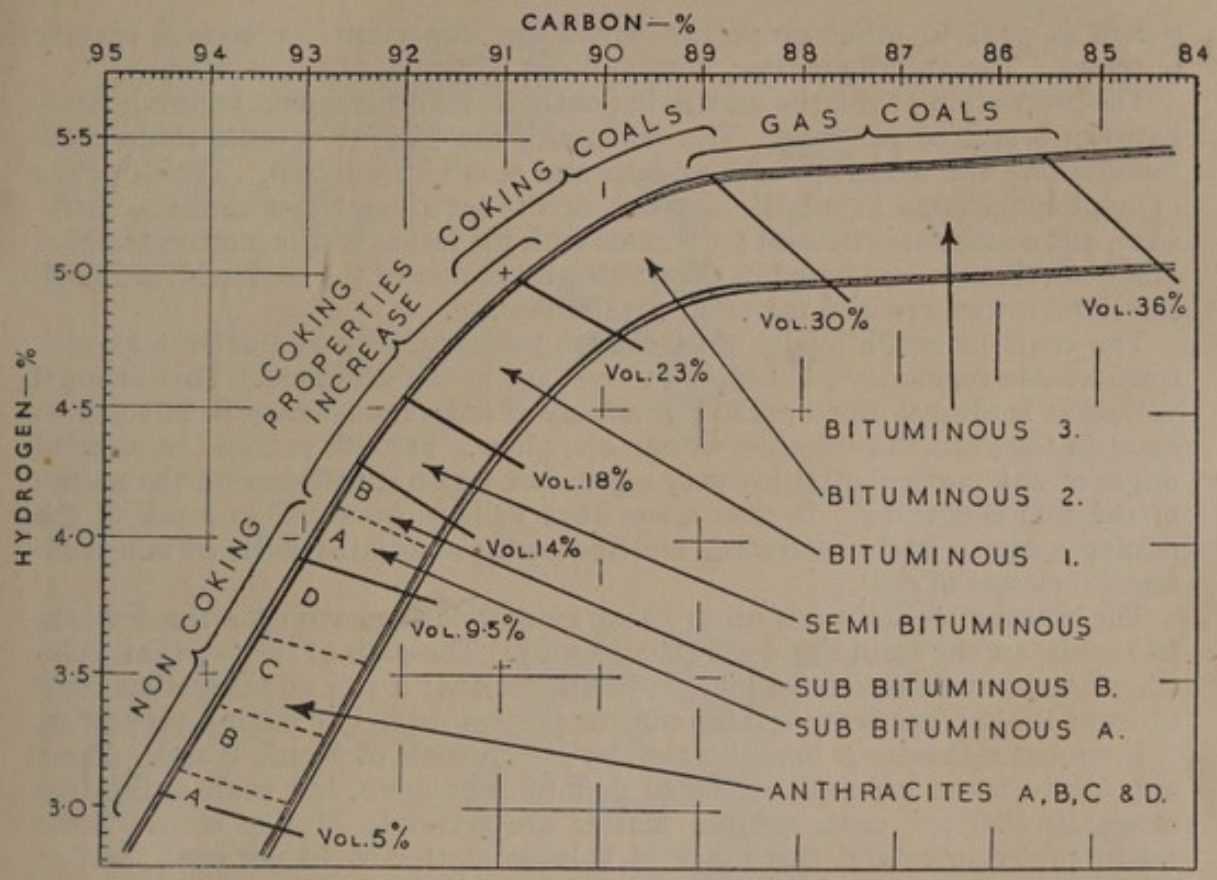

FIG. 4. The South Wales coal band. All quantities in this figure are on the dry mineral matter-free basis.

TABLE 2. CLASSIFICATION OF SOUTH WALES COALS

Description	Anthracite	Sub-bituminous	Semi-bituminous	Bituminous		
				1 Caking steam, coking, blending coals	2 Coking coals	3 Gas coals
	Non-caking	Non-caking to slightly caking	Slightly to medium caking	Caking coal	Strongly caking	Less strongly caking
Analysis of Dry Ash-Free Coal :—						
Carbon, per cent.	94·5–92	93–91	92·5–90·5	92–90	91–88	88–84
Hydrogen ..	2·8–3·9	3·9–4·25	4·0–4·5	4·3–5·0	4·8–5·3	5·0–5·5
Volatile matter	4·5–9·5	9–14	13–18	17–23	22–30	29–36
Calorific value B.Th.U./lb...	15250–15600	15600–15760	15600–15800	15650–15800	15800–15500	15500–15050
Cals./gm. ..	8472–8667	8667–8755	8667–8778	8694–8778	8778–8611	8611–8361
Inherent moisture, per cent.	1–3	0·6–1·0	0·6–1·0	0·6–1·0	1–2	1–3
Type of coke (Gray-King assay) ..	—	A B	G_2, G_4, G_1, G	G_9, G_{10} G	G_{10}	G

regard is paid to other properties which are important for certain specific purposes. The volatile matter, however, can serve only as a pointer.

The proportion of volatile matter in South Wales coals ranges approximately between 5 and 36 per cent. The coals with the highest volatile content are found along the eastern, south-eastern and southern outcrop. The direction of maximum change in volatile matter is east to north-west, south-east to north-west, and south to north-west; the coals with the least volatile matter, the high grade anthracites, are found in the north-west corner of the coal-field, although small quantities are also mined in Pembrokeshire.

The coals of South Wales thus change profoundly in character and with considerable regularity as they are traced across the coal-field. This regional variation is always much greater than any change that occurs from seam to seam in the vertical succession at any one place. For this reason the place of origin of any particular coal is very significant for on it will depend the nature of the coal concerned. It thus arises that valleys, in which so much of the mining activity is concentrated, are frequently associated with certain commercial classes of coal.

The anthracites and coals up to 10 per cent. volatile matter (on the dry ash-free basis) are shown in Fig. 4 to be non-caking. The coals of 13·5 per cent. volatile matter have incipient caking properties, so that it can be taken that coals of from 10–13·5 per cent. volatile matter possess negligible caking properties.

Frequent reference is made to the dry steam coals of South Wales. There appears to be no generally accepted definition of these, but coals within the range 10–13·5 per cent. volatile matter are certainly of this class. Some might prefer to extend this range of volatile matter to 14 per cent., but at 14–14·5 per cent. the coals have measurable caking properties.

The following facts illustrate the rapid increase of caking properties as the volatile matter increases up to about 23 per cent. A coal of 16 per cent. volatile matter has well-defined caking properties. A coal of 18–19 per cent. volatile matter has marked caking properties. There is a further increase in caking properties between 19 and 23 per cent. volatile matter until a coal with 22 to 23 per cent. is used for making a first-class metallurgical coke.

Coals with more than 30 per cent. volatile matter show a slight decrease in caking power. There are no high-volatile, non-caking or feebly-caking coals in South Wales. The South Wales coals with more than 30 per cent. volatile matter are usually described as gas coals.

When comparing South Wales coals with coals from other coal-fields, certain factors should be noted. For instance, the free-burning coals of South Wales have, compared with the free-burning coals of elsewhere, a much lower percentage of volatile matter, a much lower percentage of inherent moisture, less hydrogen and a considerably greater calorific value on the dry ash-free basis. On the other hand, for combustion they require a greater draught.

With the exception of slight variations which only occur at the extreme ends of the range, all these coals have a high fairly constant calorific value and low inherent moisture.

Most of the South Wales coals have an ash fusion temperature of over 1,350° C. in an oxidising atmosphere and above 1,300° C. in a reducing atmosphere. Many have a much higher ash fusion temperature, but coals with less than 10 per cent. and greater than 30 per cent. volatile matter tend to have a lower ash fusion temperature than the others. It is, however, not uncommon for some of these to be quite high.

THE COAL-FIELDS OF GREAT BRITAIN

Geological evidence suggests that the original vegetable debris was probably laid down in four large tracts (excluding the Kent field), but these have since



FIG. 5. The coal-fields of Great Britain.

1. Scotland.
   (a) Fife and Clackmannan.
   (b) Lothians.
   (c) Central coalfield (Lanarkshire, etc.).
   (d) Ayrshire.
2. Northumberland.
3. Durham.
4. Cumberland.
5. Lancashire and Cheshire.
6. North Wales.
7. Yorkshire.
8. Derbyshire.
9. Nottinghamshire.
10. North Staffordshire.
11. South Staffordshire and Cannock Chase.
12. Leicestershire and South Derbyshire.
13. Warwickshire.
14. Forest of Dean.
15. Bristol and Somerset.
16. East Kent.
17. South Wales.

become separated into the forty detached areas now known, of which only twenty have any importance. These four areas were :

(1) Scotland.
(2) Northumberland, Durham and Cumberland.

(3) North Wales, Lancashire, Yorkshire and the Midlands.
(4) South Wales, Forest of Dean and the Bristol and Somerset coal-fields.

The distribution of the British coal-fields is shown in Fig. 5. The exposed coal-fields, where coal measures are actually at the surface, are shown in black. The concealed coal-fields, that is, those coal measures which are covered by later rocks but where the coals are within workable distance of the surface, are shaded.

A survey of the coal-fields of the country undertaken by the Fuel Research Board has been in progress for many years. Details of this survey are given in Appendix 1.

Table 3 gives the size and outputs of the coal-fields and indicates in some measure the types of coal produced in each field. This information is to be regarded as a preliminary guide pending the preparation of more detailed data. The groups mentioned refer to the classification in Table 1, except for South Wales.

TABLE 3. THE COAL-FIELDS OF GREAT BRITAIN

District	Approx. area. sq. miles	Output in 1938. Million tons	Types of coal produced
Scotland	1,700	30·3	Freeburning ; Group 1 : in Fifeshire and the Lothians. Slightly caking ; Group 2 : in Fifeshire, the Lothians and Ayrshire and also in Stirling and Lanark. Medium caking ; Group 3 : Certain seams in the central area (Lanark). Strongly caking ; Group 4 : North Central area—Dumbarton, Stirling and Lanark. Pseudo-anthracite : Lanark, Fifeshire, Stirling.
Northumberland ..	800 with Durham	13·3	Group 1 : A few in extreme north of area—upper seams. Group 2 : Most of Northumberland and north of Backworth, especially upper seams. Group 3 : South of Backworth ; also lower seams toward north. Group 4 : Extreme south, lower seams.
Durham	See above	31·4	Group 4 : Practically all Durham coals belong to Group 4.
Cumberland ..	150	1·6	Mostly Groups 3 and 4.
Lancashire and Cheshire	500	14·3	Group 1 : Top seams in a few pits—very occasionally. Group 2 : Upper seams such as Florida and Crombouke. Group 3 : Lower seams such as Trencherbone. Group 4 : Lower Mountain Mine seams—bottom seams, as Arley in many places.
North Wales ..	150	3·7	Mostly Group 3. Some border on Group 4.
Yorkshire	2,136 with Derby and Notts	42·4	Group 2 : Shafton seam and Barnsley High Hazel seams in Doncaster area. Group 3 : Coals in West Yorkshire district, most seams in South Yorkshire except in the Sheffield-Barnsley and Doncaster areas. Group 4 : Coals in the Sheffield-Barnsley area ; also some coals in West Yorkshire.

District	Approx. area sq. miles	Output in 1938 million tons	Types of coal produced
Derbyshire ..	See above	13·4	Group 1 : In extreme south. Group 2 : A large number of coals would probably be of this group. Group 3 : North-west area.
Nottinghamshire	See above	15·4	Group 1 : Southern part of eastern area. Group 2 : Most Notts. coals would probably be in this group. Group 3 : Very few.
North Staffordshire	100	6·9	Group 2 : Great Row and Woodhead seams; many coals on eastern side. Group 3 : Most of the coals on the west. Some might be classed as Group 4.
South Staffs. and Cannock Chase	150	6·6	Group 1 : All coals although one or two border on Group 2.
Leicester and South Derbyshire	84	3·6	Group 1 : All coals.
Warwickshire ..	60	5·6	Group 1 : All coals.
Forest of Dean ..	35	1·3	Mainly Group 3.
Bristol and Somerset	240	0·8	Mainly Group 4. Some border on Group 3.
East Kent ..	206	1·8	Group 4 : Low-volatile steam coal.
South Wales ..	1,000	35·3	See special table of South Wales coals (Table 2) (5·5 million tons of anthracite).

THE MAIN USES OF COAL

At this point it is appropriate to consider the principal characteristics, apart from size, required in coals for their main uses.

Coals for Steam Raising. Formerly it was considered that the type of coal required for steam raising needed to be free-burning, and the term "steam coal" was in general use. Since the development of forced draught, mechanical stokers and modern boiler plant, the range of coal for steam raising has been greatly widened, and every type of coal, from free-burning to strongly caking, is being used. With further developments it will, in future, be possible to burn all types of coal satisfactorily for steam raising, but in some existing plants, particularly where hand firing is practised and draught is limited, coals of Groups 1 and 2 will give the best results.

Small, hand fired boilers are inclined to smoke badly when fed with high-volatile coals, and for such boilers, low-volatile coals are more suitable if adequate draught is available.

The evolution of volatile matter depends upon the nature of the coal and affects its behaviour on the grate. High-volatile coals of Group 1 release their volatile matter rapidly, giving smoky flames and low firebed temperatures. Group 3 coals do not release their volatiles so rapidly, and the gases evolved are richer in hydrocarbons. The less rapid release of volatiles permits them to be burned with a better-regulated supply of air and thus to give hotter flames and a high firebed temperature. The high firebed temperature may be due to the less reactive type of coke produced from good coking coals, which thus burn to CO_2 and form little CO in the fuel bed. These hot coals, if used with a

low ash content, may damage the links or firebars. The character of the fixed carbon, which varies considerably between the groups, has much to do with the temperature generated in the firebed with a given air supply.

Coking Coals. The essential property of a coking coal is that it will produce a strong hard coke suitable for metallurgical purposes ; this is best obtained from coals which combine a high caking power with a relatively low percentage of volatile matter. The coals which stand out pre-eminently are the coking coals of Durham and South Wales (carbon about 88 per cent. and volatile about 23–30 per cent. on the dry ash-free coal). Reserves of these coals, however, are not large, but it has been proved possible to obtain satisfactory cokes by using blends of coals which hitherto had been considered quite unsuited for coke production. The use of narrower coke ovens has also assisted in widening the choice of coal. Thus coals used to produce coke now include a number of coals formerly used only for gas manufacture or for household purposes. The term " coking coal " is therefore becoming more catholic in its application.

For metallurgical purposes the coke must be low in sulphur and phosphorus, and this restriction removes a number of otherwise suitable coals from consideration.

Gas-Making Coals. For gas making in horizontal retorts it is desirable that the coal used shall form a good coke and at the same time have a high volatile content (Groups 3 and 4) to ensure a high yield of gas. With the advent of continuous vertical retorts and the steaming of the charge, however, the range of coals which can be successfully carbonised has widened in the direction of lower caking power (Group 2), particularly when sized coals can be used. The term " gas coal " has not, therefore, the same significance as formerly.

Manufacturing Coals for other than Steam Raising Purposes. There are many types of furnace and kiln used in industry, and the design of the installation usually decides which type of coal is the most suitable. Generally a coal of moderately high volatile matter (Groups 1 and 2—Table 1) is required for furnaces of a reverberatory type, but where a high local temperature is required, lower-volatile coals such as Group 3 are more suited, but these need more draught. For producer gas manufacture the best fuels are the non-caking or weakly-caking coals of Groups 1 and 2. The fusion temperature of the ash should be reasonably high unless the producer is of the slagging type.

Where the flue gases from burning coal may come into contact with foodstuffs, attention must be paid to the amount of certain elements, such as arsenic, which may, more rarely, be present in the coal in quantities high enough to be technically significant.

THE CHARACTERISTICS OF COAL OF INDUSTRIAL IMPORTANCE

There has been given above a brief description of the properties of the various types of coal available. It is now necessary to enter into more detail of the evaluation of a coal for industrial purposes. For most practical purposes it is sufficient to have a knowledge of four characteristics :—

(1) The quantity of inherent moisture.
(2) The quantity of volatile matter.
(3) The caking properties of the coal.
(4) The quantity of ash, and its fusion temperature.

The Inherent Moisture. The quantity of inherent moisture present in a coal affords considerable information upon the general character of a coal as will be seen from Tables 1 and 2. Coals of high rank, i.e. of high carbon content and low volatile matter, have little inherent moisture (0·6 to 2 per cent.), and, roughly speaking, as the volatile matter increases, so does the inherent moisture. Good caking coals may have inherent moisture contents of from 1 to 4 per cent.

At about 5 to 7 per cent. of inherent moisture the coals are feebly caking, and the non-caking bituminous coals contain some 10–16 per cent. of inherent moisture.

The Volatile Matter. The volatile matter based on dry ash-free coal affords confirmatory evidence of the type of coal, particularly in the low-volatile range below 20 per cent. Between 20 and 30 per cent. volatile matter, good coking coals are obtained. Above 30 per cent. the value of the volatile matter is less precise in fixing the caking properties of the coal.

Roughly speaking the length of the flame obtained during combustion varies as the volatile matter. Coals with high percentages of volatile matter (Groups 1 and 2) burn with long flames (long-flaming coals), and low-volatile coals with short flames (short-flaming coals)—Groups 3 and 4.

A reverberatory type of furnace or some types of kiln require a long-flaming coal since it is desirable to obtain heat away from the grate. For intense local heating, a short-flaming coal must be employed. In boiler practice coals high in volatile matter are useful where sudden increases in load are anticipated.

Closely connected with volatile matter is the rate of combustion of a coal. Low-volatile coals are slow burning, and high-volatile coals are fast burning. The high-volatile coal evolves a considerable proportion of its weight as gas, tar and water, and the resulting gas and tar can be burned above the grate more rapidly than a corresponding weight of solid fuel on the grate, although special attention must be paid to the combustion of the gas and tar in order to avoid smoke and to obtain complete combustion.

The residue left on heating a non-caking coal is usually more reactive than the coke yielded by a caking coal. That is to say, it will combine more readily with oxygen to give carbon dioxide, or react with carbon dioxide to give carbon monoxide. A caking coal and a non-caking coal show, therefore, very different behaviour on a grate. The coke from the caking coal burns more slowly than the residue from non-caking coal and gives a higher fuel bed temperature. Moreover, a caking slack will form larger pieces of coke having a much smaller external surface area per unit volume, and thus burning more slowly.

The volatile matter has also an important effect on ignitability since it is the volatile matter which ignites first. Distillation of volatile matter precedes combustion of the fuel. Hence low-volatile coals, for example anthracites, are difficult to ignite whereas high-volatile coals ignite readily.

The Caking of Coal. The caking properties and swelling power constitute probably the most important characteristic of coals.

The reasons leading to the variations in the caking properties of coals are not fully understood. Theories have been advanced to correlate the caking of coals with the amount of extract which the coals yield when treated with solvents such as pyridine or with benzene under pressure. Although it is true that strongly-caking coals yield a larger proportion of certain of these extracts than non-caking coals, it has become clear that the non-extractable portion of the coal also plays an important part in the caking process. The theories connecting amount of extract with caking power do not completely explain the marked improvement in the coke arising from carrying out the coking under increased pressure.

The mechanism of coke formation depends on the facts that (1) caking coals soften to greater or less extent and become plastic during a certain temperature range, (2) fluid material is extruded on to the surface of the particles and grains of coal and (3) a pressure is set up in the coking mass due to gas which forces the fluid materials to cover the surfaces of the pieces of coal where they act as a cement.

It is found that when a caking coal, crushed into a mass of smaller pieces (e.g. slack) is so heated that it is raised in temperature at a constant rate

of some 1° C. to 3° C. per minute, the sequence of events is as follows (see Fig. 6) :—

No visible change occurs up to about 300° C. (512° F.), but at about this temperature the evolution of considerable amounts of oxides of carbon and water vapour shows that the coal is beginning to be decomposed. If a stream of inert gas be passed through the heated coal it is found that at about 370° C. (698° F.) the coal begins to offer a decided resistance to the passage of the gas. This temperature depends on the rank of the coal and is higher as the volatile content of the coal decreases ; it may be over 400° C. with a low-volatile bituminous coking coal. Individual particles of coal begin to swell and careful examination reveals that certain constituents of the coal have begun to fuse. Over this temperature range, in addition, small quantities of oils are evolved from the coal.

From about 370° C. (698° F.) upwards, the resistance of the coal to the

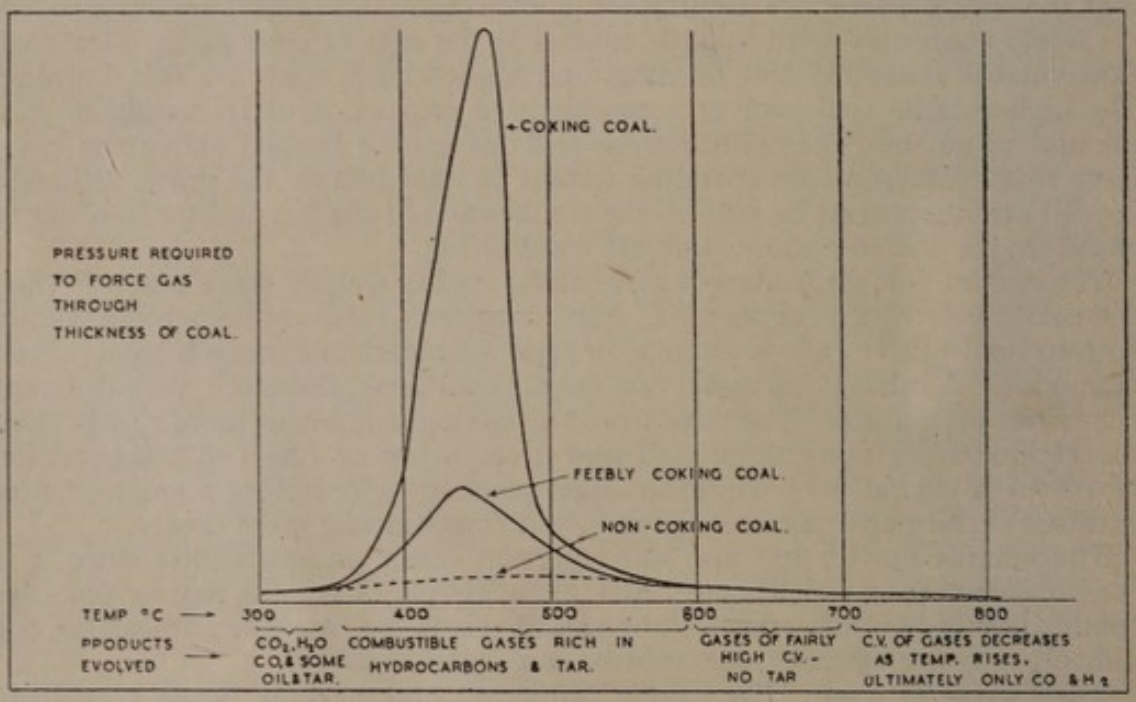

FIG. 6. The behaviour of coal on heating.

passage of gas increases very rapidly, and in a good caking coal, if the coal was originally in the form of slack, it is found that at about 430° C. (806° F.) a very considerable pressure is needed to force gas through it. During this period a certain amount of gas and tar continue to be evolved. The coking mass is in the pasty condition which is known as the " plastic state." At a somewhat higher temperature—frequently between 440° C. and 450° C. (824°–842° F.)—the resistance of the coal to the passage of gas rapidly decreases to a very low value. At this stage the plastic mass has hardened into semi-coke. Semi-coke is not plastic and there is now little further resistance. Quantities of gas and tar which have been imprisoned in the plastic mass are now released. The presence of this gas in the plastic coal has previously caused a high internal pressure, and the quality of the coke produced from the coal is dependent to a considerable degree upon this pressure. As before, the temperatures mentioned depend on the volatile content of the coal, being with a low-volatile coal higher than those here mentioned.

Tar continues to be evolved up to a temperature of about 600° C. (1,112° F.) accompanied by considerable quantities of gas rich in hydrocarbons and of a

high calorific value. It is during this stage that most of the benzole-forming constituents are evolved from the coal ; the benzole is not formed as such, but is the product of the secondary decomposition, at a higher temperature, of the primary bodies formed in carbonisation.

As the temperature increases to 700°–900° C. (1,292°–1,652° F.) the quantity of gas produced decreases, and this gas is of lower calorific value. The gas now consists mainly of carbon monoxide and hydrogen.

For coals with a lower percentage of volatile matter, the temperature range of plasticity increases. Thus, for a coal with 30–35 per cent. volatile matter, it may be 370°–450° C. (698°–842° F.), whereas for a coal with 22 per cent. volatile matter, the plastic range may be 420°–530° C. (788°–986° F.). As the coals used become less strongly caking the resistance to the passage of gas during this temperature range becomes less until, with a non-caking coal, there is, of course, no increased resistance since these coals do not pass through a plastic phase.

In practice, the caking and swelling properties of coal may be greatly dependent on the size of the coal, on its percentage of ash, on the rate of burning or gasification, on the method of firing and on the fuel bed thickness. These properties may also be affected by weathering or by heating in store. These will be briefly discussed.

(*a*) *The Size of Coal.* The phenomena of caking and swelling are greatly modified by the size of the individual pieces of the coal. Whereas a small piece of caking coal of the order of $\frac{1}{8}$ inch diameter will tend to swell and become plastic as a whole upon heating through the plastic range (370°–450° C.) (698°–842° F.), and will, therefore, easily cohere to the neighbouring particles, a larger piece of coal, say of the order of $\frac{1}{2}$ inch and over, will not swell in this way, but fluid matter will be forced out through the sides along the bedding planes to form blisters of fused material on the surface. With this behaviour there is less opportunity for the individual pieces of coal to become fused together in such a way as to form massive lumps of coke such as are formed when slack is used. These lumps present such a small relative surface that it is difficult to get effective contact between air and coal. It is thus often found that whereas a coal, by reason of its caking properties, will give considerable difficulty in a furnace when charged in the form of slack, it can readily be used when in the form of nuts or doubles. It will also be found that whereas a slack of medium caking properties (Group 3) may be too caking for the purpose for which it is used, a coal in the form of singles or nuts from the more strongly caking Group 4 may give satisfaction on account of its size, and in spite of its stronger inherent caking properties. Generally the slack from a coal is more strongly caking than the larger material, even when the larger material has been crushed to the same size as the slack.

(*b*) *The Amount of Ash.* The amount of ash has a material influence on caking properties. Under certain conditions a medium or fairly strongly-caking slack, containing 5 per cent. of ash, may be more difficult to burn on account of strong coke formation than the slack from the same coal containing 15 per cent. of ash. This reduction of caking power by non-caking matter has suggested the blending of coke breeze and anthracite duff with caking coals to modify their burning characteristics. At least 10 per cent. anthracite duff can be used in this way when the caking index of the basic coal (its " agglutinating value ") is not lower than 15 when determined by the British Standard method, corresponding generally to group 4 coals (Table 1).

(*c*) *The Rate of Burning or Gasification.* Experience with chain-grate stokers has shown that a high rate of burning per square foot of grate area reduces the caking properties of coals, and the formation of large pieces of coke, and leads to better combustion conditions.

(*d*) *The Method of Firing, and the Fuel Bed Thickness.* A free-burning or medium-caking slack, burned in a thin bed, may give unsatisfactory results through an excessive amount of fines falling through the grate, and through excessive grit emission. The same coal burned on a stoker of the retort or coking type using a thick fuel bed will give more satisfactory results in this respect, because the type of heating tends to increase the caking properties and the fines are thus trapped in the fuel bed.

On the other hand a strongly-caking slack will form massive lumps of coke on the coking type of stoker and be difficult to burn, whereas, spread out in a thin layer, the fluid plastic material will be burned away before masses of coke can be formed.

SULPHUR IN COAL

Sulphur is present in coal in general to the extent of 0·5 to 2·5 per cent., although occasionally coals are met with containing much larger amounts. It is present in the original coal in three forms, namely as pyrites (FeS_2), as organic sulphur compounds and as sulphates.

Under the action of heat the pyritic sulphur is partly evolved in the form of free sulphur according to the reaction $FeS_2 = FeS + S$; this reaction occurs at about 500° C. (932° F.). The remaining sulphur left in the form of FeS may behave in any or all of three ways according to the conditions. In the presence of ample supplies of oxygen it may be oxidised according to the reaction $4FeS + 7O_2 = 2Fe_2O_3 + 4SO_2$. The ferrous sulphide may in itself combine with some of the other constituents of the ash forming ferrous silicate with the liberation of sulphur ; this reaction assists the formation of clinker, as ferrous silicates have a low melting point. Finally, a proportion of the sulphur first liberated may combine with the carbon, forming organic sulphur compounds.

The organic sulphur compounds are burned to SO_2 in the course of combustion.

At high temperatures sulphur present as sulphate is partly removed from the coals in the form of sulphur trioxide. Sulphur is thus objectionable for the following reasons :—

(1) Oxides of sulphur contaminate the atmosphere, forming sulphurous and sulphuric acids which cause serious damage to property.
(2) Part of the sulphur may be absorbed by inter-action with the grate bars, and since sulphide of iron is comparatively fusible, it may give rise to serious trouble.
(3) When the furnace is used for metallurgical purposes the sulphur may pass into the metal under treatment.
(4) When a fuel containing hydrogen is burned, it forms water vapour. If the temperature of the gases falls too low, for example on the surface of the economiser, some of the water is condensed, and the oxides of sulphur are absorbed by this water, forming a solution of sulphurous and sulphuric acids. This may be the cause of severe external corrosion of the metal. The presence of SO_3 in flue gases has been shown to raise the dew-point of the gases materially and thus to increase the danger of deposition of acids on the metal.

Iron pyrites is a contributory cause of clinker formation, but can be removed to a great extent by appropriate cleaning methods.

COAL PREPARATION AND CLEANING

The coal as mined, termed "run-of-mine," comprises a mixture of sizes, from the finest particles of slack less than $\frac{1}{32}$ inch diameter up to very large lumps of 2 or 3 feet. It also includes dirt unavoidably collected from the roof

or floor in the process of mining, or occurring as thin dirt bands in the coal itself. The shale may vary from almost pure stone to a carbonaceous material containing 60 per cent. or more of combustible matter.

The material is first screened in order to separate the large coal. The large material may be over 3 inches or 4 inches in size according to local practice. The large coal is passed over picking belts where the dirt is removed by hand. The purity of this coal, which is used principally for domestic purposes and locomotives, depends upon the efficiency of the hand picking.

The small coal is generally washed and sold for industrial purposes. In 1927, 20 per cent. of the coal was cleaned in 500 washeries and 27 dry cleaning plants. In 1934, 40 per cent. of the coal was cleaned in 611 washeries and 150 dry cleaning plants. In 1938, the proportion of coal cleaned had risen to 45 per cent. of the total output. Since much of the large coal is hand-picked, the proportion of the coal that is cleaned is a good deal higher than these figures indicate.

The operation of coal cleaning processes depends on the relationship between

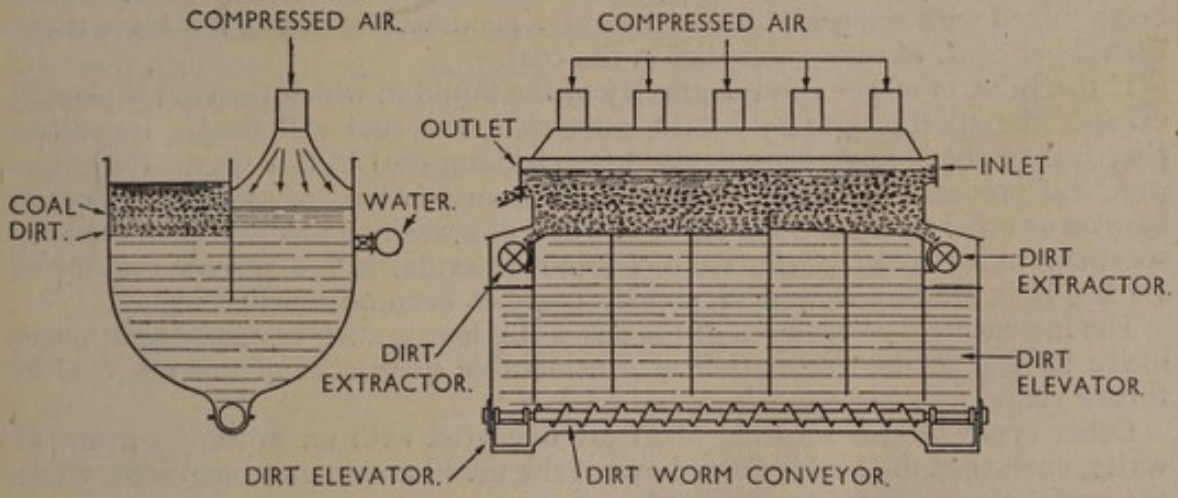

FIG. 7. The Baum coal washer.

the velocity of fall of the particle in water and its density. Expressed mathematically :—

$$V = C\sqrt{d(s\text{-}w)}$$

V = velocity of fall in water, ft./sec.
d = diameter of particle (inches).
$(s\text{-}w)$ = diff. in sp. gr. of material and washing medium (water = 1).
C = constant, about 1·8 for nodular pieces.

The specific gravity, s, of the pure coal is between 1·25 and 1·4, of the shale 2·5, and of pyrites, 5. Thus, for a constant diameter of particle, i.e. a constant value of d, pyrites and shale will fall to the bottom of a tank of water more rapidly than pure coal. If the water be given an up-and-down motion, the upward velocity of the water can be made sufficiently high to cause the coal then to travel upward while the shale and pyrites continually travel downward. Fig. 7, showing a Baum washer, gives an illustration of a practical form of washing plant. A pulsating motion set up by compressed air causes the water to move up and down in the coal bed with the necessary velocity, and thus keeps the dirt at the bottom of the bed and the coal at the top. A fast-flowing stream of water carries the raw material into the wash-box, and the clean coal is carried out of the box, over a suitable gate to the drainage screens. A gate at the outlet end of the wash-box is lifted periodically by the washery attendant to remove the dirt, or can be operated automatically. The pulsating

motion was formerly applied by a plunger acting in the water in the plunger box. This arrangement is still found on older plants.

The velocity of fall, however, also depends upon the size of the particle, d. In the form of purely relative figures it can be calculated that :—

Where d =	$\frac{1}{120}''$	$\frac{1}{64}''$	$\frac{1}{32}''$	$\frac{1}{8}''$	$\frac{1}{4}''$	$\frac{1}{2}''$	$1''$	$2''$
V =	0·09	0·12	0·18	0·35	0·5	0·71	1	1·4

The shape of the particle has also an important bearing on the rate of fall, a flat piece of shale will fall with different motion and generally with a faster velocity than a rounded piece of the same gravity.

In some types of washing plant it is necessary to screen the coal into a number of sizes before washing. In others, however, in order to reduce the amount of plant necessary, to diminish the labour costs and to simplify the procedure as much as possible, normal practice is to treat the unscreened coal in one operation and, after washing, to screen out the slack below about $\frac{1}{8}$ inch. This fine material may be re-washed.

Under these conditions it is impossible to avoid a small proportion of dirt being mixed with the coal, but, in general, a good washer should not leave more than 2 per cent. of removable dirt in the coal.

If the value of w, the specific gravity of the liquid in which the coal is placed, exceeds the specific gravity of the pure coal, the coal will float. Ingenious practical methods have been devised for cleaning coal in this way. Recently, practical processes have been evolved using suspensions of sand, clay (loess), barytes or other finely divided solids in water as a washing medium of the correct specific gravity. Also a solution of calcium chloride, with a specific gravity of 1·4 has been employed, and very clean coal has been obtained.

The increasing development of processes employing such suspensions appears likely to lead to the production of considerable quantities of low-ash coal in future years.

Other types of coal washing plant are operated with an upward current of water, such that the coal is floated out of the main stream and recovered, while the impurities sink and are rejected.

Another highly important type of plant is that of dry cleaning. In this the material does not come into contact with water so that the ash removed is not, even in part, replaced by water mechanically held. Dry cleaning plants utilise an upward current of air introduced beneath a pulsating table. Raised parallel riffles cause the clean coal, the high-ash coal, and the dirt each to flow to different parts of the table edge where they are recovered separately.

In coal washing the fines present a problem of no little difficulty. In dry cleaning methods they are given off from the coal as a cloud of dust in the air current from which they have to be separated by mechanical devices. In wet washing they form a slurry with the water that is very difficult to settle. Methods have now, however, been devised whereby more rapid settlement can be effected. The settled slurry is high in ash. The formation of slurry can be prevented to a considerable extent by dedusting the coal before wet cleaning.

The distribution of dirt in the raw coal coming from the pits may be very variable according to the friability of the shale. Where the pieces of dirt are separate from the coal there is no particular difficulty in separation, but it often happens that where there are dirt partings in the seam, small pieces adhere to the coal, and the combined specific gravity is sufficiently low for the particle to be classified as coal. A certain amount of ash inevitably arises from this cause, and this forms in the coal washing process an intermediate product known as " middlings." The intermediate product may also be due to a close inter-mixture of the dirt with the coal so that separation of dirt and coal cannot be effected unless the particles are crushed to a fine powder. Finally,

in some instances, the coal and dirt are so intimately associated that separation of the two is not possible by any known means.

It will be recognised, therefore, that the operation of coal washing must be carried out in a skilled manner. Excess ash in coal may be due to practical difficulties of operation at the washery or to the output of the colliery under war conditions being so great that the existing washery is over-loaded. Some collieries are not equipped with coal cleaning plant.

THE SIZE OF COALS

The smaller coals below 3–4 inches, separated from the first screening, are generally (although not invariably) washed. They may be sold without further treatment, but usually are screened into sizes for industrial uses. They may be divided into two broad classes :—

(1) Sized or graded coal.
(2) Slacks.

Sized Coals. Sized coals are those which have been screened between two sizes and from which the greater part of the fines has been removed. They are known by such names as pearls, peas, beans, singles, nuts, etc., but these names are used unsystematically, the beans of one colliery often being equivalent in size to the singles or nuts of another colliery. The real criterion of size should be a size analysis test where a sample is taken and passed through B.S.I. standard square mesh sieves. Such tests show the results given in Table 4 for sized industrial coals :—

TABLE 4. GRADING OF COMMERCIAL SIZES OF COALS

	¼–0 inch per cent.	½–¼ inch per cent.	1–½ inch per cent.	1½–1 inch per cent.
Pearls	40	55	5	—
	30	64	6	—
Peas ..	8	78	14	—
	10	65	25	—
Beans	8	53	39	—
	6	41	53	—
Singles	6	30	60	4
	4	18	72	6

There is, however, no sharp line of demarcation between the groups. In view of the importance now attached to the influence of size on combustion and gasification rates and efficiency, it is desirable that, in future, precise information of the above nature should be used rather than that reliance should be placed on trade names which may be misleading.

Slacks. In general, slack may be defined as that material which has been screened through one upper screen and contains, with certain exceptions to be described later, all the fine coal.

The upper size may be as big as 2 inches ; the slack may then be almost as useful as a graded fuel ; or the upper size may be as low as ⅜ inch when the slack is often difficult to handle if wet.

Slacks are known in various districts by different names, such as dross, smalls, duff, gum, etc. These names do not guarantee any specific size, and, as stated above, a size analysis should be made. According to the size of the

aperture in the upper screen, the size grading of slacks should be approximately as in Table 5.

TABLE 5. SIZE GRADING OF SLACKS

	⅛–0 inch per cent.	¼–⅛ inch per cent.	½–¼ inch per cent.	1–½ inch per cent.	Over 1 inch per cent.
2 inch slacks ..	20	14	20	24	22
1½ inch	22	16	23	27	12
1¼ inch	26	20	26	25	3
1 inch	30	21	30	19	—
¾ inch	41	29	27	3	—
½ inch	45	39	16	—	—
⅜ inch	52	40	8	—	—
South Wales ..	52	15	16	12	5
do. ..	35	14	15	18	13

The size distribution of South Wales slacks may differ from that of slacks in the rest of Great Britain because, whereas it is customary in the rest of Great Britain to size coals through screens perforated by round, square or other shaped holes, in South Wales it is often graded over parallel bars set 1⅛ inches apart, which pass all the fines and also much larger pieces. This gives an entirely different size distribution.

The Size of Coal : General. Partly owing to the failure to use size analysis there is a lack of reliable information about the relationship between size and combustion and gasification efficiencies, and draught requirements.

The size of coal should be such that the lumps present enough surface for good combustion but are not so small as to prevent easy penetration of air. Uniform grading (or sizing) is a valuable asset in that it produces a fuel bed which offers even resistance to the passage of air through it, and therefore will result in uniform combustion or gasification.

It must not be forgotten, however, that the size of particle of the fuel when it is carbonised or partly carbonised may be altered by its caking properties, causing particles to cohere.

Sized coal avoids segregation. In a slack which consists of relatively large particles together with fine dust, the larger coal frequently becomes concentrated in one zone of the hopper or heap, with the result that the mechanical stoker may at one time be fired with nutty material and at another time with little more than dust, or, as frequently occurs on chain-grate stokers, the sides of the stoker may receive all the nutty material, and the centre the dust. The result is an uneven fuel bed resistance and indifferent combustion conditions. Care should be taken to mix the coal thoroughly when waggons of dry slack are unloaded since it is usual for the centre of the waggons to contain a greater proportion of fines than the sides or ends. If the centre of the waggon is discharged first, then for several hours the combustion appliance will be using fuel with a high percentage of fines. Later on it will receive much coarser material drawn from the ends of the waggon.

Many of these, and other, difficulties would be avoided if all the fines were removed from coal and used in powdered fuel installations, and only " sized " coal were marketed for general industrial purposes.

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CHAPTER II

INDUSTRIAL FUELS: FUELS DERIVED FROM COAL

Carbonisation—Coke—Coal gas (Town gas)—Producer gas—Blast furnace gas—Water gas—Coal tar fuel oils—Electricity.

FUELS derived from coal comprise coke, coal gas, producer gas, water gas, blast furnace gas, and oil fuels derived from the distillation of the tar produced in coal carbonisation. Electricity may also be regarded as falling in this category. Many of these fuels are obtained by the carbonisation of coal. Large quantities of mixed gas (coke oven gas and blast furnace gas) are used in the heavy industries, where both of these gases are by-products.

All coals do not when carbonised produce coke as a commercial product, a fact which has been made clear in the preceding chapter. For industrial carbonisation therefore, it is at present necessary to confine ourselves to certain classes of coal. Some coals are highly coking and the slack from these is carbonised in coke ovens for the production of hard metallurgical coke. These coals generally yield from 35 per cent. to 22 per cent. of volatile matter as carbonised in practice, determined on dry coal containing about 5 per cent. of ash. Coals towards the higher-volatile end of this group and caking coals of still higher-volatile content generally produce a high gas yield and a slightly weaker coke and are used in the gas industry.

Coking coals are limited in quantity and within a comparatively short time this country will face a real shortage of first quality coking coals. It is of the utmost importance, therefore, that the best use should be made of the coal that we possess, and that wherever possible the best coking coals should be reserved for metallurgical purposes. Research is in progress which it is hoped may lead to a commercial method of manufacturing town gas of the present quality from any coal irrespective of its coking qualities, but this process even if fully successful is not likely to become commercially available for some years.

In coal carbonisation the coal is heated in a closed chamber from which air is excluded, the heat being transmitted to the coal through walls constructed of fire brick or silica brick. A portion of the gas or coke obtained in carbonisation is used to produce the necessary heat.

The gases are led from the carbonising chamber through mains and are first cooled in order to remove as much as possible of the tar and water vapour evolved from the coal; the remaining tar is removed from the gases by a mechanical process, the gases being then washed by water or sulphuric acid to remove ammonia from which sulphate of ammonia is manufactured. Benzole is extracted from the gases by washing with oil or by other methods, the oil being subsequently distilled to recover the benzole which is then purified for use. The residual gases comprise coal gas which according to its origin and use is variously styled coke oven gas or town gas. There is no essential difference between these gases. Where the gas is to be used for town purposes and also for many industrial purposes it is purified from hydrogen sulphide by being passed over iron oxide.

The coke which remains in the retort at a temperature of perhaps 1,000° C. is discharged, cooled, e.g. by being sprayed with water, by heat interchange with the incoming secondary air, or by means of cold flue gas free from oxygen (dry-cooling), and is then sent to the coke preparation plant where it is graded into sizes. A proportion of it is generally broken and re-sized for special purposes.

COKE

The yield of coke from coal in practice varies from 60 per cent. to 80 per cent. of the weight of the coal charged into the retort. It thus represents both by weight and thermal value the principal product of coal carbonisation.

Coke consists essentially of carbon and mineral matter accompanied by some 1 to 2 per cent. of sulphur and small quantities of hydrogen, nitrogen and phosphorus. When marketed, gas coke should contain less than 6 per cent. of water and metallurgical coke less than half that quantity. It is possible to calculate the calorific value of coke within sufficiently close limits for most industrial purposes from a knowledge of the ash and moisture contents and on the assumption that the remainder of the coke consists of carbon mostly in a graphitic form. Anticipating the information to be given in Chapter V the calorific value of coke in B.Th.U per lb. can be calculated from the expression :—

(carbon per cent. — ash per cent. — water per cent.) × 14,200 ÷ 100
for oven coke

(carbon per cent. — ash per cent. — water per cent.) × 14,400 ÷ 100
for gas coke.

The composition of a typical dry ash-free coke is :—

Carbon	95 per cent.
Hydrogen	1 ,,
Nitrogen and sulphur	2 ,,
Oxygen (and errors)	2 ,,

Coke possesses very little volatile matter, this being rarely over 4 per cent. and generally from 2 to 3 per cent. as derived from gas works. Many gas cokes and most metallurgical cokes contain less than 2 per cent. of volatile matter. This volatile matter is evolved almost wholly as carbon monoxide and hydrogen.

An important property of the coke is its resistance to handling, a property often termed its " hardness." By the nature of its method of manufacture and from its subsequent quenching with water, coke is strained mechanically and may contain a number of small cracks. Upon handling or while in use these cracks may cause the coke to break and to form smalls. The diminution in the size of coke is deleterious for many purposes though not for others. The test most commonly used for measuring the strength of coke is the shatter test. In this test 50 lb. of lump coke over 2 inches in size is dropped four times from a height of 6 feet on to a cast iron or steel plate and the amount of breakage is measured by screening the coke. Table 6 gives typical figures :—

TABLE 6

	Gas retort coke	Metallurgical coke			Low temperature coke
		South Wales	Yorkshire	Derby	
Shatter test :					
Over 2 inch, per cent.	76	87·5	78	57	73
,, 1½ inch ,,	88	95	88	72	76
Through ½ in. ,,	2	1	2	3	14

A good metallurgical coke should show 75 per cent. over 2 inches, 85 per cent. over 1½ inches, and less than 3 per cent. below ½ inch size. This test has been found by experience to be of great value in selecting cokes for specific purposes. A hard coke, for example, is a good hot melting coke.

A metallurgical coke is dense whereas most gas cokes are more porous. A

high porosity decreases the crushing strength of the coke and increases the rate of reaction with oxygen to some extent. Where a very high temperature is desired a dense hard coke burned with mechanical draught will give the best results. Where a less high temperature is needed an open, porous and softer coke will prove quite satisfactory and can be burned with less draught.

The size of cokes has an important bearing on their combustion properties. On small appliances particularly, a reduction in size may compensate for low combustibility. The size of coke should be selected to suit the appliance in

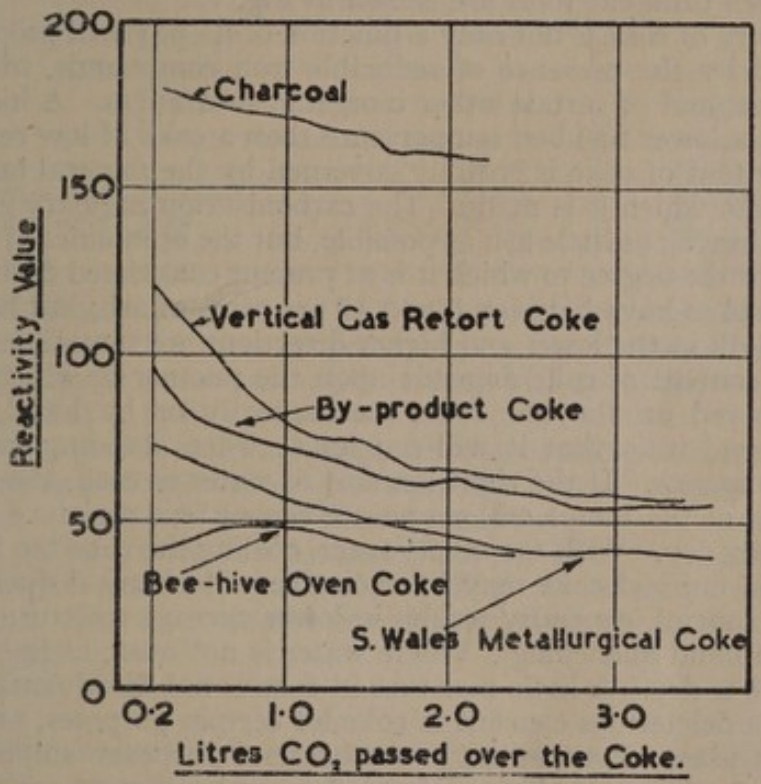

FIG. 7A. Curves showing the reactivity of several cokes and charcoal, determined by passing carbon dioxide over the material heated to 950° C. (Fuel Research Station). The reactivity as measured by the conversion of the carbon dioxide to carbon monoxide will be seen to decrease as the material is heated for a longer time and more carbon dioxide is passed over it, until it reaches a virtually constant value.

which it is used. The standard sizes (square mesh screens) for gas coke are as follows :—

No. 1. No upper limit over $1\frac{3}{4}$ inches.
,, 1a. Within the limits $3\frac{1}{2}$ inches to $1\frac{3}{4}$ inches.
,, 2. Within the limits 2 inches to 1 inch.
,, 3. Within the limits $1\frac{1}{4}$ inches to $\frac{1}{2}$ inch.
,, 4. Within the limits $\frac{3}{4}$ inch to $\frac{3}{8}$ inch.
,, 5. Unwashed fuel approximately $\frac{3}{4}$ inch to $\frac{1}{4}$ inch from which the fines up to $\frac{1}{8}$ inch have been removed.
Breeze. Generally $\frac{1}{2}$ inch to 0.

The local coke associations (or the gas companies) will advise on which size of coke should be selected for a particular purpose.

The temperature at which coke ignites depends on the temperature to which it has been subjected during the process of manufacture, and upon the conditions in the retort. Some cokes will ignite at temperatures as low as 450° C. and according to the conditions under which they have been made the temperature will increase until a hard coke made by the old beehive process that is used for foundry work may not ignite until well over 600° C. These tem-

peratures are somewhat arbitrary, however, since they depend to some extent upon the experimental conditions under which the determination was made.

Reactivity or combustibility is another important property of coke. The stronger the coking power of the coal from which the coke is produced and the higher the temperature at which it has been carbonised the less easily will the coke react with oxygen or carbon dioxide. Reactivity is generally measured by the rate at which a coke will convert carbon dioxide into carbon monoxide at a given temperature and under standard conditions. Differences in this respect between different fuels are shown in Fig. 7A.

The reactivity of coke is not only a function of its physical properties, but is also increased by the presence of reducible iron compounds, of sodium carbonate, of lime, and of certain other inorganic substances. A highly reactive coke will give a lower fuel bed temperature than a coke of low reactivity.

The ash content of coke is entirely governed by the mineral matter content of the coal from which it is made. The carbonisation industry endeavours to produce coke having as little ash as possible, but the economics of coal cleaning place a limit to the degree to which it is at present considered desirable to clean coals. Most cokes have between 5 and 12 per cent. of ash, but figures outside these limits both in the lower and higher directions are sometimes found.

The water content of coke depends upon the manner in which it is cooled. Water is sprayed on the hot coke, mechanically or by hand, in quantity sufficient to cool it so that it will not ignite when standing in air or when loaded into waggons. If the right amount of water is used, the residual heat in the interior of the lumps will evaporate the surface moisture and the coke will be virtually dry. With too much water, cooling the coke too far, the water content of the finished coke may be excessive. Mechanical quenching, being under better control, generally results in lower average moisture content over a period than hand quenching. Where water is not used, as in " dry quenching," the coke is dry. A little moisture in coke is not disadvantageous.

Sulphur is a deleterious element in coke for certain purposes, particularly for metallurgical uses. For ordinary combustion purposes sulphur has little influence unless the gases derived from the combustion of coke or from the producer gas made from coke are to come in contact with goods in a furnace which would be detrimentally affected by sulphur dioxide. Any sulphur present in the coal as iron pyrites (FeS_2) is first converted into iron sulphide (FeS) at a temperature of some 500° C., and at higher temperatures part of the ferrous sulphide will react with hydrogen forming some metallic iron and liberating hydrogen sulphide. Most of the sulphur present in the form of FeS, however, remains in that form and thus appears in the coke. Sulphur present as sulphates may be distilled from the coal, though generally the amount of sulphates in a coking coal is very small indeed. Sulphur present in the coal as organic sulphur is partly retained in the coke and is partly evolved in the form of carbon disulphide and other organic sulphur compounds which are to be found in coal gas. Similarly a part at least of the sulphur distilled from pyrites reacts with the coke to form carbon disulphide (CS_2), while part is now known to interact with the coal to form organic sulphur that is retained in the coke. Generally the sulphur content of cokes ranges from 0·75 to 1·5 per cent.

The quantity of phosphorus present in cokes varies between 0·002 per cent. and 0·025 per cent. in this country, but some may contain as much as 0·3 to 0·5 per cent. Phosphorus is only important when the coke is required for metallurgical uses.

Coke has a true specific gravity of 1·75 to 2, this being the specific gravity of the material without pores. The " apparent " specific gravity, i.e. the specific gravity of the lump of coke with the pores varies from 0·75 to 1. The porosity is generally between 40 and 60 per cent.

Table 7 gives some general information upon the bulk density of coal and coke broken for use and having a normal content of moisture and ash.

TABLE 7

Fuel	Bulk density Pounds per cubic foot	
Anthracite	55–60	
Bituminous coal	50–55	
Steam coal	50–55	
	¾–1½ inch	Rough
Coke oven coke	26–30	29–32
Horizontal retort coke	23–26	25–28
Vertical retort coke	21–22	23

Evaporative Value of Coke. Tests which have been carried out to determine the net evaporative value of coke prove that 1 lb. of coke of average good quality has in a boiler a net evaporative value of 9 to 10 lb. of water per lb. of coke, from and at 212° F.

Combustion Rate. In ordinary circumstances combustion rates of 12 lb. to 16 lb. per square foot of grate per hour can be maintained with natural draught. With a suitable forced draught furnace a good average rate of combustion is 25 lb. per square foot of grate per hour, although it is sometimes possible to increase this to as high as 35 lb. to 40 lb. per square foot of grate per hour. The best results are usually obtained under conditions of balanced draught.

COAL GAS

The composition and calorific value of coal gas depends upon the method by which it is made. In coke ovens and horizontal gas retorts the gas produced is of high calorific value and consists of the coal gas without any admixture. Removal of benzole and purification may bring the gas down to the calorific value at which it is supplied to the public without further treatment. Sometimes, however, producer gas or water gas is added to it in order to reduce the calorific value.

The calorific value of town gas is controlled by Act of Parliament. Through the Ministry of Fuel and Power, gas companies make a statutory declaration of the calorific value in B.Th.U. per cubic foot gross at which they will supply gas. Tests are made independently of the gas companies to ascertain that at no time does the calorific value fall below the declared standard. Coke oven gas is supplied to domestic and industrial purchasers through the gas undertakings according to the powers given to them by Act of Parliament.

In continuous vertical retorts a certain amount of steam is blown in from the bottom of the retort achieving the double objective of partly quenching the coke and of being itself converted into water gas through its passage up the retort by the reaction : $C + H_2O = CO + H_2$. Since water gas consists essentially of carbon monoxide and hydrogen, the different methods of carbonisation influence the percentage of these two gases in the ultimate gaseous mixture. The typical composition of a number of forms of coal gas is as in Table 8.

Town gas, whether derived from coke ovens or made from coal at a gas works, has the same general composition and characteristics. The gas companies supply gas of a declared calorific value which is generally between 400

TABLE 8. COMPOSITION OF VARIOUS TYPES OF COAL GAS

	Horizontal retorts	Continuous vertical retorts without steaming	Continuous vertical retorts steamed	Debenzolised coke oven gas
Oxygen, O_2	0·4	0·4	0·4	0·4
Carbon dioxide, CO_2 ..	2	3	4	2
Hydrocarbons, "C_mH_n" ..	3·6	3	2	2·6
Carbon monoxide, CO ..	8	9	18	7·4
Hydrogen, H_2	52	53·6	49·4	54·0
Methane, CH_4	30	25	20	28
Nitrogen, N_2	4	6	6·2	5·6
Calorific value— B.Th.U./cu. ft.				
Gross	560	540	475	525
Net	500	482	426	467
Sp. gr. (air = 1)	0·42	0·43	0·48	0·38

and 500 B.Th.U. per cubic foot. In 1940, of 654 gas undertakings in the country that had a declared calorific value, 219, including some of the biggest companies in the country, operated at 500 B.Th.U. per cubic feet; 119 between 475 and 495; and 223 between 450 and 470 B.Th.U. The consumer is charged by the therm, a therm being 100,000 B.Th.U. The number of cubic feet of gas comprising a therm is given by the fraction

$$\frac{100{,}000}{\text{declared C.V.}}$$

The consumer is thus charged on the therm basis for the actual gross amount of heat with which he is supplied. The declared value of a gas is always based on the gross C.V. of the gas (see Chapter V).

The calorific value of any mixture of gases is the sum of the calorific values of the constituents. The calorific value of town gas can thus be calculated by adding the calorific value of its content of H_2, CH_4, CO, C_mH_n, the oxygen, CO_2 and N_2 having no calorific value. The uncertainty that arises from making this calculation in town gas is that the methane generally contains a small proportion of ethane which has a higher C.V. than methane, and more particularly that the composition of the hydrocarbons designed C_mH_n depends on the method of manufacture. According to data contained in British Standard publications the several gross calorific values are as follows at 60° F. and 30 inch bar. saturated with water vapour :—

Carbon monoxide, CO	318	B.Th.U. per cubic foot
Hydrogen, H_2	320	,, ,, ,, ,,
Methane, CH_4	995	,, ,, ,, ,,
Ethane, C_2H_6	1,730	,, ,, ,, ,,
Ethylene, C_2H_4	1,560	,, ,, ,, ,,
C_mH_n (unstripped horizontal retort and coke oven gas)	2,365	,, ,, ,, ,,
C_mH_n (unstripped vertical retort gas)	3,040	,, ,, ,, ,,
C_mH_n (for benzole-stripped gas) ..	1,560	,, ,, ,, ,,

The cost of gas per gross therm as received is considerably greater than the cost for solid fuels on the same basis. There are, however, reasons why town gas is proving of great value in industry and why its use in industry is increasing (see Chapter XXXIV).

The comparative freedom from sulphur of town gas is a matter of con-

siderable importance in many heating operations for which it is used. This matter will be further discussed in Chapter XVIII. The crude gas produced in the retorts and after washing for ammonia and benzole may still contain some 500–700 grains of sulphur per 100 cubic feet, or even greater quantities than this, the sulphur being principally present in the form of hydrogen sulphide. By extraction of hydrogen sulphide by means of iron oxide, the total sulphur content of the gas is reduced to some 20–30 grains per 100 cubic feet. Even this small quantity, however, is sometimes deleterious and the present trend of practice in many large gas companies is to reduce the sulphur content of a gas to less than 10 grains per 100 cubic feet. To effect this further reduction special plant is necessary, this being either a modification of the benzole extraction process or involving a catalytic process.

Coal gas cannot be highly preheated on account of its hydrocarbon content. The higher hydrocarbons, C_mH_n, and to a lesser extent methane, decompose into carbon and hydrogen at a red heat. The air for combustion can, however, be preheated. A coal gas generally requires some $4\frac{1}{2}$ to 5 volumes of air per volume of gas so that if the air for combustion is preheated the fact that the gas is not preheated has comparatively little influence on the final temperature reached.

PRODUCER GAS

The method of manufacturing producer gas is discussed in a later chapter in this book. Here it is sufficient to state that it is manufactured from coal or coke by blowing a mixture of steam and air through a bed of incandescent fuel, generally not less than 3 feet in depth. The gases leaving the fuel bed consist of a mixture of carbon monoxide and hydrogen accompanied by all the nitrogen in the incoming air. When coal is used as the fuel, distillation of volatile matter in the upper portions of the fuel bed causes the producer gas to be accompanied by the tar and gas liberated from the coal. As the temperature of the gases at this point may be comparatively high a large part of the tar is retained in the vapour state, whilst the remainder is carried mechanically in the form of minute drops of " tar fog."

The characteristics of producer gas will therefore depend upon whether the gases are made from coke or from coal, and upon whether it is used in the hot crude state or whether it is cooled and cleaned. Typical compositions of cooled, clean gas are as in Table 9.

TABLE 9. COMPOSITION OF COOL, TAR-FREE PRODUCER GAS

	Bituminous coal		Coke	Anthracite
	Static producers	Mechanical producers		
Composition, per cent. by volume				
Carbon dioxide, CO_2	4–7	4–7	4–6	4–6
Carbon monoxide, CO ..	20–28	22–29	24–28	25–27
Hydrogen, H_2	10–13	10–16	10–12	14–18
Hydrocarbons, C_mH_n	0·3–0·7	0·3–0·7	*Nil*	About 0·4
Methane, CH_4	2·8–3·5	2·8–3·3	0·4–0·6	1·2–1·3
Nitrogen, N_2	52–57	52–57	52–55	50–53
C.V.—B.Th.U/cu. ft. at 60°F. and 30 inch bar.				
Gross	140–160	140–160	130–140	133–153
Net	134–152	134–152	125–134	126–148
Sp. gr., air = 1; (approx.) ..	0·9	0·9	0·9	0·9

If the gases are derived from bituminous coal and are not cooled or cleaned the additional quantities of tar contained in them may increase the calorific value by some 15 B.Th.U. per cubic foot gross. Gases derived from coke and anthracite contain no appreciable quantities of tar.

If the gases are not cooled they contain in addition to their heat of combustion sensible heat which adds to the heat developed in the furnace on combustion. The sensible heat of the gases between 60° F. and 1,470° F. (15°–800° C.) is some 17 B.Th.U. per cubic foot.

Producer gas is characterised by a large quantity of inerts ($CO_2 + N_2$) and consequently by a low calorific value. It requires, however, considerably less air for its combustion than town gas, approximately equal volumes of producer gas and air being required. Thus a theoretical mixture of gas and air required for combustion, containing no excess air, has for coal gas a gross calorific value of 94–95 B.Th.U. per cubic foot, whilst for producer gas the calorific value is between 65 and 72 B.Th.U. per cubic foot. By using the producer gas without allowing it to cool a good deal of this difference can be made up and of course the overall efficiency of gas production can be increased. If the air is also preheated producer gas can give substantially the same furnace temperature as coal gas.

Producer gas is generally not purified from sulphur for industrial heating purposes because of the cost of purifying so large a volume of gas. Most of the sulphur contained in the coal will be found in producer gas generally in the form of hydrogen sulphide, but accompanied by small quantities of sulphur dioxide.

For furnace heating by coals containing appreciable quantities of volatile matter, by oil and by coal gas the composition of the air-gas mixture fed to the furnace and the setting of the primary and secondary air inlets is exceedingly important. Much of the efficiency of utilisation depends on the correct setting of the primary air controls and the secondary air dampers. When using producer gas at temperatures up to about 1,000° C. it is generally advisable to close the air control at the injector and thus to use no primary air.

The uses of producer gas are limited by :—

(*a*) Its comparatively low flame temperature (see Chapter V).
(*b*) The presence of dust.
(*c*) The presence of sulphur compounds.

TABLE 9A. PRINCIPAL DIFFERENCES BETWEEN PRODUCER GAS AND TOWN GAS

Property	Producer gas	Town gas
(1) Calorific value B.Th.U./cu. ft. (gross), 60° F., 30 inches mercury, saturated	130–160	450–500
(2) Flame temperature, deg. C. (approx.)	1,600	2,200
(3) Dust, oz./1,000 cu. ft.	0·02–0·03	*Nil*
(4) Sulphur grains/therm with H_2S removed	10–15	2–6
Sulphur grains/therm no H_2S removed	40–55	(All H_2S must be removed as a Statutory requirement)
(5) Specific gravity (air = 1)	0·9	0·42–0·48
(6) Air required for complete combustion cu. ft./cu. ft.	0·9–1·2	4·25–4·5
(7) Hydrogen content, per cent.	14–18	45–55

BLAST FURNACE GAS

Blast furnace gas is rarely used outside the iron and steel industry and comparatively little will therefore be included here upon this subject. When iron

is smelted with coke in the blast furnace a blast of air is introduced through the tuyeres at the bottom of the furnace which burns the coke to carbon monoxide. Part of the carbon monoxide reacts with iron ore in the furnace forming CO_2. The gases issuing from the top of a blast furnace are therefore similar to producer gas made without the addition of steam, but higher in CO_2 content. The calorific value of blast furnace gas is from 90–105 B.Th.U. per cubic foot, and the gas customarily contains about 11 per cent. of CO_2, 27 per cent. of CO, 2 per cent. of H_2 and 60 per cent. of N_2.

The calorific value of the theoretically completely combustible air-gas mixture is only 54 B.Th.U. per cubic foot, and for the production of a high temperature it is thus necessary to preheat both gas and air.

Blast furnace gas can be used under proper conditions quite successfully for industrial heating and for gas engines. It contains a great deal of dust from which it can be purified by appropriate methods. For gas engine work and for heating coke ovens the dust must be removed from the gas almost completely. As an illustration of the potentialities it may be mentioned that like producer gas it is used successfully for heating coke ovens operating with a flue temperature of 1,300°–1,400° C., both gas and air being preheated to 1,000° C.

WATER GAS

Water gas is principally used by gas undertakings as a method of making town gas, complementary to carbonisation. When water gas is distributed by a gas undertaking, it is always mixed with coal gas. There are comparatively few works outside the gas industry that manufacture water gas ; a small number of engineering works use it for such processes as water gas welding, it has been used for heating metallurgical furnaces, and in the chemical industry it finds certain limited applications principally as a source of hydrogen. It seems probable that it will find more applications in the chemical industry in the future.

What is known as " blue " water gas, because it burns with a blue flame, is produced by the action of steam on carbon at a high temperature, the gases produced being carbon monoxide and hydrogen, accompanied by small quantities of carbon dioxide and nitrogen. The range of composition is as follows :—

	Per cent.
Hydrogen	45–51
Carbon monoxide	40–45
Carbon dioxide	3–6
Nitrogen	3–7
Methane	0·1–0·5

The calorific value is about 290–300 B.Th.U. per cubic foot gross, and the net value some 20 B.Th.U. lower.

The solid fuel, coke or anthracite, is raised to incandescence by blowing the ignited fuel with air, the blow gases being allowed to escape to atmosphere, after giving up their heat to waste heat boilers. When the fuel bed is sufficiently hot, the air blast is discontinued, and steam is introduced in its stead. The steam is decomposed in contact with the carbon according to the chemical equation (Chapter IV) :—

$$C + H_2O = CO + H_2$$

at somewhat lower temperatures a reaction sets in which forms carbon dioxide,

$$C + 2H_2O = CO_2 + 2H_2.$$

Both these reactions absorb heat and when the amount of CO_2 in the gases becomes too high because the fuel bed has been cooled, the steam is discontinued and the air blast is restarted. A typical cycle is as follows :—

(1) Blow with air for one minute, gases rejected to atmosphere.
(2) Discontinue air blast, open steam valve, and after allowing a few seconds to remove the blow gas from the generator, open gas valve to scrubber, the gas being now made for use, for four minutes. Sometimes steam is blown up the fuel bed for one minute, down for two minutes, and up for one minute.
(3) Steam shut off, stack valve and air blast opened, and the sequence repeated.

For town gas purposes, a second chamber is added filled with chequer firebrick-work. This chamber is maintained at a high temperature and into it, during part of the time when steam is being blown into the fuel bed, gas oil is injected. The gas oil is decomposed and vaporised yielding a gas rich in hydrocarbons. Some of these hydrocarbons condense forming water gas tar, but those which are permanently vapourised into the gas increase its calorific value considerably. When the blue gas has been carburetted, for example by the addition of some 7–8 per cent. of hydrocarbons, its C.V. may be raised to about 500 B.Th.U. per cubic foot. This gas is known as carburetted water gas.

COAL TAR FUELS

The subject of the production, characteristics and use of coal tar fuels is discussed more exhaustively in this chapter than are other fuels. References to other fuels will be found scattered through this book, often in special chapters devoted to their utilisation, but apart from a little general information in Chapter XXVIII no specific reference is again made to coal tar fuels.

(i) *Sources of Coal Tar.* In Great Britain, crude coal tar is produced mainly as a by-product of the carbonisation of coal at high temperatures. A relatively small amount is produced by the various processes of low temperature carbonisation at present being operated.

The yield of crude tar varies with the nature of the coal and with the type of carbonising plant and the operating conditions.

By high temperature carbonisation, gas works vertical retorts produce 12–14 gallons of tar per ton of coal, gas works horizontal retorts 9–11 gallons and coke ovens about 8·5 gallons. The average amount is about 10 gallons, which is equivalent to a little more than 1 cwt., or approximately 5 per cent. by weight of the coal carbonised.

Low temperature processes yield 15–22 gallons of tar per ton of coal carbonised.

The total annual production of crude coal tar is of the order of 2,000,000 tons, of which approximately 60 per cent. is derived from gas works and 40 per cent. from coke ovens. Practically the whole make is distilled for the production of refined products. These include a range of materials which may be used as liquid fuels and a product suitable for use as a pulverised fuel.

(ii) *Nomenclature of Coal Tar Fuels.* It is possible to manufacture coal tar liquid fuels with kinematic viscosities (cf. Chapter IX), at ordinary temperature (60° F.), within the range 0·1 to 10^{12} stokes. In practice, the differences in viscosity will mean that the various fuels must be heated to different temperatures for their viscosities to fall to the value required for efficient atomisation (0·25 stokes). There is thus established an approximate scale of equi-viscous temperatures which can be introduced into a scheme of nomenclature for the fuels. According to the handbook on the subject of coal tar fuels prepared by the Association of Tar Distillers, there is available a range of coal tar liquid fuels known respectively as Coal Tar Fuel 50, Coal Tar Fuel 100, Coal Tar Fuel 200, Coal Tar Fuel 250, Coal Tar Fuel 300 and Coal Tar Fuel 400. The figures represent the approximate temperature in degrees Fahrenheit at which the particular fuel is in a condition suitable for atomisation. It is the intention,

in this section, to give an outline of the derivation and general characteristics of these fuels. For further information relating to their properties and the technique of their application, the handbook mentioned should be consulted. General information on the use of oil fuel will be found in Chapter XXVIII.

(iii) *Derivation and Characteristics of Coal Tar Fuels.* The extent to which crude tar is distilled is governed by varying market requirements, particularly for the heavier products such as the coal tar fuels. Fig. 8 gives, in outline, the general scheme of tar distillation, showing the composition of the fuels. Special arrangements are generally made for distilling off the aqueous ammoniacal liquor and part of the light oil, after which the water-free tar is put through the main distillation plant.

A further quantity of light oil is then obtained, followed, in order of increasing boiling point, by the fractions known as middle oil, creosote oil and anthracene

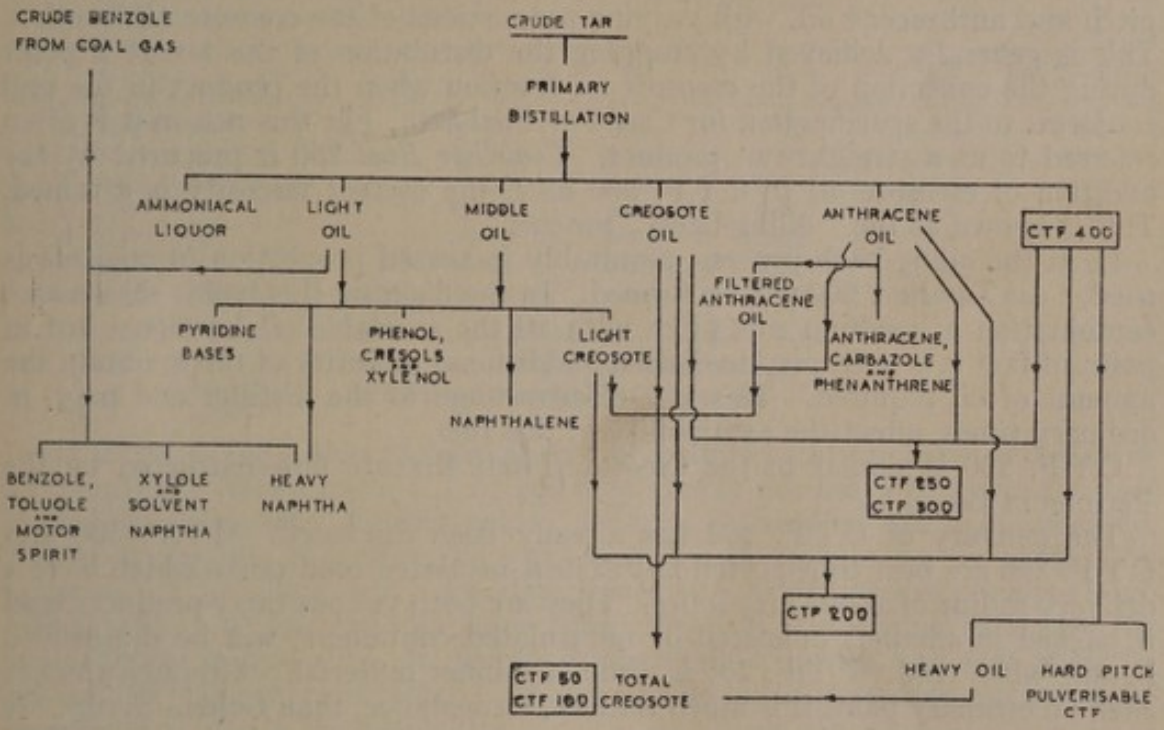

FIG. 8. Tar distillation.

oil respectively. The product remaining in the still is medium soft pitch or Coal Tar Fuel 400. It is generally run off into beds or bays where it is allowed to cool and solidify. It can then be excavated for delivery to consumers in lump form. In practice, it is generally used as a liquid fuel which means that if the lump fuel is received melting plant will be required. The process of melting and handling this fuel is comparatively simple. In certain circumstances this material is supplied in liquid form at a temperature of 300° F. and over, in loads up to about 12 tons. For this purpose, heat-insulated road tanks are employed and deliveries are usually restricted to a radius of about fifty miles from the tar works.

The distillation of tar normally finishes with the production of medium soft pitch. If a *Pulverisable Coal Tar Fuel* or hard pitch is required it is necessary to distil a further quantity of heavy oil from the pitch until a hard and brittle product is obtained which will pulverise readily at ordinary temperature. This part of the distillation process requires experience and special care.

The creosote of commerce is derived from several fractions. Creosote oil and anthracene oil are the main constituents, coupled with the light creosote remaining after the removal of phenols, naphthalene, etc., from the light and

middle oils. The heavy oil obtained during the manufacture of the hard pitch can also be included.

Coal Tar Fuel 50 and *Coal Tar Fuel* 100 are among the several grades of creosote produced by the careful blending of those oils. They distil above about 200° C. (392° F.) and contain at least 99·5 per cent. of material soluble in toluole. C.T.F. 50 is fluid down to 32° F., but is only available for special purposes in certain districts. At ordinary temperatures C.T.F. 100 generally contains a quantity of separated solids consisting mainly of naphthalene, but at temperatures above 90° F. it is completely fluid. It is available throughout the whole country.

Creosote—Grade A, and Creosote—Grade B, as marketed during war-time by the Petroleum Board, are identical with C.T.F. 50 and C.T.F. 100 respectively.

C.T.F. 250 and C.T.F. 300 may be regarded as a combination of medium soft pitch and anthracene oil, with varying proportions of the creosote oil fraction. This is generally achieved by stopping the distillation of the tar at a point during the collection of the creosote oil fraction when the product in the still conforms to the specification for *Coal Tar Fuel* 300. For this reason it is often referred to as a straight-run product. *Coal Tar Fuel* 250 is prepared by the addition of creosote oil to C.T.F. 300 until the correct viscosity is attained. This is known as the " oiling-back " process.

If, in the oiling-back process, a suitably increased proportion of creosote is used, *Coal Tar Fuel* 200 will be formed. In the diagram this fuel is shown as a combination of medium soft pitch with all the available oil fractions, but in point of fact it is necessary to distil an additional quantity of tar to obtain the amount of oil required. This is a disadvantage to the distiller and may, in ordinary times, affect the availability of this fuel.

C.T.F. 200 is similar to the Creosote/Pitch Mixture now marketed by the Petroleum Board.

The delivery of C.T.F. 400 has already been discussed. C.T.F. 300 and C.T.F. 250 are best transported hot in heat-insulated road tanks which have a delivery radius of about fifty miles. They are both viscous tarry products, and if packed in smaller, unheated or uninsulated containers, will be difficult to handle when cold. C.T.F. 200 is a much thinner material. Although thicker than an ordinary paint it is much thinner, for instance, than Golden Syrup. It can be transported by loading warm into road or rail tanks or by barge. C.T.F. 100 can be delivered warm in road tanks, but if rail transport is used, the tanks must be fitted with coils so that the fuel can be warmed by steam on arrival and any separated solids (naphthalene) thereby re-dissolved. Smaller packages, e.g. drums, would likewise require gentle warming before being emptied. Barge-loads can be carried short distances and it is advantageous if the barge tanks are fitted with steam pipes. C.T.F. 50, being fluid down to 32° F., can be carried in any suitable container under any but the most severe weather conditions.

Hard pitch or Pulverisable Coal Tar Fuel is generally delivered in lumps ranging in size from 18 inches cube downwards. Before being fed to the pulveriser the lumps must be crushed into pieces $\frac{1}{2}$–1 inch in size, this operation being best performed by a spike roll crusher. If stored in the open the hard pitch should be kept in lump form as rain or casual water will then easily drain away. The pre-crushing operation should be delayed as long as possible and should immediately precede the pulverising process. The presence of water can increase the load on the pulveriser. The handling of the pulverised fuel is discussed later.

(iv) *General Properties of the Coal Tar Fuels.* As delivered, the coal tar fuels are technically free from water. They have a low sulphur content which never

exceeds 1 per cent. by weight and is generally considerably below that figure. They are almost ash-free, the amount of ash never exceeding 0·75 per cent. by weight, which is the maximum permitted in the solid pulverisable fuel. In the creosotes and the thinner tarry fuels it is almost negligible.

The gross calorific values of the fuels range from 17,000 B.Th.U. per lb. for C.T.F. 50 and C.T.F. 100, down to 16,000 B.Th.U. per lb. for C.T.F. 400 and the pulverisable fuel. The others have intermediate values. Calculated on a basis of 1 gallon of fuel at ordinary temperature the above-mentioned calorific values become 177,000 B.Th.U. and 200,000 B.Th.U. respectively.

Typical ultimate analyses of the fuels are given in Table 10.

TABLE 10. ULTIMATE ANALYSES OF COAL TAR FUELS

	Per cent.				
	C	H	S	N	O (by diff.)
C.T.F. 100 *	89·30	6·90	0·13	0·84	2·83
C.T.F. 100 †	86·50	8·00	0·25	0·84	4·41
C.T.F. 200	89·86	6·05	0·39	1·11	2·52
C.T.F. 250	89·97	5·90	0·44	1·16	2·45
C.T.F. 300	90·08	5·73	0·49	1·22	2·39
C.T.F. 400	90·42	5·23	0·65	1·38	2·19
Pulverisable C.T.F.	90·66	4·90	0·86	1·42	1·70

* London horizontal retort tar oil.
† Midland vertical retort tar oil.

C.T.F. 50, being a thin mobile oil, can be pumped and atomised at ordinary temperature. When using C.T.F. 100 it is merely necessary to warm the fuel to slightly above the " fluidity-point " for it to be handled with equal facility. The temperatures to which the remaining liquid fuels must be heated for purposes of handling etc., are determined by their viscosity-temperature relationships. It has been explained that the title of each fuel indicates the approximate temperature in degrees Fahrenheit to which the fuel must be heated for efficient atomisation. It should not be necessary to exceed these temperatures at the burner nozzle.

For pumping, it is generally considered that the viscosity of the fuel should not exceed 12·5 stokes. A figure of 10 stokes will allow a reasonable margin, and for their viscosities to fall to this value the fuels must be heated to the temperatures indicated.

C.T.F. 200	85° F.
C.T.F. 250	140° F.
C.T.F. 300	180° F.
C.T.F. 400	270° F.

Coal Tar Fuels 200, 250, 300 and 400, among the liquid fuels, contain varying amounts of material insoluble in toluole. The maximum allowed ranges from 15 per cent. to 30 per cent. by weight, those being the figures for C.T.F. 200 and C.T.F. 400 respectively. This toluene insoluble material consists of resinous compounds of high molecular weight containing a high percentage of carbon. The particles are of microscopic dimensions and form a stable uniform dispersion in an oily medium.

These fuels should not be mixed with petroleum fuels except on expert advice. Contact of one with only small amounts of the other may cause local instability and coagulation with the possibility of a stoppage. It may be advisable to steam out the system when changing over from one fuel to the other.

(v) *Application of Coal Tar Fuels.* Considerable experience has now been gained in the use of coal tar fuels and with the aid of the information now available the design of new installations or the conversion of existing oil-burning units can follow well-established principles.

As a general rule, any fuel oil system which is fitted with tank heating and the usual flow heaters can comfortably handle either C.T.F. 50 or C.T.F. 100. When using C.T.F. 100 it may be necessary to increase the wattage of electrical heating equipment and, unless ring mains and gravity feed lines are drained

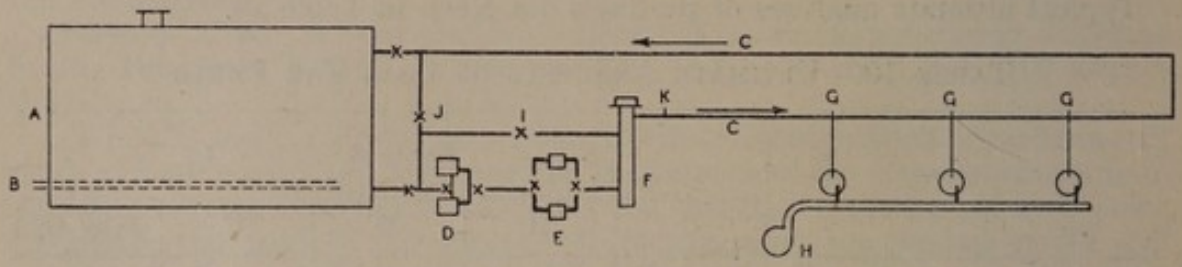

FIG. 9. General arrangement of installation for burning lighter tar oils.

A. Lagged storage tank.
B. Tank heating.
C. Ring main containing steam tracer or electric heating cable within lagging.
D. Duplicate steam jacketted filters with necessary " T " cocks.
E. Duplicate pumping arrangement with necessary " T " cocks.
F. Thermostatically controlled flow heater raising fuel to atomising temperature.
G. " T " valves to low pressure air burners.
H. Fan supplying atomising air.
I. Pressure relief valve.
J. Ring main pressure regulating valve.
K. Thermometer.

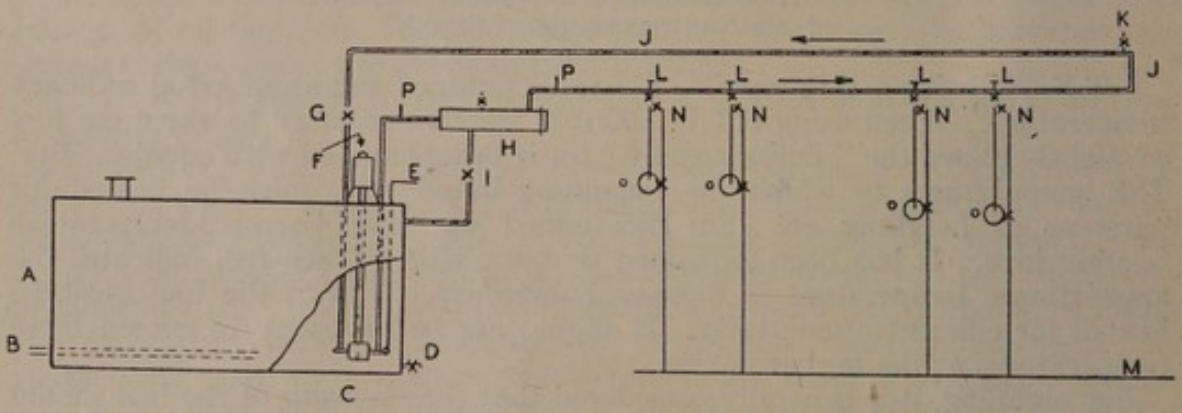

FIG. 10. General arrangement of installation for burning heavier tar oils.

A. Fuel storage tank.
B. Steam heating coil or thermostatically controlled electrical heater.
C. Submerged gear pump.
D. Tank drain cock.
E. Pump drain cock.
F. Pump motor.
G. Ring main pressure balancing valve.
H. Electrical flow heater thermostatically controlled at fuel atomising temperature.
I. Combine flow heater drain cock and pressure relie valve.
J. Ring main lagged with the inclusion of steam tracer line or electrical heating cables, graded downwards from vent cock " K."
L. " T " type valves to steam jet burners O.
M. Steam line.
N. Steam valve for scavenging lines to burners.
P.P. Thermometers.

when closing down, such pipes should be provided with a steam tracer line or electrical heating cable. Pumps should also be kept warm.

The burning of the more viscous liquid fuels calls for a rather specialised technique. The best arrangement is to circulate the fuel, by pumping, from the storage tank through a ring main past the burners, and back to the tank. Gravity feed from the storage tank to the burners is quite practicable in small installations. The temperature of the fuel circulating in the ring main is limited by the pressure of the steam available. If any further boosting is required to raise the temperature of the fuel to the atomising point, electrical or other flow heaters can be installed in the system. At least 2½ times the fuel

consumption should be circulated through the ring main. This will stabilise temperature and pressure conditions in the system.

The storage tank and all pipe-work should be free from joints made with rubber asbestos compounds, and plain asbestos millboard is quite satisfactory. The fuel lines throughout should be steam-jacketed or lagged, with the inclusion in the lagging of a steam tracer line or electrical heating cable. Steam-jacketed pumps should also be used.

Typical installations are shown diagrammatically in Figs. 9 and 10.

The usual type of heavy oil burner can be adapted to most of the fuels. Pressure jet, controlled pressure jet, rotary, medium pressure air and low pressure air burners are all applicable to one or more of the less viscous fuels, such as the Coal Tar Fuels 50, 100, 200 and 250. For the more viscous grades, such as Coal Tar Fuels 300 and 400, medium pressure steam jet burners are generally employed, using steam at a pressure of 50–100 lb. per square inch. Suitable burners are offered by a number of manufacturers. Coal tar fuels are not suitable for use with very low capacity burners with correspondingly small orifices. Similarly, needle valves should be avoided and the ported type of valve employed.

Coal tar fuels have a slight corrosive action upon brass and similar non-ferrous metals. It is advisable therefore that valves, filters and other fittings should be made of iron or steel.

Pulverised coal tar fuel practice follows pulverised coal practice very closely, but the following differences may be noted :—

(*a*) The pulveriser should be of the air swept impact type and must be fed with cold air. Edge runner mills, ball mills, tube mills and others with a positive grinding action are not suitable.

(*b*) The velocity of flame propagation is of the order of 80 feet/second, and burner nozzle velocities should exceed this figure.

(*c*) Pipe lines conveying pulverised coal tar fuel should be suitably shielded and insulated from radiant heat.

(*d*) The burner nozzle must be a plain tube either air or water cooled. Spreader devices are not applicable.

(*e*) Ring main systems such as those used for pulverised coal are not suitable for pulverised coal tar fuel.

For conveying the pulverised fuel from the pulveriser to the burners, two methods are commonly practised. In the first, where the furnaces to be fired are of large capacity, the fuel is fed direct to the burners. In the second method, which is used when a number of furnaces are to be fired at irregular periods, the fuel is discharged into a pulverised fuel storage bin *via* a cyclone separator, and the fuel is then conveyed to the burners by independent fan and feeder units which may be started up individually or severally and which do not necessarily demand that the pulveriser be operating at the same time. The storage bins should be situated in cool, dry places.

The theoretical amount of air required to burn 1 lb. of coal tar fuel is of the order of 160 cubic feet, but in designing plant it is customary to base calculations on a figure of 200 cubic feet of air per lb.

ELECTRICITY

Electricity is used for furnace heating for special purposes to an increasing extent. Heat is converted into electricity at generating stations with a thermal efficiency which up to the present is never above 30 per cent. The average value for the generating station is about 20 per cent. It is then transmitted to the place of use and is reconverted into heat within the furnace. This conversion may be effected by one or more arcs, by primary or induced currents flowing through resistances, or by currents induced within the charge itself

(cf. Chapter XXI). Heating and annealing furnaces are usually of the resistance types; steel melting furnaces operate through induced currents within the charge and with arcs; arcs are used particularly when non-conducting material is to be heated to a very high temperature, for example in the manufacture of calcium carbide, and in the production of fused alumina.

Electricity is used in furnace work for special reasons rather than from considerations of heat energy cost though, where off-peak current can be supplied at a low price, cost considerations may be favourable, e.g. the use of resistance furnaces for carbonising coal in Canada. Electric furnaces are discussed in detail in Chapter XXI.

The loss of heat in the flue gases which is unavoidable with solid, liquid and gaseous fuels does not occur with electricity. Moreover, since the heat can generally be generated exactly where it is wanted, the higher efficiency in utilisation acts against the higher first cost per therm and may cause electrical furnaces to be more economical than other types under favourable circumstances.

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CHAPTER III

THE STORAGE OF COAL

Oxidation of coal—Chemistry and physics of oxidation of coal—Practical principles of storage—Deterioration of coal in store (coking power, C.V., size, friability)—Watching temperatures—Dealing with fires—Storage of opencast coal.

THE object of storing coal is to provide reserves to keep plant working during a period when supplies may be dislocated. The dislocation may be caused by the interruption of transport facilities, due, for example, to the weather and the more normal delays incidental to rail transport. Furthermore, storing coal in the summer to meet the increase of consumption in winter

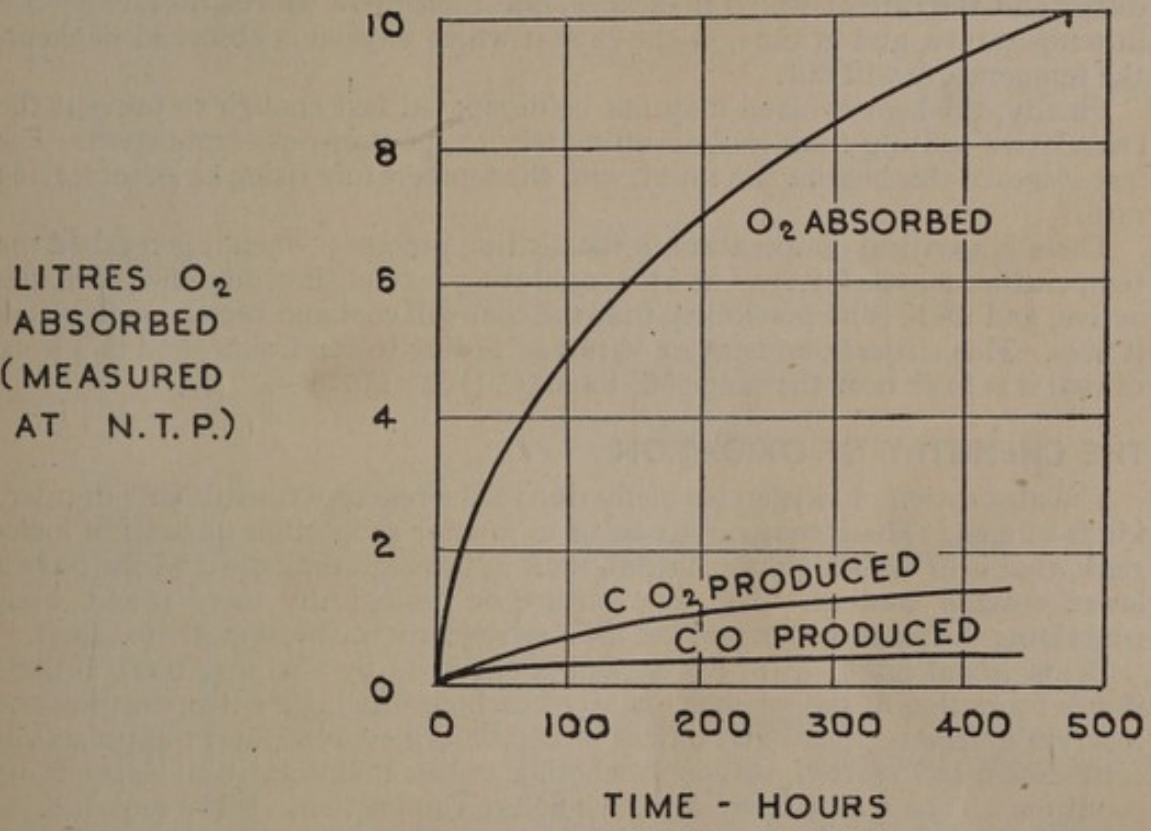

FIG. 11. Rate of absorption of oxygen by a coal.

(Reproduced by permission of the publishers (Longmans, Green) from "Coal: Its Constitution and Uses," by W. A. Bone and G. W. Himus.)

reduces the seasonal fluctuations in the demands on the mining and transport industries. Even under good conditions it is unlikely that coal will be delivered at exactly the rate at which it is consumed, and there must therefore be a stock that will absorb the difference from time to time existing between deliveries and consumption.

Stored coal, being exposed to the atmosphere, is subject to attack by the oxygen in the air. Under certain conditions the coal may oxidise so rapidly as to heat and ultimately ignite, and then the coal will depreciate greatly in value. Some coals merely disintegrate in storage. It is important to know how to store coal in order to minimise these unwelcome effects.

THE OXIDATION OF EXPOSED COAL

Fresh coal absorbs oxygen when it is exposed to the air. Figure 11 shows typically how the oxygen absorbed by a coal passing a 90-mesh sieve exposed

in an atmosphere of oxygen at about 108° C. increases with the passage of time.

The absorption of oxygen is accompanied by the evolution of heat and there is therefore a rise in temperature.

At first the temperature rises only slightly, and if the heat produced can be dissipated the rise in temperature remains slight or even imperceptible and there are no serious consequences.

However, if the conditions do not favour the dispersal of the heat produced, the temperature will rise noticeably. As the temperature rises, the rate at which oxygen is absorbed also rises, and this in turn means an acceleration in the rate at which heat is evolved, so that the process is continually aggravated.

But as the temperature rises the rate at which the heat is dissipated increases. Eventually, a balance may be struck between the rate at which heat is produced and the rate at which it escapes, and then there will be no further rise in temperature, and in time, as the rate at which oxygen is absorbed slackens, the temperature will fall.

Finally, the heat evolved may not be dissipated fast enough to prevent the cumulative heating from leading ultimately to spontaneous combustion. The last stages of this heating are rapid, with the temperature rising at an increasing rate.

There is a critical temperature in the heating process ; when it is reached the temperature inevitably rises at an accelerating rate until combustion becomes active, and there is no possibility that the coal will cool and become safe before it fires. This critical temperature varies according to conditions, and in a heap of coal it is in or near the range 50° to 80° C. (122°–176° F.).

THE CHEMISTRY OF OXIDATION

The absorption of oxygen is chiefly due to the reactive constituents in many kinds of coal. These compounds occur in greater proportion in coals of lower rank, that is in coals high in volatiles, than in the high rank coals which have a lower volatile content. The less mature or geologically more recent coals containing the least carbon and the most oxygen are the most easily oxidised.

Below about 80° C. (176° F.) oxygen is absorbed by coal and there is little if any formation of the gaseous oxides of carbon. At higher temperatures the reaction quickens, steam and oxides of carbon are evolved, and the process of combustion has started, although whether it can maintain itself depends on conditions. The temperature at which active combustion, visibly recognisable as such, occurs and progresses is much higher still.

As conditions are so variable, even for one variety of coal, there is no generally recognised or satisfactory method of measuring experimentally the liability of a coal to heat spontaneously. A method that may be found useful is to pass a stream of oxygen over the dried coal under prescribed conditions using various initial temperatures until a temperature is found from which the coal ignites in sixty minutes.* There is experimental evidence that liability to spontaneous combustion can be linked with the results of this test somewhat as follows :—

Temperature from which the coal ignites in sixty minutes ° C.	Degree of liability to spontaneous heating
100–109	dangerous
130–138	medium dangerous
above 150	not dangerous

* Burian, described in Gluud, " International Handbook of the By-Product Coking Industry," English edition, p. 78.

Figure 12 indicates the sequence of events and is based on the behaviour of fine coal in small quantities heated by external means in a current of air or oxygen ; these are conditions applied in laboratory investigations. The curve reproduces a complete history for such conditions, but needs to be interpreted for application to coal stored in heaps.

The self-sustained process of combustion, shown as starting at 200° to 270° C. (393–518 °F.) does not start lower because the cooling effects operating under laboratory conditions are greater than the heating effects due to oxidation. In such circumstances the heating cannot become cumulative so as to be self-sustained.

In a heap of coal, where there is no external source of heat, the tendency for oxygen to be absorbed decreases as the affinity of the coal for oxygen is satisfied ; the rate at which oxygen is absorbed may therefore fall so that the rate at which heat is developed is less than the rate at which it is lost, when the

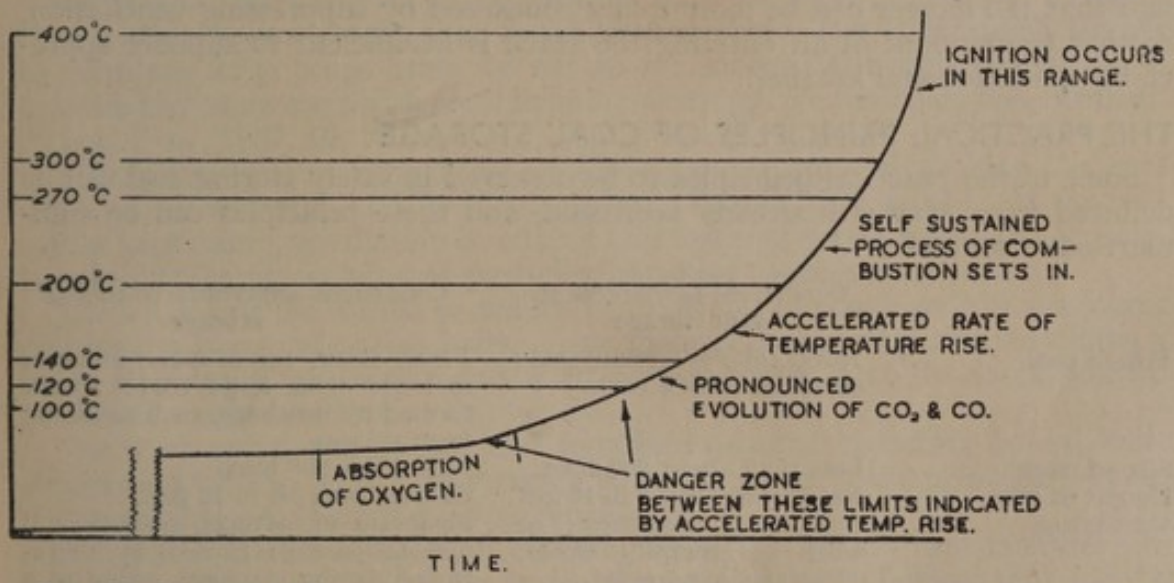

FIG. 12. Sequence of events in the oxidation and firing of coal, based on laboratory results.

curve will show a maximum, whereafter its slope is negative. This maximum usually occurs well below 100° C. (212° F.), and the subsequent negatively sloping portion of the curve is indicated by a dotted line.

If the coal in a heap is not fine enough only the very earliest portion of the curve will be traversed, and the maximum with the subsequent negatively sloping portion of the curve will appear.

If the maximum does not appear, and the curve continues with a positive slope, heating is cumulative, and the whole of the curve may be embraced in the range of self-sustained combustion.

Before this can happen a predominant fraction of the coal must be finer than $\frac{1}{8}$ or even $\frac{1}{16}$ inch.

There will be some inverse relationship between the fineness of the coal and the scale of the time-axis.

THE PHYSICAL CONDITIONS OF OXIDATION

It has been shown that there are in a store of coal two opposing processes : the production of heat due to oxidation and incipient combustion ; the loss of heat due to radiation, conduction and convection ; and that it is the balance between these two processes that decides the extent of heating.

Given access to oxygen, the production of heat varies with the surface exposed by the coal. For a given weight of coal the surface exposed is greater the finer

the coal; and conversely it is less with larger coals, so that heat is produced more rapidly from finer coals. In fact, with coals in which no fines or dust occur the rate at which heat is produced is negligible and difficulties due to spontaneous heating are confined almost exclusively to coals in which fines or dust are admixed.

In the loss of heat the chief part is played by air passing through the heap and removing heat as sensible heat. Here again the temperature rise will be kept low in a heap of large coal because the texture of the heap is open and air-currents can pass through the heap readily; whereas the greater production of heat when the coal is fine and the texture of the heap close, cannot be eased so readily because air-currents do not find a continuous passage through the heap.

It is because of this that ventilation of a heap of stored coal is practised. Experience suggests, however, that if a heap of stored coal would fire spontaneously it is difficult and expensive to ventilate it enough to make it safe, and that the danger can be more easily countered by suppressing ventilation, so that the amount of air entering the stack is insufficient to support appreciable oxidation and heating.

THE PRACTICAL PRINCIPLES OF COAL STORAGE

Some of the practical principles to be observed in safely storing coal can be deduced from what has already been said, and these principles can be summarised as follows :—

	Conditions favourable to safe storage	Conditions unfavourable to safe storage
Size of coal ..	Nuts, cobbles, round coal; exclusion of fine coal.	Fines, slack, run-of-mine, alone or included with large coal; smalls formed by breakage, such as falling from a chute.
Size of store ..	Less than about 200 tons.	More than 200 tons.
Height of store ..	Not greater than 8 to 10 feet.	Greater than 10 to 12 feet.
Ventilation ..	Adequately promoted; or what is generally easier, effectively suppressed.	Moderate or natural, especially if due to passage caused by segregation during stacking, owing to a pocket of large spaces against pillars or walls.
Age of coal ..	Greater than 3 to 6 months after mining.	Up to 10 weeks.
State of coal ..	Having heated and cooled.	Freshly wrought.
Type of coal ..	Hard or dull, anthracite, anthracitic coal, splint, cannel.	Soft and bright; bituminous, subbituminous, lignite.
Site of stack ..	Most open situations on clean, firm ground.	Near a source of heat, such as a boiler house wall, or overlying steam pipes, or mixed with organic waste or oily material.

While it is advocated that ventilation should be suppressed, in a small heap or one not too high the ventilation that does occur and the distribution of air currents will help to keep conditions safe, and with the smallest heaps the suppression of ventilation will not be so urgent; the ventilation may even be promoted.

The liability to oxidation is lessened with increasing age of the coal as some of the unstable constituents become oxidised. Old coal will not therefore be as susceptible as freshly wrought coal.

THE SIZE OF THE COAL AND THE CORRECT METHOD OF STACKING

Small coal is the cause of nearly all storage difficulties. In coals of mixed sizes, segregation is the most frequent cause of spontaneous heating, because the larger pieces of coal open the texture of the heap and facilitate the type of

ventilation necessary for heating to proceed. By coals of mixed sizes are understood coals that include fines, and run-of-mine coal. Uniformly fine coal is, if anything, less liable to heat than mixed sizes.

Singles are likely to be safe and larger sizes are quite safe, provided that fines have not been produced during handling. The danger of heating is greater under a chute where the finest coal falls or is formed by breakage. Sized coal should not therefore be dropped from a height when it is being stacked. Stacking coal by a drag-line conveyor has the advantage of producing less fines than dropping it from an overhead conveyor, and is thus less likely to lead to firing.

The coal should be spread out as it is stacked so that there is no local segregation of sizes, and lumps should be distributed as uniformly as possible among the fines.

THE SUPPRESSION OF VENTILATION

Ventilation, especially in a large heap of coal, can be suppressed to a degree that is effective more easily than it can be promoted. This is because attempts to ventilate large heaps often do not go far enough, although ventilation is practised by some users with considerable success (cf. Dunningham and Grumell, *J. Inst. Fuel*, 1937, **10**, 170).

Ventilation can be reduced by avoiding segregation and packing the coal tightly and uniformly and by blanketing the heap with fine coal. If the coal can be kept damp, ventilation is reduced still further because moisture decreases or even fills the spaces between particles, impeding the movement of air.

The stack may be walled to reduce ventilation. A rough wall of the larger rectangular lumps of coal is built. A permanent coal storage site may have a low wall most of the way round it to protect the base of the stack whence ventilating currents start.

The wind assists ventilation, and to reduce its effect the stack is best built elongated with its longer sides running in the direction of the prevailing wind ; this generally means south-west to north-east.

A method that is completely effective when the all too rare facilities are available is to store coal under water, for example in a bunker that can be flooded ; or to store it in an atmosphere of flue gas.

THE DIMENSIONS OF THE STACK

Trouble seems rarely, if ever, to arise in heaps of less than 200 tons. It is usually easier to control the height of the stack than to limit the weight of coal in it. A safe height seems to be about 8 feet, but if the coal is of a suitable size and type it may go up to 16 feet. Hard nuts have been stored safely up to 20 feet and larger coals and anthracite up to 30 feet.

A stack may be built up in stages, whereby the lower layers have had an opportunity to pass over their critical stages before the upper layers are added. Thus, a 6-foot stack may be increased to 10 feet after six months, or by 18 inches every three months ; or an 8-foot stack may be built up to 16 feet after nine months.

The advisability of adding to the heap may be judged by the temperatures in the existing part.

The height of the stack should follow the contour of the ground.

Coal occupies from 40 to 50 cubic feet per ton ; the more mixed sizes give the more compact heaps, and the heap is also more compact when the coal is packed carefully.

THE SITE OF THE STACK

A concrete floor provides an ideal base for a stack, but a well rolled floor of ashes may be satisfactory ; hard clay or chalk may also be satisfactory. Peaty

ground should be dressed with a few inches of ballast or ashes and then rolled to make it as firm as possible.

An important consideration in the preparation of the floor is the clean and complete recovery of the coal. Apart from this, vegetable matter and waste organic matter should always be cleared as it may contribute to the start of heating.

A wet site is not necessarily advantageous. Storage under cover offers no advantage to compensate for the outlay in providing the cover.

THE PERIOD OF ACTIVITY IN A COAL STORE

Coal oxidises most rapidly when it is freshly mined, and the rate at which it is affected decreases as time passes.

Spontaneous combustion usually occurs within four weeks to four months after stacking. A coal that is eight months old can be regarded as safe, if there is no heating in the heap, or if what heating there is is not increasing.

It has been reported by an undertaking that stores large quantities of coal that there is considerable risk in stacking freshly wrought coal in contact with coal which has been in stock for some time. Experience has shown that this point of junction is a fruitful site for fires.

THE DETERIORATION OF COAL ON STORAGE

Deterioration obviously depends on the amount of heating that goes on in a store of coal, but if there is no appreciable heating, deterioration is at a minimum and although data are not available to indicate the depreciation of the various qualities of coal with time, it is possible and useful to know the order of the amount of deterioration to be expected.

Bright coals—clarain and vitrain—are affected more than dull and hard coals—durain. A freshly broken surface is the most active for the absorption of oxygen and the surface of old coal becomes dull. This is said to affect the market value of the coal, but alone it does not indicate any noticeable depreciation in quality.

THE LOSS OF COKING POWER

Of all the properties of a coal, the coking power is easily the most rapidly and extensively affected. Frequently the effect is very marked when there is no other indication of a change.

The coking power falls off much more rapidly during the first few months of storage than later. It is not certain that a coal ever loses all its coking power when it is stored, but coals that are originally weakly coking may have their coking power reduced to insignificant proportions. The rate at which coking power is reduced is also seriously affected by heating.

If coking power is an important characteristic of the coal to be stored, a coal should be selected with a higher coking power than would otherwise be required, so that the excess of its coking power acts as a reserve.

THE LOSS IN CALORIFIC VALUE

Steam-raising power is often diminished by exposure to the weather and atmosphere, and is not unnaturally ascribed to a loss in calorific value. If the coal has not heated appreciably the loss in calorific value is slight, so slight as a rule as to be imperceptible. The loss in steam-raising power is likely to be due rather to an increased loss of small coal as unburnt matter through the grate.

If coal that is stored does not heat appreciably, the loss in calorific value rarely exceeds 0·5 to 1 per cent. per annum. An examination of twenty Scottish gas coals (Jamieson and Skilling, "The Deterioration of Coal on Storage," *Trans. Inst. Gas E.*, 1938, **88**, 663) showed a mean loss of calorific value of 0·7 per cent. in upwards of 3½ years.

If the coal heats, the loss in calorific value may reach any reasonable value, according to the extent of heating.

THE EFFECT OF WEATHERING ON THE SIZE OF STORED COAL

Coal may disintegrate under the action of the weather and small or fine coal may thus be formed. This formation of fine coal will not greatly affect the tendency of the coal to heat as the coal is always likely to oxidise faster than it disintegrates.

Examples of the extent of disintegration are afforded by work done on Scottish coals (Jamieson and Skilling, *loc. cit.*) and by the Fuel Research Board (Annual Reports, 1937, 69; 1938, 83; 1939, 60). Fourteen Scottish gas coals from which all coal passing a 1¼ inch screen was removed contained from 3 to 20 per cent. of coal passing a 1½ inch screen after 2 years of exposure and from 4 to 28 per cent. after 44 months. Six similar coals had originally from 30 to 45 per cent. of coal passing a 1-inch screen, and this size was increased by percentages varying from zero up to twelve. At the Fuel Research Station, midland, north-country and Welsh gas, house and steam coals were stored for periods up to a year. The results are summarised in Table 11.

TABLE 11

Period of storage—months	0	6	9	12
S. Wales steam, 2½–1⅜ in., per cent.	100	93·2	91·6	90·7
Warwickshire house, 2½–1⅜ in., per cent.	100	91·4	90·0	87·3
Nottinghamshire house and steam, 2½–1¼ in., per cent.	100	82·2	73·1	69·4
S. Yorkshire gas R, above 2 in., per cent.	43·7	37·4	34·7	30·8
S. Yorkshire gas T, above 2 in., per cent.	54·0	46·3	48·2	47·8

Coal does not seem to break down progressively to the smallest sizes. The process appears to cease eventually when "natural grades" have been produced (Scarf, Hall and Yearsley, "The Storage of Coal: the Effects of Storage without Rise in Temperature," *Trans. Inst. Min. Eng.*, 1941, Vol. C, pt. 4, 88–129) owing to the splitting of the coal along bedding planes and at partings (Annual Reports of the Fuel Research Board, *loc. cit.*).

THE FRIABILITY OF WEATHERED COAL

The Warwickshire and Nottinghamshire coals used in the investigations at the Fuel Research Station (*loc. cit.*, 1938, 83) were subjected to a tumbler type of test in a rotating drum in which the coal was dropped a few inches a few times in each revolution. Some of the results of these tests are reproduced in Table 12. These results show that a coal when it is handled after weathering will break more easily. This is important in boiler firing and other furnace work.

With small coal disintegration does not matter, but with large sized coal it

TABLE 12

Period of storage—months	0	6	9	12
Warwickshire house, 2½–1⅜ in., per cent.	87·7	75·9	74·6	68·1
,, ,, 1⅜– ⅞ in., ,, ,, ...	6·9	13·3	14·8	16·7
Nottinghamshire house and steam, 2½–1¾ in., per cent.	81·4	58·0	49·1	49·7
,, ,, ,, ,, 1¾–1⅜ in., ,, ,,	9·6	15·1	16·8	17·1
,, ,, ,, ,, 1⅜– ⅞ in., ,, ,,	4·7	12·1	14·4	13·7

may be important. The size may be an essential quality of the coal, and in furnace work for example it may mean that there will be an undue loss of combustible material through the grate, and this, as previously indicated, will produce the impression that there has been a decrease in the calorific value.

Cannels and anthracites disintegrate little if at all, and hard, dull coals disintegrate less than soft, bright ones.

THE DURATION OF STORAGE

As deterioration is most rapid in the earlier months of storage and its rate is always decreasing—provided heating does not eventually accelerate it—stored coal should not be used while fresh supplies are available. In this way the loss is least because a new lot of fresh coal is not being exposed continually, and the rate at which the stored coal deteriorates is always falling ; moreover, as the second handling of the stored coal is deferred, labour costs are reduced.

PRECAUTIONS TO BE TAKEN IN A STORE OF COAL

At regular intervals of about a week during the first three months of storage, the temperatures of different places in the stack should be observed.

A rough method that may be found suitable for doing this consists in inserting metal rods into the heap at intervals of 10 to 20 feet, with their ends projecting, and every few days feeling how hot these projecting ends are.

It is better, however, to use wrought-iron or steel pipes closed at the lower end and to take temperatures by lowering a maximum-reading thermometer down them. The pipes should be built into the stack as it is made up. They should be from 15 feet to 50 yards apart according to the size of the stack, the kind of coal and economic considerations. The temperatures can then be read at various depths and should be recorded on a chart. Once the danger period is past the readings can be discontinued. Spontaneous combustion is generally considered most likely to occur within four weeks to four months after stacking.

The critical temperature varies considerably according to conditions and is generally taken to lie between 55° and 70° C. (131° and 168° F.), but it may be lower if conditions are unfavourable. It can only be determined to within about 5° C. (9° F.) by experience.

A slow and steady rise in temperature must be expected, when the critical temperature is reached, to accelerate rapidly and culminate in a fire. Adequate graphing or logging of the temperature-history of the respective points in the stack will give clear warning of this.

DEALING WITH FIRE IN A COAL STACK

When the critical temperature is reached, it is wrong to delay in the hope of a spontaneous improvement in conditions. If the precautions are not taken forthwith, a fire is inevitable, and the affected area will spread.

The danger usually occurs at points from 3 to 7 feet below the surface, where ventilation is sufficient to promote heating and the depth of coal sufficient to prevent a rapid enough loss of heat. Temperature observations will indicate the exact zone, which may also be shown by a light grey smoke and a distinctive odour, though by this time the fire will be established.

The seat of the fire must be dug out by digging a hole, trench or valley, completely exposing the hot place, and scattering or using the hot coal.

A crane and grab or similar mechanical appliance greatly simplifies this operation. If a fire has started, spraying with water is ineffective unless a very large volume can be used.

In the early history of a stack of coal, however, spraying with water has a twofold advantage. It interferes with ventilation and may restrain the tem-

perature below its critical level while the heap is piloted through its peak heating period.

THE STORAGE OF OPENCAST COAL

A recent development in coal-getting is a reversion to the distant past in the opencast working of outcrop coal. Under suitable conditions a seam, as it dips from its bassett edge, is worked with suitable types of excavating machinery. The output of a productive seam becomes large at an early stage in the development of operations, and the storage of some or all of the coal close to the working or at selected storage sites becomes necessary or may form part of the policy of disposing of it.

Generally, the bassett edge of the seam is very inferior and has to be discarded. As the seam is followed in, the coal may remain rather poor, but nearly always improves rapidly, at times approaching pit-coal in quality ; it is usually weathered to various degrees and therefore generally has a high oxygen content, with a corresponding reduction in its carbon and hydrogen contents and its calorific value. The oxidation entails a high inherent moisture content and generally a high degree of friability. On the other hand, it is not uncommon for opencast coal to have a lower ash content than the coal from the same seam from a nearby pit ; and opencast coals with an unexpectedly high coking power have been won.

Table 13 gives a few examples of opencast coals. It must be remembered that the coals of this country, as normally mined, cover a wide range of types. When these coals are worked at their outcrops, further differences, varying rapidly with the depth of cover and with other conditions, are imposed on the original differences. The figures given in Table 13 therefore serve only as examples and are not intended to be representative.

TABLE 13

	Nottinghamshire			South Yorkshire	South Wales
	(*a*)	(*b*)	(*c*)		
Moisture as received, per cent.	31·0	15·9	—	8	8·4
Air-dry basis					
Moisture, per cent.	10·3	7·8	3·4	3·2	4·0
Volatile matter, less moisture, per cent.	33·4	32·1	33·6	33·6	29·5
Fixed carbon, per cent.	50·8	58·8	59·0	60·7	61·5
Ash, per cent.	6·3	1·2	4·0	2·5	5·0
Dry, ash-free basis					
Volatile matter, per cent.	40·0	35·3	36·4	35·6	32·4
Calorific value, B.Th.U./lb.	11,200	13,120	14,700	14,780	14,630
Carbon, per cent.	70·7	77·0	82·3	—	—
Hydrogen, per cent.	3·8	4·8	5·3	—	—
Sulphur, per cent.	—	—	—	0·78	0·75

The Nottinghamshire examples are from (*a*) near the bassett edge, (*b*) from further in, and (*c*) for purposes of comparison, from the same seam as won in a nearby pit. The South Yorkshire and South Wales coals are examples of good opencast coals, though these are not the best that can be got.

The highly oxidised opencast coal may not be liable to serious spontaneous heating. But generally it would be unwise to assume that the coal is any less liable than pit coal. The coal is in some ways of a new type, and is often stored under conditions much less favourable to the avoidance of trouble than in

ordinary times. Until experience shows that in any particular set of circumstances the normal precautions for storing coal are unnecessary, it is not advisable to relax any of the precautions that would otherwise be practised. This is especially so as the quality of the coal improves steadily as the working of the seam progresses.

Occasionally at sites for the storing of opencast coal, space may not be seriously limited, and then there is no difficulty in confining the height of the stack to 8 feet, which is a compromise between a height low enough to be quite safe and an economic use of the area of the site.

Unfortunately, however, ground is usually more precious and then it is necessary to bestow every precaution possible on the formation of the heap.

The whole of the site on which the coal is stored should be used at once so that the coal is built up all over the heap in thin layers. Owing to the great range and uneven mixture of sizes in opencast coal, it is very easy to get a segregation of large coal at the bottom of the stack so that air passes freely through large channels traversing the whole of the base of the stack ; and this condition is made worse by occasional vertical or oblique veins of large coal running from top to bottom through the heap. No effort should be spared to avoid this very dangerous segregation.

If excavating machinery of a kind that can dig into the heap is available, and in connection with opencast working it often is, precautions need not be so stringent when these appliances can be brought to bear on the heap whenever it is necessary. If, when trouble occurs, the heap has to be dug into by hand, everything that can be done to ease this operation is helpful. In particular, if the coal is disposed in long narrow heaps, a heated portion can be isolated by a valley cut across the heap. A long narrow heap is also sometimes easier for certain kinds of excavating machinery to deal with than a compact one.

There should be no attempt at ventilation, unless on the very smallest heaps —say of 200 or 300 tons. If the coal can be compacted in its heap by the traffic of lorries and dumping cars over it, much will be done to impede the percolation of air through the coal.

At appropriate points in the heap, where they will not interfere with traffic handling the coal, pipes or rods can be inserted for observations on the temperature of the heap. A temperature of 50° C. (122° F.), especially if the rise in the temperature is occurring at an accelerating rate, indicates dangerous conditions. A dangerous situation must never be expected to improve spontaneously.

When a fire, according to observations of the temperature in the stack or other signs, appears imminent, the heated coal must be dug out and used or scattered in a thin layer over adjacent parts of the heap or on neighbouring ground.

If the coal shows signs of heating the condition may be satisfactorily reduced by steadily spraying water over the affected area in the form of a strong rain for some days. If the heating is more serious and is localised, the area may be dealt with effectively by running water on to it from a hose. It should be realised, however, that water is not necessarily a reliable way of meeting such a situation. This is especially true if a diffused heated area is treated with a jet of water that only covers a small area, for a strong flow of water in one place will find a preferred path down through the heap instead of percolating all through the area that needs treatment.

Spontaneous heating may become apparent within six weeks, but is not likely to show after about four months. There have been indications that if opencast coal has heated and the temperature has been reduced by suitable treatment it is not likely to heat again.

The general nature of stored opencast coal, both as to quality and size, makes it likely that deterioration will be negligible.

CHAPTER IV

THE CHEMICAL PRINCIPLES OF COMBUSTION AND GASIFICATION

Combustion and gasification—Chemistry of combustion—Calculation of analysis of flue gas from solid and gaseous fuels and of air requirements—Chemistry of gasification.

COMBUSTION AND GASIFICATION

THE art of *combustion* consists in causing chemical reactions which generate heat to take place between the combustible material (generally carbon, hydrogen or compounds of these elements) and oxygen (generally in the form of air) in such a way that the maximum amount of heat is released. A secondary, but none the less important, branch of the art is to secure the type of flame needed for any particular industrial operation, e.g. an intensely hot flame or a slow, " lazy " flame of lower temperature.

Combustion of coal is brought about by the combination of the carbon (C) and hydrogen (H_2) with the oxygen (O_2) contained in the air. When carbon burns completely, it results in the formation of a gas, carbon dioxide (CO_2). When carbon burns incompletely it forms carbon monoxide (CO) which when compared with carbon dioxide contains only half the amount of oxygen per unit weight of carbon. The combination of hydrogen with oxygen forms water vapour (H_2O).

Air may be considered for combustion purposes as a mixture of 79·1 per cent. of nitrogen with 20·9 per cent. of oxygen by volume. As far as burning is concerned, the oxygen is the only useful part. The nitrogen is an adulterant ; it does not play any part in the combustion and it has to be heated up to the temperature of the other gases by the expenditure of part of the heat generated in the process.

The object of *gasification* is to cause chemical reactions to take place between carbon, oxygen and steam that will give rise to a combustible gas of the highest practicable calorific value.

Whereas the art of combustion is directed to the production of heat, the art of gasification is directed to the conversion of the solid into a gaseous fuel which is ultimately burnt to produce heat in another part of the plant.

Gasification is used where solid fuel is unsuitable or inconvenient for the process for which heat is required.

COMBUSTION

Fuels consist essentially of carbon, hydrogen and oxygen combined to form more or less complex bodies ; the hydrogen may be uncombined and it is present as free hydrogen in most gaseous fuels ; similarly the carbon may exist uncombined as in coke. In combustion the compounds and elements are alike burnt to carbon dioxide and water vapour. In this process heat is developed. Combustion technique is thus concerned primarily with the development of the maximum quantity of heat from the fuel ; it must also be concerned with the quantity of heat that escapes unused from the plant. It is obviously necessary to understand the principles involved and to have means for measuring quantities of heat.

In this chapter the chemistry of combustion and gasification is described, leaving the thermal aspects of the subject to Chapter V.

THE CHEMISTRY OF COMBUSTION

The products of complete combustion are carbon dioxide and water when the

substance burnt is coal, coke, oil, gas or other fuel in normal use which contains as combustibles carbon and hydrogen. Since the combustion is effected with air the carbon dioxide and water are accompanied by a large amount of nitrogen and any excess oxygen that may have been used. The products of combustion are called " flue gas " or " waste gas."

Because most fuels contain small quantities of sulphur the flue gas also comprises oxides of sulphur, mainly the dioxide (SO_2), but with some trioxide (SO_3) in addition.

In the simplest chemical language these changes are expressed by the equations :—

$$C + O_2 = CO_2$$
$$2H_2 + O_2 = 2H_2O$$
$$S + O_2 = SO_2$$

Such equations not only indicate what substances take part in the reaction, but also the proportions in which they take part. Thus the equation,

$$C + O_2 = CO_2$$

means that 1 atom of carbon combines with 2 atoms of oxygen to form 1 molecule of carbon dioxide which therefore contains 3 atoms—1 of carbon and 2 of oxygen. Similarly,

$$2H_2 + O_2 = 2H_2O$$

means that $2 \times 2 = 4$ atoms of hydrogen combine with 2 atoms of oxygen to give 2 molecules of water. The number of atoms of the several elements on each side of the equation must be equal, since matter is indestructible.

Since the atom and molecule of any substance possess definite weights and the relative weights of the atoms of the several elements are known, it follows that these equations show also the weights in which the various substances take part. For example, since the atomic weight of carbon is 12 and that of oxygen is 16,

$$C + O_2 = CO_2$$

also indicates that 12 parts by weight of carbon combine with $2 \times 16 = 32$ parts of oxygen to give $12 + 2 \times 16 = 44$ parts of carbon dioxide. Similarly, since the atomic weight of hydrogen is unity,

$$2H_2 + O_2 = 2H_2O$$

implies that $2 \times 2 = 4$ parts by weight of hydrogen combine with $2 \times 16 = 32$ parts of oxygen to give $2 \times (2 + 16) = 36$ parts by weight of water.

" Parts by weight " may be grams, kilograms, pounds, tons and so forth as desired provided the same units are used throughout.

In reactions involving gases, simple relations by volume also exist between the gases taking part because the molecular weight of gases expressed in the same units of weight occupy the same volume.

The molecular weights (expressed in the nearest whole numbers) of some of the more common gases are :—

hydrogen	2	nitrogen	28
oxygen	32	water vapour ..	18
carbon dioxide.. ..	44	sulphur dioxide ..	64

These weights of gases, dry and expressed in grams, occupy 22·412 litres at 0° C. (32° F.) and 760 mm. bar. (30 inches bar.).* If the weights on the same

* Although 760 mm. is 29·92 inches, it is convenient and sufficiently accurate for most practical purposes to regard 760 mm. bar. as equivalent to 30 inches bar., 60° F. and 30 inches bar. being taken as the standard temperature and pressure for gas measurement in English units and 15° C. (or 0° C.) and 760 mm. for metric units.

basis are expressed in pounds, these several weights of gases all occupy 359 cubic feet, again at 0° C. (32° F.) and 760 mm. bar. (30 inches bar.). The volume occupied at any other temperature and pressure may be calculated from these data, the volume being directly proportional to the absolute temperature and inversely proportional to the pressure.

If a volume of gas V is measured at t_1° C. and p_1 mm. mercury pressure, its volume at t_2° C. and p_2 mm. pressure is

$$V \cdot \frac{t_2 + 273}{t_1 + 273} \cdot \frac{p_1}{p_2}$$

Similarly if the volume is measured at t_1° F. and p_1 inches bar., its volume at t_2° F. and p_2 inches bar. is

$$V \cdot \frac{t_2 + 460}{t_1 + 460} \cdot \frac{p_1}{p_2}$$

In the equation :

$$C + O_2 = CO_2$$

1 volume of oxygen gives rise to 1 volume of CO_2. For the reaction

$$2H_2 + O_2 = 2H_2O$$

2 volumes of hydrogen and 1 volume of oxygen combine to form 2 volumes of steam.

A " volume " may be taken as any desired quantity, such as 1 cubic foot, 1,000 cubic feet, 1 litre, 1 cubic metre, etc., provided that the same units are used throughout.

Since air may be taken to consist for practical purposes of 20·9 per cent. by volume of oxygen (O_2) and 79·1 per cent. by volume of nitrogen (N_2), 1 volume of oxygen in air is accompanied by (79·1/20·9 =) 3·78 volumes of nitrogen, and these equations from the point of view of the combustion engineer are more correctly written :

$$\begin{aligned} C + O_2 + 3{\cdot}78\ N_2 &= CO_2 + 3{\cdot}78\ N_2 \\ 2H_2 + O_2 + 3{\cdot}78\ N_2 &= 2H_2O + 3{\cdot}78\ N_2 \\ S + O_2 + 3{\cdot}78\ N_2 &= SO_2 + 3{\cdot}78\ N_2 \end{aligned}$$

From these chemical relationships it is possible to calculate the composition of the flue gases from a coal or coke or oil. To assist in making the calculation, Table 14 has been derived from the simple chemical equations just discussed, it being assumed throughout that the steam (H_2O) produced from combustion of the hydrogen remains uncondensed as a constituent of the gases.

TABLE 14

1 lb. of	Requires lb.			Yields lb.				Yields cu. ft. Measured at 60° F. and 30″ bar.			
	O_2	N_2	Air	CO_2	H_2O	SO_2	N_2	CO_2	H_2O	SO_2	N_2
C to CO_2	2·667	8·833	11·500	3·667	—	—	8·833	31·4	—	—	119
H_2 to H_2O	7·939	26·287	34·226	—	8·939	—	26·287	—	188	—	353
S to SO_2	1·000	3·305	4·305	—	—	2·0	3·305	—	—	11·5	44

Density—lb./cu. ft. at 60° F. and 30″ dry						Specific volume—cu. ft. at 60° F. and 30″ per lb.					
CO_2	0·117	SO_2	0·173	H_2O	0·0476	CO_2	8·55	SO_2	5·78	H_2O	21·00
N_2	0·0744	O_2	0·0846	Air	0·0765	N_2	13·43	O_2	11·82	Air	13·07

The oxygen in the coal is combined with carbon, hydrogen and other elements. It is assumed for the purpose of calculation that this oxygen is combined wholly with hydrogen. An allowance of hydrogen equivalent to the oxygen present must be deducted from the effective weight of the hydrogen. This deduction is equal to one-eighth of the oxygen content, since 1 lb. of hydrogen combines with 8 lb. of oxygen to form 9 lb. of water.

CALCULATION OF THE COMPOSITION OF FLUE GASES FROM COAL

To show how Table 14 may be used, and as the simplest method of illustrating the procedure, the composition of the flue gas (assuming that no excess air is used) is here calculated from a coal having the following composition :—

	Per cent.
Carbon	74
Hydrogen	5
Oxygen	5
Nitrogen	1
Sulphur	1
Moisture	9
Ash	5
	100

Water consists of 1 part of hydrogen to 8 parts of oxygen by weight. The first step is to assume that the 5 per cent. of oxygen in the coal is combined with one-eighth of its weight of hydrogen to form water as just indicated.

The water formed in this way is derived from the hydrogen that has combined with this oxygen (i.e. $\frac{5}{8}$ per cent. of the coal) plus the oxygen contained in the coal (5 per cent.). With 9 per cent. of water already existing in the coal, the water liberated as steam is :

$$9 + 5 + \tfrac{5}{8} = 14{\cdot}6 \text{ per cent. of the coal by weight.}$$

The remaining hydrogen (the " available hydrogen " as it is termed) is thus $5 - \frac{5}{8} = 4{\cdot}4$ per cent.

This 4·4 per cent. of available hydrogen will require oxygen from the air for its combustion, and will yield a further quantity of steam.

Thus from 1 lb. of coal there is derived :—

Incombustibles :—

	Per cent.	lb. per lb. of coal
Nitrogen	1	.. 0·01
Ash	5	.. 0·05
Water	14·6	.. 0·146

Combustibles requiring air for their combustion :—

Carbon	74	.. 0·74
Hydrogen	4·4	.. 0·044
Sulphur	1·0	.. 0·01

The next step is to consider the quantity of each element present in 1 lb. of the coal and to determine the quantity in cubic feet of the gases to which it gives rise, using the factors given in the preceding table. Due account must be taken of the nitrogen which accompanies the oxygen in the air required for combustion, and this is done in Table 14. The calculation, using the data given in Table 14, is as follows :—

Weight of constituent		Volume of products of combustion— cu. ft.			
	lb.	CO_2	H_2O	SO_2	N_2
C	0·74	23·3	—	—	88
H	0·044	—	8·27	—	15·5
S	0·01	—	—	0·11	0·44
H_2O	0·146	—	3·06	—	—
N	0·01	—	—	—	0·13
Ash	0·05	—	—	—	—
	1·000	23·3	11·33	0·11	104·07

At higher temperatures all these four gases are contained in the products of combustion ; if the gases are cooled to atmospheric temperature, as they would be if they were to be analysed by gas analysis apparatus, the bulk of the water vapour would condense. The following figures represent the difference between the gases in the flues and in the gas analysis apparatus :—

Total wet gases if calculated to 60° F. 138·81 cubic feet per lb. coal
,, dry ,, ,, ,, ,, 127·48 cubic feet per lb. coal

Composition by volume of flue gas (the coal being burnt with the exact theoretical amount of air) is obtained, for example, thus :—

$$CO_2 \text{ on wet gas} = \frac{23·3}{138·81} \times 100 = 16·7 \text{ per cent. by volume}$$

and

$$CO_2 \text{ on dry gas} = \frac{23·3}{127·48} \times 100 = 18·3 \text{ per cent. by volume.}$$

The complete analysis is :—

	On wet gas Per cent.	On dry gas Per cent.
CO_2	16·7	18·3
SO_2	0·1	0·1
H_2O	8·2	—
N_2	75·0	81·6

If excess of air is used, its volume must be added to the total as here calculated. The nitrogen in the air supply was 104·07 − 0·13 = 103·94 cubic feet per lb. of coal. If 50 per cent. of excess air is used in the example given, the air used for combustion is :—

103·94 × 100/79·1, or 131·4 cubic feet (= N_2 expressed as air)
Add to this 50 per cent. of 131·4, i.e. 65·7 cubic feet (= excess air)

Total air used per lb. of coal = 197·1 cubic feet

Total waste gases (wet) = 139·79 + 65·7 = 205·49
(dry) = 127·48 + 65·7 = 193·18

The 65·7 cubic feet of excess air comprises 52 cubic feet of nitrogen and 13·7 cubic feet of oxygen.

The complete analysis is then :—

	Cu. ft. from 1 lb. coal		Composition	
	Wet gases	Dry gases	Wet Per cent.	Dry Per cent.
CO_2	23·3	23·3	11·4	12·0
SO_2	0·11	0·11	0·1	0·1
H_2O	11·33	—	5·6	—
N_2	156·07	156·07	76·2	80·8
O_2	13·7	13·7	6·7	7·1
	204·51	193·18	100·0	100·0

COMBUSTION DATA FOR FUELS

Proceeding in the manner indicated for a number of fuels, Table 15 can be derived on the assumption (already made in the preceding calculation) that all the fuels are burnt with the exact amount of air required for combustion.

TABLE 15

	Theoretical amount of air required per lb. fuel		Vol. of total waste gases cu. ft. at 60° F. and 30″ bar.	Calculated wet waste gas analysis			CO_2 as determined by analysis i.e.with the water in the gas condensed
	lb.	cu. ft. at 60° F. and 30″ bar.		CO_2 Per cent.	N_2 Per cent.	H_2O Per cent.	Per cent.
Non-coking coal 40% V.M.; 5% ash; 10% moisture ..	8·68	113	122	17·2	73·8	9·0	18·9
Coking coal 30% V.M.; 5% ash; 2% moisture	10·75	140	147	17·3	75·8	6·9	18·6
Low V.M. coal 18% V.M.; 4% ash; 1% moisture ..	11·28	148	152	18·0	76·6	5·4	19·0
Anthracite 4% V.M.; 3% ash; 1% moisture	11·34	150	152	18·9	77·4	3·7	19·6
Coke 1% V.M.; 7% ash; 2% moisture	10·22	134	135	20·2	78·2	1·6	20·5
Creosote and pitch	11·65	154	160	18·3	76·1	4·6	19·2
Fuel oil (C 86%; H 12%)..	14·0	184	194	14·1	74·1	11·4	15·9
Coal gas (560 B.Th.U.) .. (35 cu. ft. coal gas/lb.)	13·1	172	197	8·7	69·8	21·5	11·1

CALCULATION OF THE COMPOSITION OF THE FLUE GASES FROM GASEOUS FUELS

The calculation of the composition of flue gases from a gas such as coal gas can be made on similar lines, but the fundamental combustion equations are more complicated and are utilised on a volume basis only. As an example, the combustion of methane is expressed by the equation :—

$$CH_4 + 2O_2 + 7{\cdot}56\,N_2 = CO_2 + 2H_2O + 7{\cdot}56\,N_2.$$

The interpretation of this equation in terms of waste gas analysis, as will be evident from previous explanations, is that when 1 cubic foot of methane is burned with air, it requires 2 cubic feet of oxygen (and consequently 9·56 cubic feet of air) and it forms as waste gas, 1 cubic foot of carbon dioxide, 2 cubic feet of water vapour and 7·56 cubic feet of nitrogen, all assumed to be measured at the same temperature.

Particulars of the combustion of some industrial gases are contained in Table 16.

TABLE 16

	Coal gas	Producer gas		Blast furnace gas
C.V.—B.Th.U./ cu. ft. gross ..	475	163	132	92
Cu. ft. air required for 1 cu. ft. gas (theoretical)..	4·06	1·28	1·00	0·69
Products of combustion from 1 cu. ft. of gas :	cu. ft.	cu. ft.	cu. ft.	cu. ft.
CO_2	0·50	0·36	0·34	0·38
H_2O	0·97	0·18	0·12	0·02
N_2	3·27	1·53	1·34	1·14
Total	4·74	2·07	1·80	1·54
Theoretical CO_2 content as analysed, i.e. on dry gas, per cent.	13·3	19·2	20·5	24·9

METHOD OF CALCULATION

Taking as an example the analysis of horizontal retort gas from Chapter II (Table 8), the combustion reactions and their resultant products can be summarised as in Table 17, noting that each cubic foot of oxygen required is accompanied by 3·78 cubic feet of nitrogen.

TABLE 17

Constituent	Combustion reaction	Cu. ft. per cu. ft. of constituent			
		O_2 req.	N_2 (from air)	CO_2 formed	H_2O formed
Oxygen—O_2	Assists in burning the combustible	—	—	—	—
Carbon dioxide—CO_2	None	—	—	—	—
Hydrocarbons—C_mH_n taken as C_3H_6	$2C_3H_6 + 9O_2 = 6CO_2 + 6H_2O$	4·5	17·01	3·0	3·0
Carbon monoxide—CO	$2CO + O_2 = 2CO_2$	0·5	1·89	1·0	—
Hydrogen—H_2 ..	$2H_2 + O_2 = 2H_2O$	0·5	1·89	—	1·0
Methane—CH_4 ..	$CH_4 + 2O_2 = CO_2 + 2H_2O$	2·0	7·56	1·0	2·0
Nitrogen—N_2.. ..	None	—	—	—	—

The calculation in cu. ft./100 cu. ft. of gas then proceeds as follows :—

Constituent	Per cent. by vol.	O_2 required for combustion cu. ft.	Products cu. ft.		
			CO_2	H_2O	N_2
O_2	0·4	−0·4	—	—	− 1·5
CO_2	2·0	—	2·0	—	—
C_mH_n taken as C_3H_6	3·6	16·2	10·8	10·8	61·3
CO	8·0	4·0	8·0	—	15·1
H_2	52·0	26·0	—	52·0	98·3
CH_4	30·0	60·0	30·0	60·0	226·8
N_2	4·0	—	—	—	4·0
Volume per 100 cu. ft. of town gas .. .. .. cu. ft.		105·8	50·8	122·8	404·0

The oxygen is provided from air ; thus to provide 105·8 volumes of oxygen

$$105{\cdot}8 \times \frac{100}{20{\cdot}9} = 506{\cdot}2 \text{ volumes of air are required.}$$

In other words, the air required for the combustion of 100 volumes of gas is 506·2 volumes.

Thus 1 cubic foot of gas requires theoretically for combustion 5·062 cubic feet of air.

The products from 100 cubic feet of gas and 506·2 cubic feet of air will be :—

	Volume calc. at 60° F. and 30″ bar., the water assumed to be uncondensed. cu. ft.	Per cent. by volume.
CO_2	50·8	8·8
H_2O (steam)	122·8	21·3
N_2	404·0	69·9
Total	577·6	100·0

The volume of products formed when 1 cubic foot of this gas is completely burnt with 5·062 cubic feet of air is therefore 5·776 cubic feet at 60° F. and 30 inches bar.

If the gases are cooled below the dew point water vapour will condense and the volume will be less. The products for purposes of analysis are therefore generally expressed on the dry basis without the water vapour.

The volume of products from 100 cubic feet of gas is therefore on the dry basis :—

CO_2	50·8	cubic feet
N_2	404·0	,,
Total	454·8	,,

The volume of products of combustion (dry) from 1 cubic foot of this sample of town gas with 5·062 cubic feet of air is therefore 4·548 cubic feet.

The percentage of CO_2 as analysed from this gas is given by :—

$$\frac{50{\cdot}8 \times 100}{454{\cdot}8} = 11{\cdot}2 \text{ per cent.}$$

The calculation can be extended to include excess air in the same way as for coal in the example previously given.

THE CHEMISTRY OF GASIFICATION

A fuel bed may be shallow or deep. If shallow, the carbon tends to burn wholly to carbon dioxide. The carbon dioxide first formed, or water vapour present in the incoming air, may be brought into contact with red-hot carbon, in which event certain further reactions occur, to promote which is the function of gasification. A deep fuel bed naturally offers the optimum conditions for these further reactions, though they may occur to a lesser extent in the shallow fuel bed of the boiler furnace. A deep fuel bed of the order of 3 feet or more is used for gasification for the manufacture of producer gas.

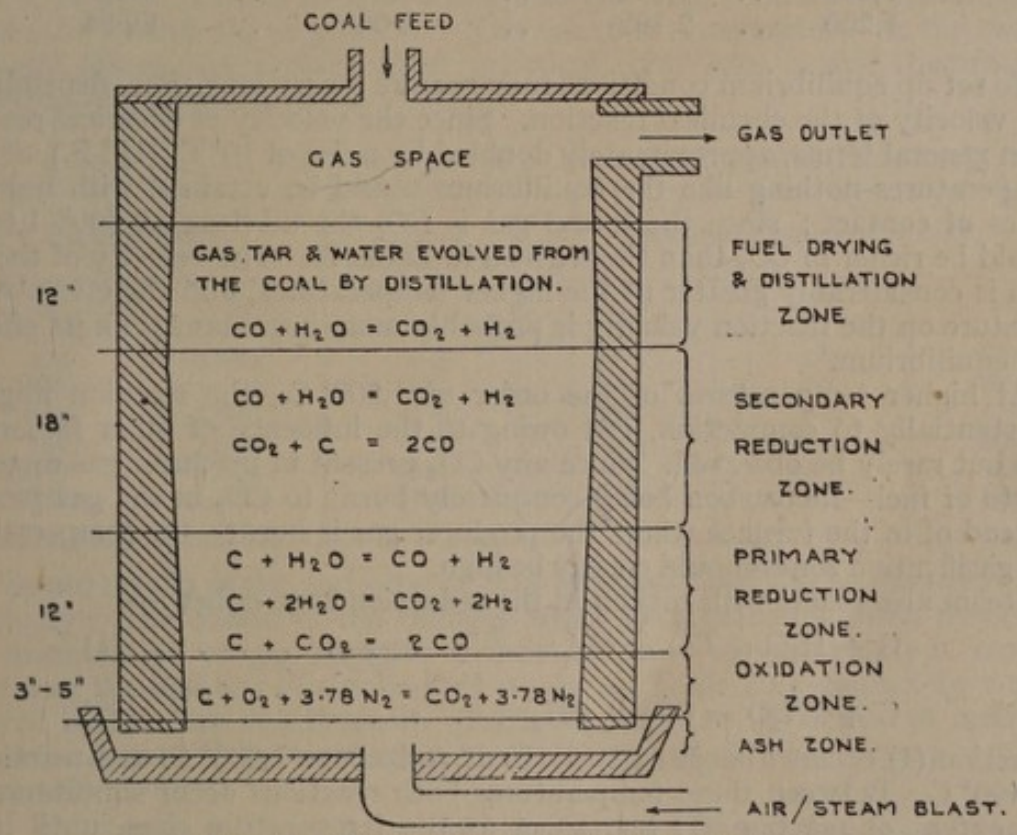

FIG. 13. Chemical reactions in a deep fuel bed.

It is convenient to consider these reactions as confined to several zones (Fig. 13) though they overlap into the zones above and below. The gases passing into the base of the fuel bed may comprise air and steam and when reacting in the fuel bed give rise to chemical and thermal effects of great importance. The significance of the thermal reactions here referred to will be explained in Chapter V.

The ash zone at the base of the fuel bed serves to protect the grate from the intense heat and to distribute the air and steam over the bed.

Above this is the oxidation zone at which the reaction $C + O_2 = CO_2$ takes place, free oxygen disappearing about 4 or 5 inches above the top of the layer of ash. This reaction generates heat (see Chapter V) and provides practically the whole of the heat available in the fuel bed that is required for the subsequent gasification reactions.

The CO_2, accompanied by nitrogen and steam, travels upwards into the reducing zone. Here a reaction between CO_2 and carbon takes place resulting in the production of the combustible gas, carbon monoxide :—

$$CO_2 + C = 2CO$$

This reaction absorbs heat. The extent to which it occurs depends on the temperature and on the time available. Equilibrium values which are only attained by sustained contact in the absence of other disturbing factors, are given in Table 18.

TABLE 18

Temperature		Composition of gas in contact with carbon at equilibrium	
° C.	° F.	CO_2—per cent.	CO—per cent.
600	1,112	68·5	31·5
700	1,292	36·6	63·4
800	1,472	13·7	86·3
1,000	1,832	0·6	99·4
1,200	2,192	0·06	99·94

To set up equilibrium conditions may require a very long time, depending on the velocity of the chemical reaction. Since the velocity of chemical reactions is, in general terms, approximately doubled by a rise of 10° C. (18° F.), at lower temperatures nothing like the equilibrium would be attained with industrial times of contact; since the initial gas is CO_2 the mixture at 600°–1,000° C. would be richer in CO_2 than those given in Table 18. The velocity of the reaction is considerably greater at the higher temperatures, and the effect of temperature on the reaction velocity is probably more important than its effect on the equilibrium.

At higher temperatures of the order of 1,200° C., the reaction might go substantially to completion, but owing to the influence of other factors this can but rarely be observed. Since any CO_2 present in producer gas represents waste of fuel—the carbon being completely burnt to CO_2 in the gas producer instead of in the furnace where the producer gas is burnt—the temperature in the gasification zones should clearly be high.

Steam also reacts with carbon in the reduction zone either :—

$$C + H_2O = CO + H_2 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

or

$$C + 2H_2O = CO_2 + 2H_2 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

Reaction (1) occurs almost exclusively at and above 1,000° C. and reaction (2) at 600° C. Between these temperatures both reactions occur simultaneously, the extent of reaction (1) increasing as the temperature rises, until it predominates. To avoid the production of CO_2 in the producer, the temperature in the reduction zone should clearly be high for gasification. Both of these reactions absorb heat.

The importance of a high temperature in the reduction zone is thus clearly shown. The necessary heat is provided wholly by the combustion reaction in the oxidation zone, and this heat has (*a*) to heat up the fuel in the reduction zones to the necessary temperature, and (*b*) to compensate for the absorption of heat caused by the reactions in the reduction zone.

Higher still in the fuel bed, the reduction reaction $CO_2 + C = 2CO$ and the reactions between carbon and steam still continue slowly and are now accompanied by a reaction between CO and still undecomposed steam, known as the water gas equilibrium reaction :—

$$CO + H_2O = CO_2 + H_2.$$

This reaction persists until the temperature drops to a dull red heat and is responsible for the production of some additional CO_2 in the producer gas.

All the reactions are equilibrium reactions, and under suitable conditions may proceed in the reverse direction to those here indicated. At any tem-

perature there is an equilibrium composition of the substances taking part in the reaction which is the more closely approached as time and temperatures are adequate ; an example was given when dealing with the reaction $C + CO_2 = 2CO$.

At the top of the fuel bed, the fresh fuel (if coal is used) is distilled, permanent gases, water and tar being added to the gases leaving the fuel bed. Any water contained in the incoming fuel is evaporated at this stage.

These reactions are summarised in Fig. 13 ; their thermal effects are summarised in Fig. 17.

The quantity of steam used has an important influence on the reactions. This will be discussed in detail in the chapter on gas producers (Chapter XVII). It is obviously important that as much of the steam as possible shall be converted into gas, and in its conversion into gas shall produce as little CO_2 and as much CO and H_2 as possible. The effect of temperature on the reaction between steam and carbon under practical conditions is well illustrated by experimental figures given in Table 19.

TABLE. 19. THE ACTION OF STEAM ON CARBON (HARRIES)

Temperature		Steam decomposed Per cent.	Analysis of gases—Per cent.		
° C.	° F.		CO_2	CO	H_2
674	1,245	8·8	29·8	4·9	65·2
758	1,396	25·3	27·0	7·8	65·2
838	1,540	41·0	22·9	15·1	61·9
954	1,750	70·2	6·8	39·3	53·5
1,010	1,850	94·0	1·5	49·7	48·8
1,125	2,057	99·4	0·6	48·5	50·9

The net results of the various reactions here described can be followed experimentally. The course of the changes which take place has been determined by examination of gas samples withdrawn from closely adjacent sampling points in the fuel bed of a coke-fired producer, Table 20 (adapted from results quoted by Haslam and Russell). Table 30 (Chapter VI) is also of interest in this connection, though referring to the action of air only, steam not being admitted.

TABLE 20. COMPOSITION OF GASES IN FUEL BED OF A GAS PRODUCER

Zone	Height of sampling point above grate in.	Composition of gases—Per cent. by vol.				
		O_2	CO_2	CO	H_2	H_2O
Ash	5	18·4	—	—	—	13·4
Oxidation ..	7·5	nil	17·6	2·8	—	13·1
Primary reduction	10	—	11·5	12·2	—	12·2
,, ,,	15	—	7·2	20·4	4·0	8·1
,, ,,	20	—	5·9	23·3	7·0	4·7
Secondary reduction	30	—	4·6	25·6	8·2	3·5
Top of fuel bed ..	40	—	4·7	27·0	8·5	2·7

The final results of the gasification process are given in Chapter II, Table 9, under producer gas. If, as in some older producers, air only is blown through the fuel bed, the resulting gas is similar to blast furnace gas, though containing less CO_2 and proportionately more CO.

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CHAPTER V

HEATING EFFECTS IN COMBUSTION AND GASIFICATION

The measurement of heat—Specific heat—Excess air for combustion and its effects—Heat losses in flue gases—Thermal effects in gasification—Ignition temperatures—Ignition limits—Flame speed—Flame temperatures.

THE MEASUREMENT OF HEAT

QUANTITY of heat is defined as, and measured by, the amount of heat that is required to raise a given weight of water through a given temperature range.

On the pound-Fahrenheit scale heat is measured in terms of British Thermal Units (B.Th.U.). One B.Th.U. is the quantity of heat required to raise 1 lb. of water through 1° F. Since the heat capacity of water varies slightly at different temperatures the more accurate definition of the heat unit is the amount of heat required to raise 1 lb. of water from 60° to 61° F. Another definition is the " mean B.Th.U." which is $\frac{1}{180}$ of the sensible heat required to raise 1 lb. of water at 32° F. to 212° F. without vaporisation.

On the kilogram-centigrade scale 1 large calorie (known as 1 kg. cal.) is the amount of heat required to raise 1 kilogram of water from 15° C. to 16° C. This is often written " K.C.U."

The pound-centigrade unit sometimes used is the amount of heat required to raise 1 lb. of water 1° C. It is denoted " C.H.U."

The relationship between these units is as follows :—

1 kg. cal. = 3·968 B.Th.U.
1 C.H.U. = 1·8 B.Th.U.
1 B.Th.U. = 0·252 kg. cal.
= 0·555 C.H.U.

Thus if the calorific value of a coal is given as 14,000 B.Th.U. per lb. this means that if the whole of the heat generated by the combustion of 1 lb. of the coal under consideration were usefully employed in heating water, it would serve to raise 14,000 lb. of water through 1° F., or alternatively (for example) 140 lb. of water through 100° F. Thus,

lb. of water heated × °F. temperature rise = B.Th.U. given to the water.

Heat, work and energy are interchangeable, and it is useful to note here certain other conversion factors, all in terms of British Thermal Units.

Unit.	Equivalent B.Th.U
1 foot-pound	0·001285
778 foot-pounds	1·0
1 therm	100,000
1 kilowatt-hour (kWh)	3,413
1 horse-power hour (H.P.H.)	2,546

PHYSICAL CONSTANTS

Figures of physical constants required in fuel technology can be found in " Technical Data on Fuel," edited by H. M. Spiers and published by the British National Committee of the World Power Conference. Only such figures are published here as are required for an explanation of the subject.

DATUM CONDITIONS

In making thermal calculations it is necessary to select a basis temperature from which to start. For steam tables (Chapter VII) this temperature is 32° F. (0° C.) on the ground that below this temperature water solidifies and 32° F. is the lowest temperature at which water exists in the liquid state.

For most combustion calculations 60° F. is used and this temperature will be used in this book as the basis for this purpose.

SPECIFIC HEAT

It is found experimentally that the quantity of heat required to raise 1 lb. of solid, liquid and gaseous substances through a given temperature range is not the same for each substance. Each substance is said to have its own particular specific heat, and this is defined with reference to water as the standard substance. Thus the specific heat of a substance is the number of B.Th.U. required to raise 1 lb. of the substance through 1° F., or alternatively the number of kg. cal. required to raise 1 kilogram through 1° C. The actual figure will be the same whichever system is used. By the definition of a heat unit, the specific heat of water at 15° C. or 60° F. is taken as 1.

The specific heats of solids and liquids are always given on a weight basis. The specific heat of a gas may be given on a weight basis, but for the purpose of calculation it is usually more convenient to give it on a volume basis. The volume basis will be adopted in this book. The specific heat of a gas, therefore, is expressed as the quantity of heat required to raise 1 cubic foot of the gas through 1° F. ; or on the metric system, the quantity of heat required to raise 1 cubic metre of the gas through 1° C. ; the figures on the two systems are different. It is, of course, necessary to state the temperature and pressure of the gas when quoting a specific heat on the volume basis.

The value of the specific heat of a substance varies to some extent with the temperature. Among examples of specific heats may be cited those in Table 21, which are given solely to illustrate the wide differences existing in this respect between various substances.

All gases have two specific heats, dependent upon whether they are being heated while keeping the pressure constant or the volume constant. Heating at constant pressure is far commoner than heating with constant volume ; specific heats at constant volume are not generally needed in boiler and furnace combustion practice and are not therefore discussed here.

TABLE 21

Mean specific heats on a weight basis :—

	Specific heat	Over the temperature range	
		° F.	° C.
Water	1·00	60–61	15·6–16·1
Aluminium	0·216	32–212	0–100
Copper	0·102	32–600	0–315
Iron	0·128	32–750	0–399
Firebrick	0·23	32–1,800	0–982
Coal	0·25	32–300	0–149
Coke	0·26	80–750	26·7–399
Steam (at constant pressure)	0·45–0·52	212–600	100–315
Air (constant pressure)	0·24	32–500	0–260
Flue gases of average composition, dry (constant pressure)	0·24	212–600	100–315

Specific heats (on a volume basis) at a temperature of 0° C. (32° F.) unless otherwise stated, at constant pressure :—

	B.Th.U./cu. ft./° F. for gas measured at 60° F. and 30″	Kg. cal./cu. m./° C. for gas measured at 0° C. and 760 mm.
Air (dry)	·0183	0·310
Carbon monoxide and nitrogen (dry) ..	·0184	0·311
Hydrogen (dry)	·0181	0·306
Carbon dioxide (dry)	·0226	0·384
Water vapour	·0210	0·356
Mixed products of combustion (typical figures):— at 0° C.	·0191	0·324
Do. do. do. 0–300° C.	·0194	0·329
Do. do. do. 0–1,000° C.	·0207	0·352
Do. do. do. 0–1,500° C.	·0214	0·363

To illustrate the use of specific heat by examples, suppose it is required to calculate the heat required to heat 1 cwt. of firebrick from 60° F. to 1,780° F.

$$\begin{aligned}\text{Heat required} &= 112 \times (1{,}780–60) \times 0{\cdot}23\\ &= \text{lb.} \times \text{temp. rise} \times \text{sp. ht.}\\ &= 44{\cdot}31 \text{ B.Th.U.}\end{aligned}$$

Similarly, if it be required to calculate how much heat would be obtained if 750 cubic feet (measured at 60° F. and 30 inches) of products of combustion were cooled from 300° F. to 220° F., the figures would be :—

$$\begin{aligned}&750 \times (300 - 220) \times 0{\cdot}0192\\ &\text{cubic feet} \times \text{temp. fall} \times \text{sp. ht.}\\ &= 1{,}152 \text{ B.Th.U.}\end{aligned}$$

Thus, whilst for water,

$$\text{lb. weight} \times \text{temp. rise in °F.} = \text{B.Th.U.},$$

for all other materials,

$$\text{lb. weight} \times \text{temp. rise in °F.} \times \text{sp. ht. by weight} = \text{B.Th.U.}$$

and for gases, if using volumes,

$$\text{volume in cubic feet} \times \text{temp. rise in °F.} \times \text{sp. ht. by volume} = \text{B.Th.U.}$$

In accurate work it is necessary to take account of the variation of the specific heat with temperature and to use the mean specific heats between the temperatures involved. An example of this is given above when dealing with the specific heat of mixed products of combustion. Thus it may be required to calculate the heat above 60° F. remaining in 200 cubic feet of flue gases measured at 60° F. and 30 inches (*a*) at 575° F. (301° C.) and (*b*) 1,835° F. (1,002° C.). The volume of flue gas obtained from the combustion of 1 lb. of coal under good conditions in a boiler plant is approximately 200 cubic feet measured at 60° F.

(*a*) Gases at 575° F.

$$\begin{aligned}\text{Heat content} &= 200 \times (575 - 60) \times 0{\cdot}0194\\ &\text{cubic feet} \times \text{temp. rise} \times \text{sp. ht.}\\ &= 1{,}998 \text{ B.Th.U.}\end{aligned}$$

(*b*) Gases at 1,835° F.

$$\begin{aligned}\text{Heat content} &= 200 \times (1{,}835 - 60) \times 0{\cdot}0207\\ &\text{cubic feet} \times \text{temp. rise} \times \text{sp. ht.}\\ &= 7{,}348 \text{ B.Th.U.}\end{aligned}$$

EXCESS AIR REQUIRED FOR COMBUSTION

When dealing with the chemistry of combustion, all calculations were made on the assumption that the fuels were burnt completely with the theoretical amount of air. If more than this amount of air is used, the foregoing calculations can be readily adjusted by adding the additional oxygen (20·9 per cent. of the extra volume of air) and nitrogen (79·1 per cent.) to the gases derived from combustion.

In practice complete combustion cannot be secured unless more than the theoretical amount of air is used. When burning solid fuel this is partly because of the difficulty of mixing the air and combustible volatiles above the fuel bed. It is also a consequence of the necessity of completing the combustion within the combustion space. Physical chemistry has shown that the speed of a chemical reaction is proportional to the concentration of the reactants. As the concentration of oxygen and combustible in the combustion space decreases from front to back of the combustion zone, it is necessary to add an excess of oxygen to keep the reaction rate high enough to complete the reaction before the gases leave the furnace.

The velocity of combustion depends upon the rate at which oxygen can reach the fuel. Scrubbing action due to the velocity of the gases promotes the interchange of O_2 and CO_2 with solid fuel surfaces and accelerates the combustion ; this is indicated by the rapidity with which coal burns at a thin spot or over the end of a tuyere where the air velocity is great.

HEAT LOSSES THROUGH EXCESS AIR IN FLUE GASES

It is important to recognise the effect of excess air in combustion. When gases are hot they possess what is known as " sensible heat," the amount of which (as has just been shown) is given in B.Th.U. by the product :—

Vol. of gases in cubic feet measured at 60° F. × specific heat in B.Th.U. per cubic foot × °F. rise in temperature.

The quantity of heat they contain is thus dependent on their specific heat and on their temperature, and, of course, it is directly proportional to their volume measured at standard temperature and pressure. When excess air is used the specific heat of the flue gases is not appreciably affected. The primary effect is to increase the volume of gases and thus directly to increase the loss of heat (see Table 22).

In the measurement of flue gas temperature, sometimes serious errors can be caused if precautions are not taken to shield the thermometer or thermocouple from the radiation effect of the surrounding flue walls or of cooler materials around it, e.g. economiser tubes. This effect is discussed in Chapter XXX.

LATENT HEAT IN FLUE GASES

In addition to their sensible heat, flue gases contain also latent heat in the latent heat of condensation of the steam resulting from combustion or evaporated from the fuel. The subject of latent heat is dealt with in Chapter VII. It is there shown that the latent heat of steam depends on the pressure at which it is generated or condensed. For condensation from flue gases, it may be taken as 1,055 B.Th.U. per lb. or 49·35 B.Th.U. per cubic foot of water vapour. This latent heat, which must be added to the sensible heat, is thus given by

(*a*) Wt. of steam (in lb.) × 1,055 B.Th.U.

or

(*b*) Vol. of steam (in cubic feet) × 49·35 B.Th.U.

All gas and water vapour volumes are taken for the purpose of these calculations as being at 60° F. and 30 inches barometric pressure.

EFFECT OF EXCESS AIR ON CO_2 CONTENT OF FLUE GASES

The dilution with air proportionately decreases the CO_2 content of the waste gases, because the same volume of CO_2 is contained in a greater volume of other gases. Conversely, it is clearly possible from a knowledge of the CO_2 content and the temperature of the gases to deduce the quantity of heat carried away by hot products of combustion.

TOTAL HEAT LOSSES IN FLUE GAS

The heat losses in flue gas are thus seen to be made up of several quantities, some of which are controllable and others outside our control.

(*a*) Sensible heat of gases derived from combustion.
(*b*) Sensible heat of excess air.
(*c*) Latent heat of steam arising from moisture in fuel.
(*d*) Latent heat of steam arising from combustion of the hydrogen in the fuel.
(*e*) Sensible heat of steam arising from moisture in fuel.
(*f*) Sensible heat of steam from hydrogen in fuel.

To the extent that the outlet temperature of the gases from the plant can be controlled through more efficient operation or through the installation of more efficient apparatus (e.g. economisers in a boiler plant, or regenerators or waste heat boilers in a furnace) all of the foregoing sources of loss are partly controllable, except (*c*) and (*d*). (*c*) is controllable if the moisture content of the fuel can be reduced. No control can be exercised over (*d*). The maximum of control can be exercised over (*b*).

To indicate the magnitude of these losses, the same coal will be considered that was the subject of the calculation of flue gas analysis in Chapter IV. This coal contained 9 per cent. of moisture and 5 per cent. of hydrogen. Neglecting SO_2, the composition of the flue gases with 50 per cent. excess air, and taking all volumes at 60° F., was :—

	Cu. ft./1 lb. coal	
CO_2	23·3	127·4
N_2	104·1	
Excess air	65·7	
Steam from moisture in coal ..	1·9	assumed uncondensed.
Steam from hydrogen in coal ..	9·4	
	204·4	

If the gases are leaving the system at 560° F., the following calculations may be made to indicate the relative effects of the losses (*a*) to (*f*).

(*a*) Volume of gases ($CO_2 + N_2$) .. .. .. 127·4 cubic feet
Sp. ht. over the range 560°–60° F. .. .. 0·0195
Heat in gases = 127·4 × 0·0195 × (560–60)
= 1,242 B.Th.U./lb. coal.

(*b*) Volume of excess air .. .. 65·7 cubic feet
Sp. ht. over the range 560°–60° F. .. .. 0·0196
Heat in excess air = 65·7 × 0·0196 × (560–60)
= 644 B.Th.U./lb. coal

(*c*) Volume of steam from moisture in fuel (calculated to 60° F. and 30 in. bar.) .. .. 1·9 cubic feet
Latent heat per cubic foot .. .. .. 49·35 B.Th.U.
Total latent heat = 1·9 × 49·35
= 93·8 B.Th.U./lb. coal

(*d*) Volume of steam from combustion of hydrogen in fuel (calc. to 60° F. and 30 in. bar.) .. 9·4 cubic feet

Total latent heat = 9·4 × 49·35
= 463 B.Th.U./lb. coal

(*e*) and (*f*) Total heat of superheated steam at 560° F. at the pressure at which it exists in the flue gas, i.e. (14·7 × 11·3/205·4 =) 0·8 lb. per sq. in. absolute (see Chapter VII), taken from steam tables .. .. .. 1,317 B.Th.U. per lb.

Since steam tables are based on 32° F. and the flue gas calculations are based on 60° F., from this must be deducted the heat required for heating the condensed water from 32° to 60° F. .. .. 28 B.Th.U. per lb.

Total heat of water vapour above 60° F. 1,289 do.

To obtain the sensible heat (or superheat) in the steam, the latent heat must be deducted also; this is .. .. 1,055 do.

The sensible heat per lb. of steam is thus .. 234 B.Th.U.

Since (Table 14) water vapour at 60° F. weighs 0·0476 lb./cu. ft., the sensible heat per cubic foot is 234 × 0·0476 = ... 11·1 B.Th.U.

Thus, for (*e*) sensible heat of steam = 1·9 × 11·1 = 21·1 B.Th.U./lb. coal.

for (*f*) sensible heat of steam = 9·4 × 11·1 = 104·3 B.Th.U./lb. coal.

These figures may be summarised thus, the coal being taken to have a calorific value of 14,000 B.Th.U./lb. :—

	B.Th.U./lb. of coal burnt	Per cent. of heat in coal
(*a*) Sensible heat of gases derived from combustion ..	1,242	8·87
(*b*) Sensible heat of excess air	644	4·60
(*c*) Latent heat of steam from moisture in fuel ..	93·8	0·67
(*d*) Latent heat of steam from hydrogen in fuel ..	463	3·30
(*e*) Sensible heat of steam from moisture in fuel ..	21·1	0·15
(*f*) Sensible heat of steam from hydrogen in fuel ..	104·3	0·75
	2,568·2	18·34

Since (*d*) cannot be controlled, this calculation indicates that (*a*) and (*b*) are of much greater importance in promoting fuel economy than any other factors. Hence the importance of

(1) Operating with the lowest amount of excess air that will avoid still greater losses due to unburnt combustible gases.
(2) Operating with the lowest practicable outlet temperature from the system.

The result of a calculation of this character is shown in Fig. 14 for a bituminous coal and in Fig. 15 for coal gas.

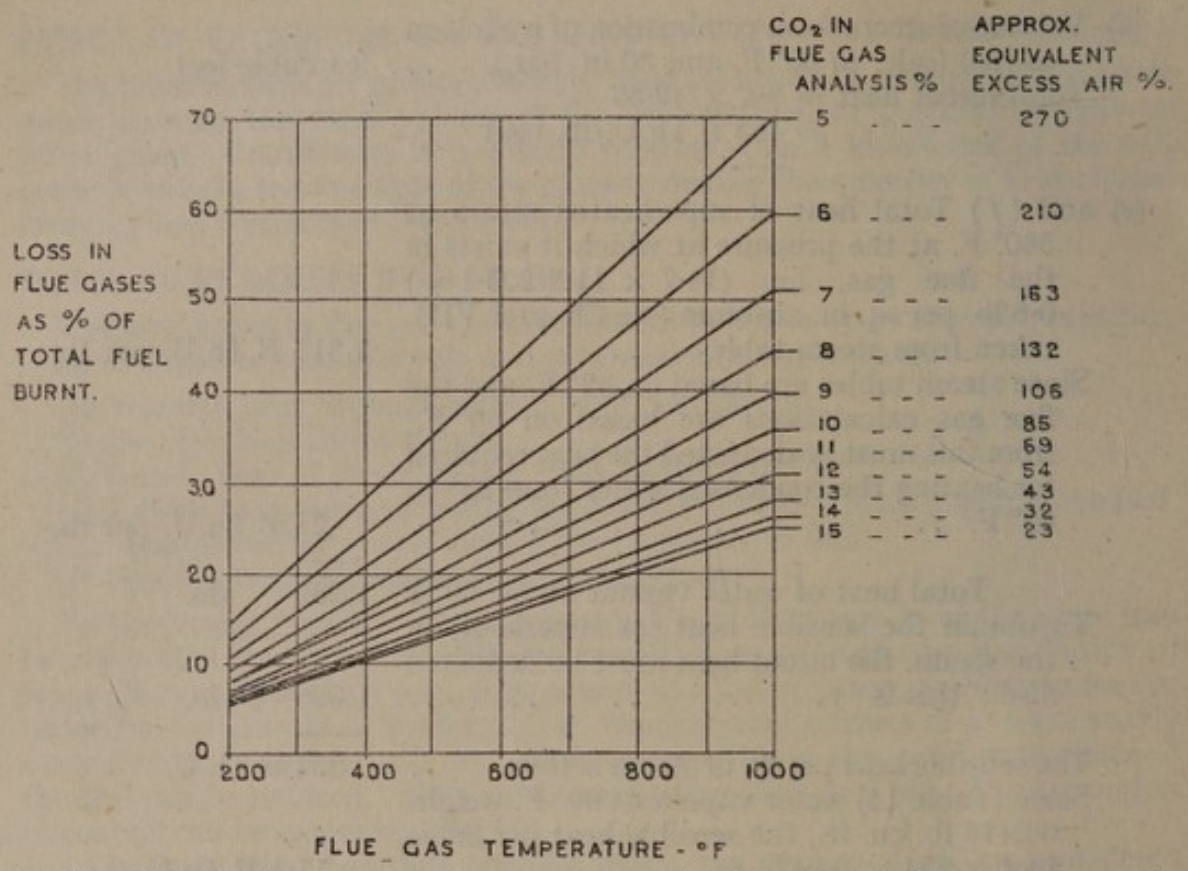

FIG. 14. Effect of temperature and excess air upon the loss of heat in flue gases from coal.

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Table 22 further illustrates this point and gives the percentage of the heat of the incoming coal that is contained in the flue gases under various conditions.

The figures in Table 22 also show that the heat losses increase rapidly as the percentage of CO_2 falls and that 1 per cent. of CO_2 in the waste gases means more in terms of heat saved or lost for low CO_2 content than for high CO_2 content.

TABLE 22. HEAT LOSSES IN FLUE GASES FROM BITUMINOUS COAL

Per cent. CO_2 in flue gas	Per cent. excess air used	Heat losses at flue gas temperatures of		
		120° C. (248° F.) per cent.	170° C. (338° F.) per cent.	200° C. (392° F.) per cent.
15	25	8·9	11·2	12·6
14	33	9·2	11·6	13·1
13	43	9·5	12·1	13·7
12	55	9·9	12·7	14·4
11	68	10·4	13·4	15·3
10	88	11·0	14·3	16·3
9	103	11·7	15·3	17·5
8	128	12·5	16·6	19·0
7	160	13·7	18·3	21·0
6	206	15·2	20·5	23·6

The first two columns of this table indicate how the proportion of excess air used can be broadly ascertained for a bituminous coal from the flue gas analysis. These figures would not apply to other fuels.

CALCULATION OF FLUE GAS LOSSES FROM SOLID FUELS

In practice it is usually found that the general characteristics of the coal used can be ascertained by simple methods of analysis, but the elementary composition (per cent. of C, H, O, etc., by weight) is not known. The CO_2 content of the flue gases and their temperature can be determined on the plant. It is then required to calculate the flue gas loss.

An indication sufficient for practical purposes can be obtained from Fig. 14. For those who wish to follow the calculation in greater detail, the following method is used. The coal in use is assumed to be known to be similar to that containing 30 per cent. of volatile matter in Table 15. It is found by tests that the flue gases contain 10 per cent. CO_2 and are at a temperature of 500° F. at the plant outlet. How much of the heat in the coal is lost ?

From Table 15 :—

Theoretical air required per lb. coal	140 cubic feet at 60° F. and 30 inches pressure
Theoretical volume of wet waste gases per lb. coal ..	147 cubic feet at 60° F. and 30 inches pressure
CO_2 content of wet waste gas	17·3 per cent...25·4 cubic feet per lb. coal
,, ,, dry ,, ,,	18·6 per cent.
H_2O content of wet waste gas	6·9 per cent...10·1 cubic feet per lb. coal

Volume of CO_2 per lb. of coal	25·4	cubic feet
Volume of total dry gas per lb. coal when CO_2 content of dry gas is 10 per cent.	254·0	,, ,,
Volume of water vapour	10·1	,, ,,
Volume of total wet gas	264·1	,, ,,
Volume of gases of combustion	147·0	,, ,,
Volume of excess air	117·1	,, ,,
Percentage of excess air	$\frac{117·1}{140} \times 100 = 83·7$	per cent.

Mean specific heat of total flue gases between the temperatures 60° F. and 500° F. .. 0·0197 B.Th.U. per cubic foot.

Sensible heat content of gases per lb. coal $= 264·1 \times 0·0197 \times (500–60)$
$= 2,287$ B.Th.U.

Latent heat content of gases per lb. coal $= 10·1 \times 49·35$
$= 498$ B.Th.U.

Total heat content $= 2,287 + 498$
$= 2,785$ B.Th.U.

If the calorific value of this coal is 14,400 B.Th.U. per lb. the loss of heat in the flue gases is

$$\frac{2,785}{14,400} \times 100 = 19·35 \text{ per cent.}$$

CALCULATION OF FLUE GAS LOSSES FROM TOWN GAS

As an example of the method of dealing with gaseous combustion the flue

gas losses from town gas will be next discussed. The gas selected for calculation in Chapter IV yielded :—

	Per 100 cubic feet of gas cubic feet	Per cubic foot of gas cubic feet
CO_2	50·8	0·508
H_2O	122·8	1·228
N_2	404·0	4·040
Total	577·6	5·776

(*a*) *Amount of Excess Air.* If more air than required is passing into a combustion chamber the process may be represented by the equation

Gas + air for combustion + excess air = Products of combustion + excess air

the amounts of each constituent being expressed by weight. The flue gas analysis is made on gas from which the water vapour has been condensed and thus, taking CO_2 as an example,

$$CO_2\,\% = \frac{\text{cu. ft. } CO_2 \text{ produced per cu. ft. of gas} \times 100}{\left\{\begin{matrix}\text{cu. ft. } CO_2 + N_2 \text{ produced} \\ \text{per cu. ft. of gas}\end{matrix}\right\} + \left\{\begin{matrix}\text{cu. ft. of excess air} \\ \text{per cu. ft. of gas}\end{matrix}\right\}}$$

The excess air contains 20·9 per cent. of oxygen, so that for the example taken, if X be the cubic feet of excess air used per cubic foot of gas, an analysis of the flue gases will give

$$CO_2\,\% = \frac{0{\cdot}508 \times 100}{0{\cdot}508 + 4{\cdot}04 + X} \quad \ldots\ldots\ldots\ldots \quad (1)$$

$$O_2\,\% = \frac{0{\cdot}209X \times 100}{0{\cdot}508 + 4{\cdot}04 + X} \quad \ldots\ldots\ldots\ldots \quad (2)$$

Both CO_2 per cent. and O_2 per cent. can be obtained from the Orsat analysis so that excess air can be expressed as :—

$$\text{From (1) } CO_2\,(0{\cdot}508 + 4{\cdot}04 + X) = 0{\cdot}508 \times 100$$

$$X = \frac{50{\cdot}8}{CO_2} - 4{\cdot}548$$

$$\text{From (2) } (4{\cdot}548 + X)\,O_2 = 20{\cdot}9X$$

$$4{\cdot}548\,O_2 = (20{\cdot}9 - O_2)X$$

$$X = \frac{4{\cdot}548\,O_2}{20{\cdot}9 - O_2}$$

(*b*) *Incomplete Combustion.* If the air for combustion is insufficient, combustible gases are present in the flue. With small deficiency of air CO and H_2 are present, but as the air is further decreased complex partially oxidised hydrocarbons are formed which have an unpleasant smell. Thus incomplete combustion cannot be detected by smell until quite serious waste is occurring, and 2 to 3 per cent. of CO may be present before a distinct smell is noticeable. Frequently, however, local incomplete combustion occurs owing to a badly designed (or defective) combustion chamber and the products may have an odour before general incomplete combustion occurs.

Analysis shows that in the initial stages of incomplete combustion the ratio of hydrogen to carbon monoxide is 2 : 1, so that for every 1 per cent. carbon monoxide in the flue there is a potential heat loss due to incomplete combustion alone of 7 per cent. The importance of controlling combustion by flue gas

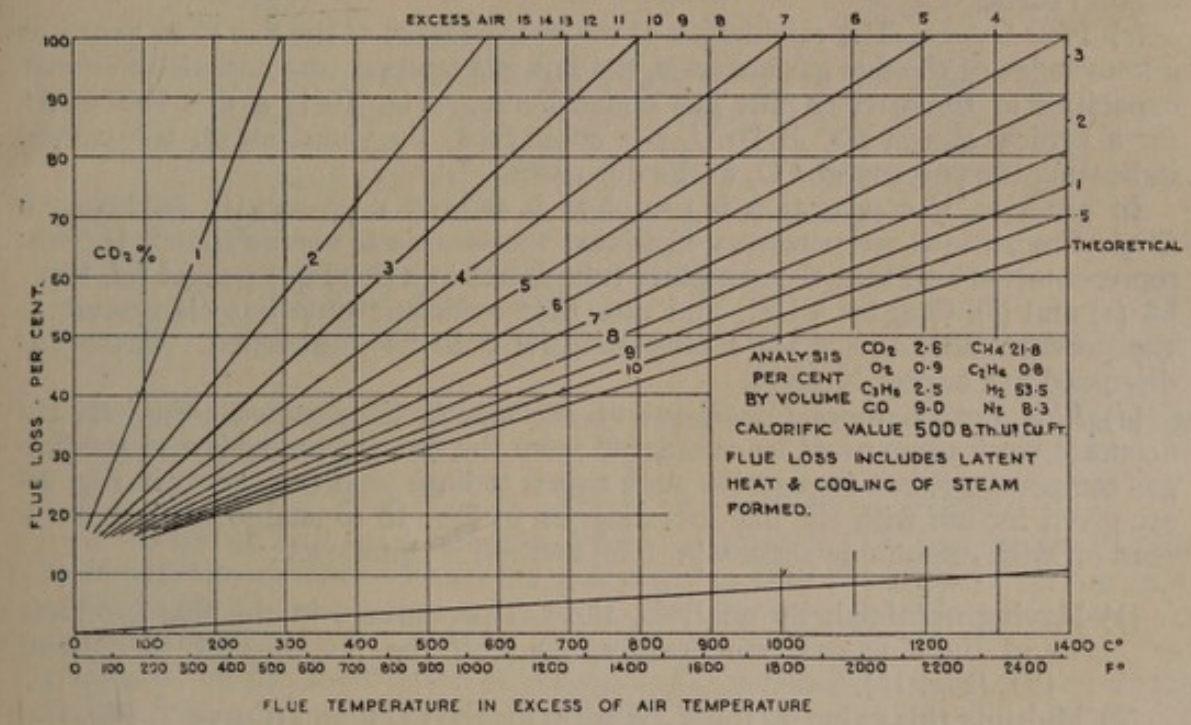

FIG. 15. Effect of temperature and excess air upon the loss of heat in flue gases from town gas.

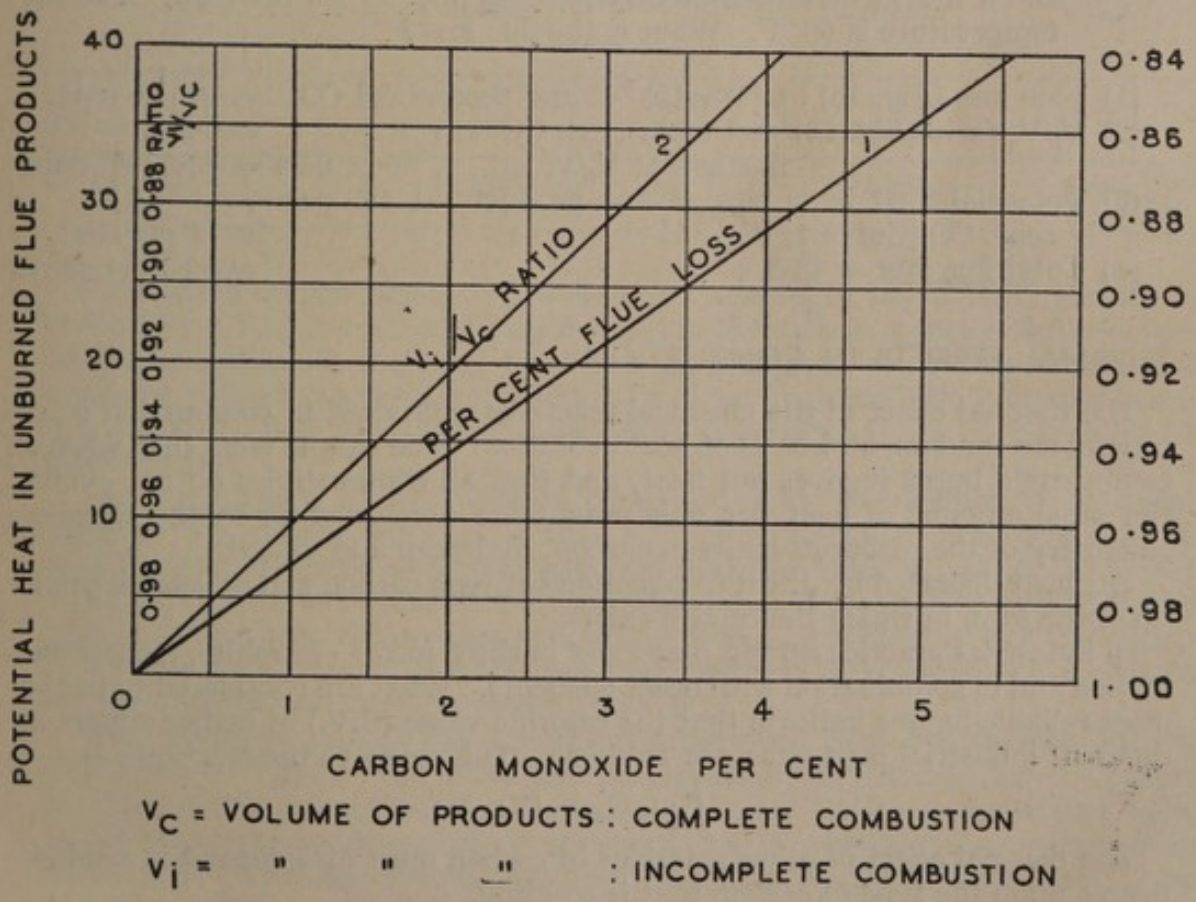

FIG. 16. Diagram for calculation of flue gas loss in the presence of unburned gases.

analysis is at once apparent. This applies equally, of course, to the combustion of solid fuels.

(*c*) *Flue Losses.* The calculation of the heat content of the flue gases requires a knowledge of the flue gas analysis, the flue gas temperature and also the heat capacities of the various flue gas constituents. Fig. 15 is a flue loss chart for a typical gas of 500 B.Th.U. per cubic foot, the numbers on the curves indicating the percentage CO_2 in the flue gases.

In practice, if combustion is complete it is only necessary to analyse for CO_2. Precautions are necessary in taking the sample to ensure that it is truly representative and that the sampling tube is not in a dead gas pocket (cf. Figs. 63 (*a*) and (*b*), Chapter VIII), and also that sufficient time has elapsed since the previous adjustment for steady conditions to be established. These are discussed in Chapter XXXII.

(*d*) *Flue Losses under Conditions of Incomplete Combustion.* For strictly accurate work these must be calculated from the flue gas analysis and the flue gas temperature, but for routine work this is tedious, and the curves in Fig. 16 are given for use with the flue loss diagram in Fig. 15 to enable the loss to be read off with reasonable accuracy. The method is as follows :—

(1) Having obtained, by analysis, the CO percentage in the flue products and the flue gas temperature, read off the flue loss at theoretical CO_2 (Fig. 15).
(2) Multiply this value by the V_i/V_c ratio for the CO found (Curve 2, Fig. 16).
(3) Read the potential heat in the unburned flue products for the CO found (Curve 1, Fig. 16).
(4) The total flue loss is given by the sum of the results from (2) and (3).

Example : A carburising furnace has a flue temperature of 925° C., and a flue gas analysis of CO_2 10·5, O_2 nil, CO 1·5 per cent. The air temperature is 60° F. What is the flue loss ?

(1) Flue loss (Fig. 15) at (925–15)° C. and theoretical CO_2 = 46 per cent.
(2) V_i/V_c ratio (Fig. 16) at 1·5 per cent. CO = 0·94
flue loss × V_i/V_c = 46 × 0·94 = 43·3 per cent.
(3) Potential heat in unburned flue products at 1·5 per cent. CO (Curve 1, Fig. 16) = 11 per cent.
(4) Total flue loss = 43·3 + 11 = 54·3 per cent.

THERMAL EFFECTS IN COMBUSTION

The thermal effect of the chemical reactions comprised in combustion is, of course, the fundamental basis of heat evolution. The simple facts that when a combustible burns it gives out heat, and that all combustibles do not evolve the same amount of heat per unit weight are accounted for by the thermo-chemistry of the reactions between carbon, hydrogen and oxygen.

The heat liberated by direct combustion between carbon and oxygen depends upon the form in which the carbon exists.

In industrial practice carbon may exist in three forms—graphite, amorphous carbon and in combination with other elements. What are considered to be the most reliable figures indicate that the calorific value (C.V.) of carbon differs in different industrial products as in Table 23. This refers to the reaction :—

$$C + O_2 = CO_2$$

The thermal effect of other reactions of carbon must also depend in a minor degree upon the type of carbon that is reacting.

Under conditions of insufficient air supply the reaction will go only half-way

TABLE 23. C.V. OF CARBON AND HIGH-CARBON FUELS

	B.Th.U./lb.	
Carbon:		
Carbon as graphite	14,130	
Carbon in metallurgical coke ..	14,200	
Carbon in horizontal retort coke ..	14,290	
Carbon in vertical retort coke ..	14,380	
Carbon in coal	14,450	
Fuels:		
Ash-free, dry metallurgical coke ..	14,250	
Ash-free, dry gas coke	14,400	
Combustible matter in ashes	14,540	(B.S.I. figure)

and carbon monoxide is formed. The thermal value of the reaction, based on carbon as graphite, is :—

$$2C + O_2 = 2CO + 4{,}000 \text{ B.Th.U. per lb. of carbon.}$$

When the carbon monoxide thus formed is burnt to CO_2 the net result is the same as if the carbon had been burnt directly to CO_2. The C.V. of carbon monoxide is an invariable figure for which the most reliable determinations are close to 10,150 B.Th.U. per lb. of carbon contained in the CO. The heat generated by the combustion of carbon to CO depends on the character of the carbon from which it is formed (Table 24).

TABLE 24. HEATS OF REACTION OF C, CO AND O_2 IN B.TH.U. PER LB. CARBON

	Graphite	Metallurgical coke	Gas coke	Coal
C to CO	3,980	4,050	4,200	4,300
CO to CO_2	10,150	10,150	10,150	10,150
C to CO_2	14,130	14,200	14,350	14,450

Thus when carbon burns to CO there is a loss of 10,150 B.Th.U. per lb. of carbon, over 70 per cent. of the initial heat in the carbon, a fact which provides adequate reason for the need for obtaining complete combustion. In a gas producer, there is evolved some 4,000 B.Th.U. per lb. of carbon, and the remaining 10,150 B.Th.U. is generated when the CO is burnt as a constituent of the producer gas.

The combustion of hydrogen is complicated by the formation of water vapour. In furnace work, it is rare for the products of combustion to be cooled within the process to such a degree that the water vapour is condensed to liquid. When the water formed by combustion is not condensed within the system, as has been pointed out previously, heat is unavoidably lost amounting to 1,055 B.Th.U. per lb. of water not condensed (49·35 B.Th.U. per cubic foot of water vapour assumed measured at 60° F.).

Thus hydrogen and all fuels containing hydrogen have two calorific values. The higher or " gross " value assumes that all the water is condensed and cooled to atmospheric temperature. The lower or " net " value assumes that none of the water is condensed, and is obtained from the gross value by deducting 1,055 B.Th.U. per lb. of water condensed from the products of combustion by cooling to 60° F.

Combustion of hydrogen :—

$2H_2 + O_2 = 2H_2O$ (liquid) + 61,500 B.Th.U. per lb. of hydrogen

and

$2H_2 + O_2 = 2H_2O$ (vapour) + 52,000 B.Th.U. per lb. of hydrogen.

Combustion of sulphur :—

$S + O_2 = SO_2$ + 4,000 B.Th.U. per lb. of sulphur.

When the combustion of a solid or liquid fuel containing oxygen is considered, since the oxygen present is assumed to be already combined with the hydrogen of the fuel an allowance of hydrogen equivalent to the oxygen present must be deducted from the effective weight of combustible hydrogen. This deduction is equal to one-eighth of the oxygen content, as has already been explained.

The calorific value of a coal may be calculated from its composition (see Table 1) by the following expression :—

$$\text{Gross C.V. (B.Th.U. per lb.)} = [\text{C per cent.} \times 14{,}450 + (\text{H per cent.} - \frac{\text{O per cent.}}{8}) \times 61{,}500 + \text{S per cent.} \times 4{,}000]/100$$

For calculating the net C.V. of coal, the C.V. of 61,500 for available hydrogen must be replaced by 52,000.

This formula gives correct results for coal within about ± 200 B.Th.U. per lb., the discrepancy arising from the fact that formulæ of this character cannot take into account the heat of formation of the original material.

The calorific value of gases is determined and calculated by volume, and in British units is expressed in B.Th.U. per cubic foot measured at 60° F. sat.

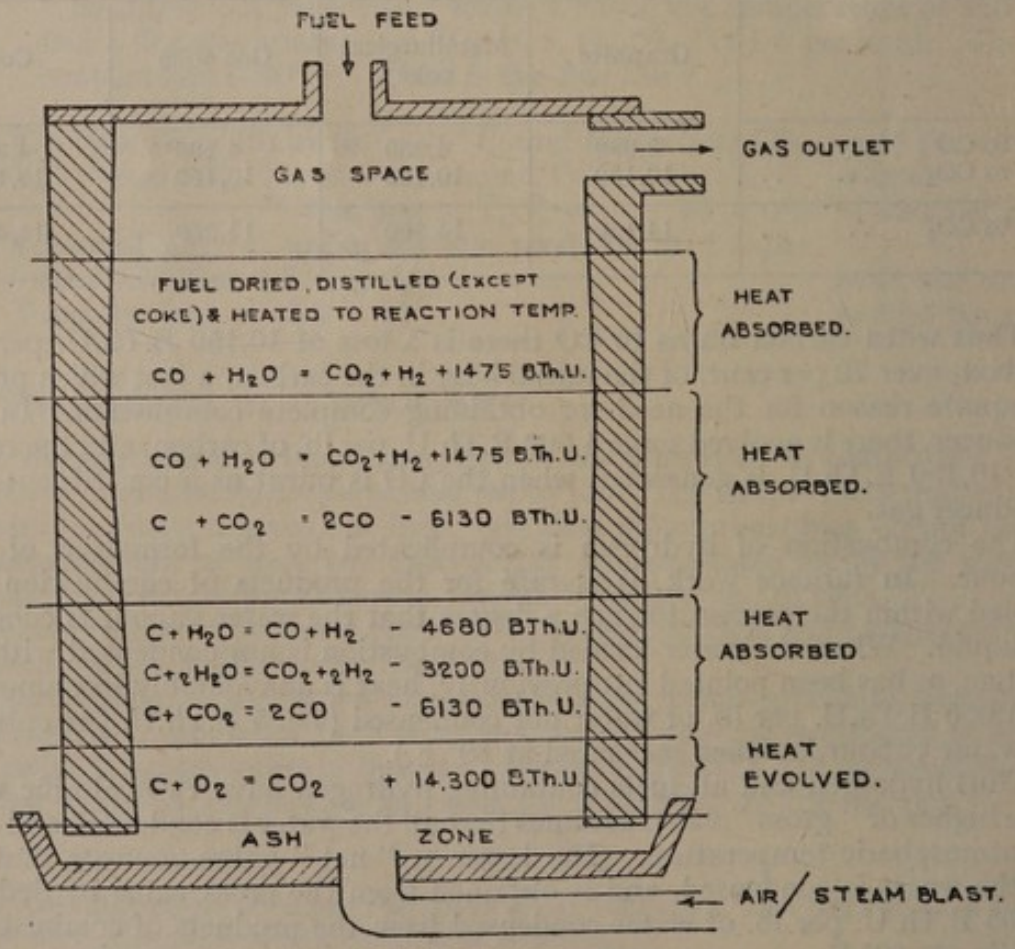

FIG. 17. Thermal effects in a deep fuel bed (gasification).

and 30 inches pressure. This subject was dealt with under "Coal Gas" in Chapter II, where a list of the C.V. of industrial gases under these conditions was given.

THERMAL EFFECTS IN GASIFICATION

The chemical reactions which take place when a mixture of air and steam are blown into a deep fuel bed—and to a lesser extent in a shallow fuel bed—have been described in Chapter IV (Fig. 13). The importance of temperature in enabling the desired reactions to be effected with the necessary velocity was there emphasised. To obtain the required temperature in the reaction zones it is necessary to generate sufficient heat from the combustion of the fuel to compensate for all losses, including the heat absorbed in chemical reactions. A knowledge of the thermal effects in gasification is thus very necessary.

Fig. 17 indicates these thermal effects expressed as B.Th.U. per lb. of carbon taking part in the reaction. The thermal effect of some of these reactions depends on the condition of the carbon as has just been indicated. With one minor exception, all the reactions absorb heat except the direct combustion of the fuel at the base of the producer, as indicated by the "minus" sign.

IGNITION TEMPERATURE

No substance will burn until it has been raised to a temperature at which its reaction with oxygen is sufficiently rapid. Visible ignition is preceded by a temperature range in which slow combustion is occurring. This was illustrated by the behaviour of coal during storage (Fig. 12, Chapter III).

The ignition temperatures in air of a number of fuels are given in Table 25. These temperatures are not absolute, but depend to a greater or lesser extent on the conditions.

TABLE 25. IGNITION TEMPERATURES IN AIR

	°C.	°F.
Bituminous gas coal	370	700
Ordinary bituminous coal	400–425	750–800
Welsh steam coal	470	880
Anthracite	500	930
Coke (soft)	425–500	800–930
Coke (hard)	500–650	930–1,200
Hydrogen	580–590	1,080–1,095
Carbon monoxide	644–658	1,191–1,216
Methane	650–750	1,200–1,380
Ethane	520–630	970–1,170
Ethylene	542–547	1,008–1,016
Benzene	740	1,365
Toluene	810	1,490
Ethyl alcohol	558	1,036
Cylinder oil	417	783
Gas oil	336	638
Kerosene	295	562

IGNITION LIMITS

If a trace of gas is added to air and the mixture is brought into contact with a flame or an electric spark the mixture will not ignite. If, for example, the mixture consisted of 3 per cent. of coal gas with air there will be no ignition. As the percentage of coal gas is increased ignition does not occur until a limiting value is reached of 5·3 per cent. This is known as the lower limit of inflam-

mability and the mixture will ignite. Increasing the proportion of gas renders the mixture more readily ignitable, but as the percentage of gas is still further increased the inflammability becomes less until at and over 31 per cent. of gas there is again no ignition. This is known as the higher limit of inflammability. The limits of some industrially important gases will be found in Table 26.

TABLE 26. LIMITS OF INFLAMMABILITY IN AIR

Substances	Lower limit per cent. by volume	Higher limit per cent. by volume
Coal gas	5·3	31
Benzene	1·4	7·5
Blast furnace gas	35·0	74
Carbon monoxide	12·5	74
Hydrogen	4·0	75
Methane	5·0	15

FLAME SPEED AND ITS EFFECT ON INDUSTRIAL BURNERS

When an inflammable mixture of gas and air in a tube is ignited the resultant flame travels through the mixture, at first with a constant velocity and subsequently under suitable conditions with an accelerating velocity. This occurs particularly in the presence of turbulence. Conditions of deterioration may ultimately be reached in which the flame speed becomes accelerated to a very high velocity of the order of 3,300–6,600 feet per second.

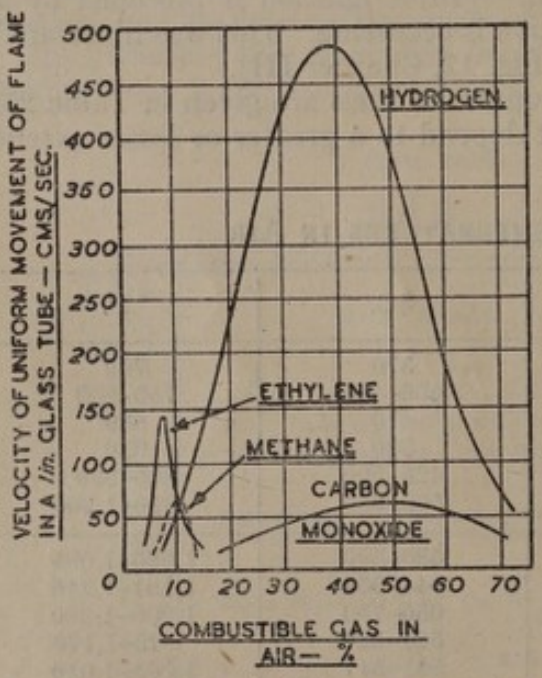

FIG. 18. Speed of flame propagation for various mixtures of air with combustible gases. Summary of Professor Wheeler's results.

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Fig. 18 shows the relation between speed of flame propagation and composition of the mixture of combustible and air for the important constituents of gaseous fuels.

The industrial significance of these properties of flame propagation is that as the composition of a gaseous fuel varies so is there a corresponding change in the tendency:

(i) for the gas burner to "back-fire." The burner is said to "back-fire" when the speed of backward propagation of the flame exceeds the velocity of flow of the combustible mixture of gas and air in the burner tube. The high flame speed of hydrogen causes gases rich in hydrogen to tend to back-fire.

(ii) for the burning gases to be blown off the burner head when the forward velocity of the combustible mixture exceeds the speed of flame propagation. The high flame speed of hydrogen assists stability on the burner head.

FLAME TEMPERATURE

The temperature which is attained by a gas flame (or the flame of the volatile matter from coal) depends upon several factors.

(*a*) The calorific value of the gas or vapour.
The heat supplied to the flame is mainly obtained from the combustion of the fuel. Since the steam formed by combustion will leave the flame uncondensed the calorific value that is available is represented by the net C.V. and not the gross C.V.

(*b*) The sensible heat of the air and gas.
If either the air or the gas is preheated the sensible heat thus given to the reacting substances will be added to the heat derived from combustion.
These two items together (minus item (*e*) below), make up the total heat input into the flame.

(*c*) The quantity and specific heat of the products of combustion.
The heat derived from the combustion of the gas is used for heating the products of combustion up to the flame temperature. The mass of the products of combustion multiplied by their total heat capacity over the temperature range represents the heat thus utilised. Since the specific heat of gases varies with the temperature, precise knowledge of the specific heat at high temperatures is required for the determination of the " theoretical flame temperature " so defined.
Clearly the less the heat required by the products of combustion, the greater will be the flame temperature. Inerts in the gas or any gases added to the flame which do not take part in the combustion must reduce the flame temperature.

(*d*) Excess air.
Any excess air used in the flame will add heat if the air is preheated but will on balance remove heat from the flame since the excess air must itself be raised to flame temperature.

(*e*) Dissociation.
The combustion reactions $C + O_2 = CO_2$ and $2H_2 + O_2 = 2H_2O$ are reversible at high temperatures, which in simple language means that at very high temperatures these reactions do not go to completion. Consequently a proportion of the heat of combustion must be subtracted from item (*a*) in accordance with the extent to which the combustion is incomplete. The higher the temperature of the flame the more incomplete will be the combustion in the flame. This effect can be ignored generally in practical applications of combustion.

(*f*) Radiation from the flame.
Theoretical flame temperatures are calculated on the assumption that no heat is radiated from the flame. This is obviously incorrect since the purpose of producing a flame is that it shall do work, i.e. that heat shall be radiated from it. For this reason flame temperatures in practice are always lower than the theoretical flame temperatures calculated according to the factors (*a*) to (*e*).
Neglecting the loss of heat from radiation under (*f*) the theoretical flame temperature is thus given by the fraction :—

$$\frac{\text{Net C.V. of gas} + \text{sensible heat of gas and air} - \text{heat loss by dissociation}}{\text{Vol. of products} \times \text{sp. ht. per unit volume at constant pressure}}$$

Calculations of flame temperature made upon these principles, the accuracy of which has been confirmed experimentally, indicate that with the exception of acetylene which is somewhat above the range and with a slight exception for hydrogen and carbon monoxide all gases containing the normal quantity of inerts up to, say, 10 per cent. or so have virtually the same flame temperature. This is because any additional heat given to the flame by increased

calorific value is counter-balanced by a greater quantity of products of combustion.

Flame temperature has a special significance as governing the thermal efficiency of the evolution of heat from flames. This efficiency is measured by the ratio of temperature difference between the flame temperature (t_1), and that of the issuing flue gases (t_2), i.e.

$$(t_1 - t_2)/t_1$$

Some figures are given in Table 27.

TABLE. 27. FLAME TEMPERATURE OF GASES AND VAPOURS

	Gross C.V. of gas	Per cent. inerts in gas	Flame temperature	
			° F.	° C.
Acetylene	1,500	nil	4,230	2,340
Ethylene	1,560	nil	3,700	2,040
Natural gas	1,220	1	3,645	2,030
,, ,, ...	1,000	0·5	3,612	1,990
Coal gas	560	6	3,710	2,045
,, ,, ...	475	10	3,710	2,045
Blue water gas ..	295	9	3,780	2,080
Hydrogen	320	nil	3,960	2,180
Carbon monoxide ..	318	nil	3,960	2,180
Producer gas	165	56	3,280	1,800
,, ,, ...	128	61	3,050	1,690
Blast furnace gas ..	92	71	2,660	1,460

The lower flame temperature of the last three gases is due to their high content of inerts. The flame temperature can be raised by preheating both air and gas as indicated in Fig. 19. This fact also indicates the very considerable influence of excess air in cooling the flames.

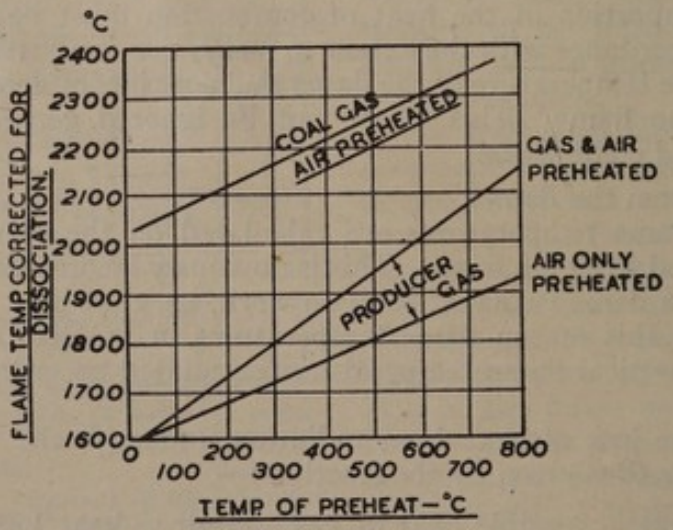

FIG. 19. The effect on flame temperature of preheating air.
(A. Fells, *Trans. Inst. Gas E.*, 83, 612.)

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CHAPTER VI

COMBUSTION IN PRACTICE

The art of securing proper combustion—Draught—The calculation of natural draught—Dimensions of the chimney—Primary and secondary air—Flow of air through fuel beds—Excess air—Flue gas losses—Time, temperature and turbulence as factors in combustion technique—Physical characteristics of the fuel bed—Combustion of various fuels and industrial combustion processes—Clinker formation—Firebed cooling—The utilisation of draught.

THE ART OF SECURING PROPER COMBUSTION

THE basic requirements for complete combustion are that the air and fuel shall be brought into contact under the right conditions and in the right proportions. To bring the air into contact with the fuel is the function of draught. Excess air above that needed to effect complete combustion causes waste of fuel as has been pointed out in Chapter V.

The conditions that must be fulfilled may be summarised as: an adequate and properly proportioned air supply introduced in the right place, together with adequate time, turbulence, and temperature to enable the combustion reactions to be effected and completed within the combustion space.

THE AIR SUPPLY

It is necessary to bring the air into contact with the fuel and to remove the products of combustion when they are formed. To do this there must be a flow

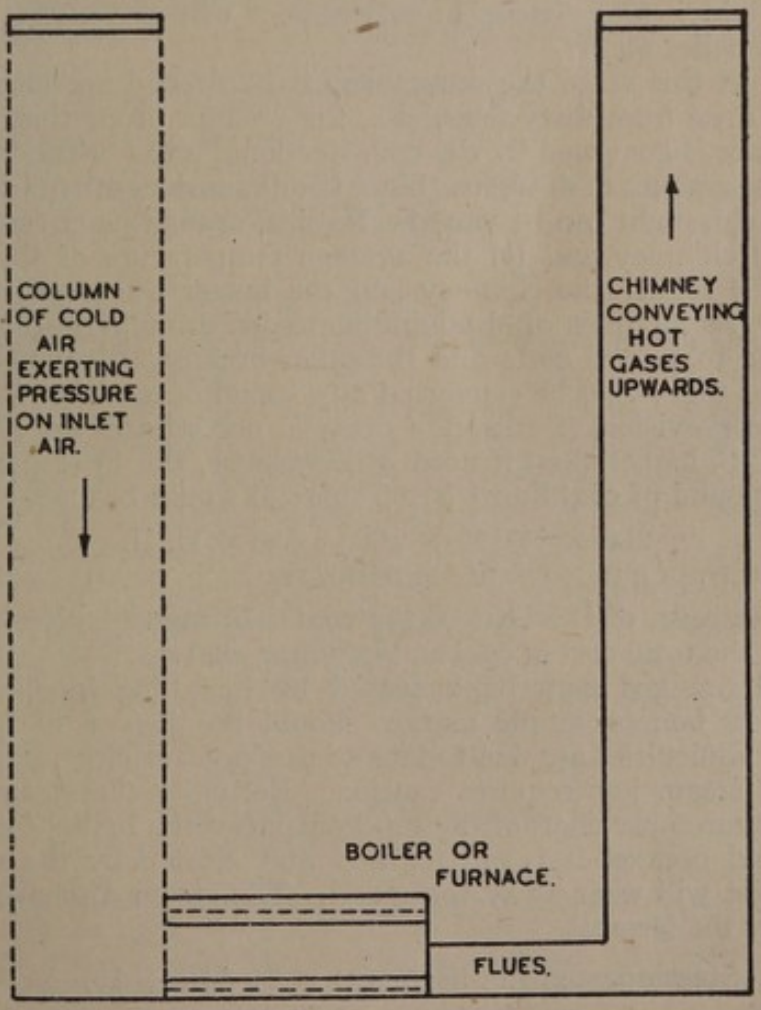

FIG. 20. Diagram illustrating natural draught and showing similarity between a chimney and a U-tube having fluids of different density in each limb.

of gases through the system as illustrated in Fig. 20. Air enters the system and passes through the furnace. Here it becomes heated by the combustion reactions in which it takes part. The gases formed in the furnace pass through the remainder of the plant to which they must give up the greater part of their heat and finally flow up the chimney. The flow of the gases is discussed in Chapter IX and the transmission of heat in Chapter VIII.

The movement of the gases, including the admission of air to the furnace, is effected in several ways :—

(*a*) The air may be drawn in by the natural draught of a chimney, or
(*b*) by an induced draught fan at the chimney base.
(*c*) The air may be forced in under pressure by a fan, this pressure serving to propel the gases through the firebed, a chimney being used to exhaust the gases.
(*d*) The air may be forced in under pressure by induction produced by a steam jet, a chimney being used as in (*c*).
(*e*) The air may be forced in by a pressure fan and the products of combustion withdrawn by a suction fan in such a way as to leave zero pressure in the combustion space above the firebed, this being known as " balanced draught."

DRAUGHT

Inefficient combustion is frequently due to the draught being inadequate to burn the amount of fuel fed on to the grate, or to failure to make use of the draught available. Deficiencies in this respect will also affect the output of steam from a boiler plant.

Neglecting at this stage the resistance of the firebed and other resistances such as may arise from dirty flues, etc., the amount of air that can be drawn into the furnace is governed by the cross-sectional areas of the flues and chimney and by the amount of draught. Since the dimensions are fixed in an existing plant only the draught can be varied. Natural draught is governed by (*a*) the height of the chimney and (*b*) the average temperature of the gases in the chimney. The higher the chimney and the hotter the gases, the greater is the draught. As methods of obtaining increased draught, both are expensive, the one owing to capital costs and the other because to create draught heat has to be used that could be converted into steam or used in other ways.

Thus if the provision of adequate draught necessitates an outlet gas temperature 100° F. higher than it need otherwise be, the heat used to produce draught per pound of coal burnt (see Table 29) would be

$$220 \times 0{\cdot}0195 \times 100 = 430 \text{ B.Th.U.}$$
$$\text{cu. ft. of gas} \times \text{sp. ht.} \times \text{temp. rise}$$

or about 3 per cent. of the heat of the coal. In monetary terms this means 3 per cent. of the total cost of coal and handling charges.

Mechanical draught may be increased by installing larger fans. When putting in new boilers, ample margin should be allowed for possible overloads ; many difficulties are due to lack of this precaution.

The use of steam jets requires caution. Normally the steam used should not be more than 5 per cent. of the total output of the boiler, but this is often much exceeded because it is not realised that erosion by the steam passing through the jet will wear away the metal. The steam discharged through a jet is given by the formula

$$\text{Steam passed (lb. per hour)} = 51\ A(P + 15)$$

where A is the area of the jet in square inches and P is the gauge pressure of the steam in lb. per square inch. Since the area of the jet varies as the

square of the diameter, a very small amount of wear involves a disproportionately great amount of steam used. Fine jets operated at high pressure are more economical in their induction effect in introducing air than large jets with low pressure.

Table 28, showing steam discharged from jets, has been calculated from the preceding formula.

TABLE 28

Gauge pressure lb./sq. in.	Lb. of sat. steam passed/hr. by each jet having diameter :—			
	$\frac{1}{32}$ in.	$\frac{1}{16}$ in.	$\frac{1}{8}$ in.	$\frac{1}{4}$ in.
10	1	4	16	63
35	2	8	32	126
60	3	12	47	189
85	4	16	63	252

There are advantages in balanced draught, the chief being that air does not leak into the furnace through cracks or imperfections in the boiler front, neither is air drawn in nor flames blown out when the boiler or furnace door is opened.

Among the advantages of mechanical draught are :—

(1) Cheaper low-grade fuels can be used, and the rate of evaporation increased by the additional draught available.
(2) Combustion is more efficient and smoke is largely prevented by the provision of adequate air supply.
(3) Furnace control is simplified.
(4) Lower chimneys can be used.

Mechanical draught is discussed in Chapter XIV ; in this present chapter attention is concentrated on natural draught.

CALCULATION OF NATURAL DRAUGHT

The limitations of the available draught will be made clearer by indicating the basic cause of air movement.

The chimney, the furnace and the outer air form in effect a gigantic U-tube, (Fig. 20). The chimney is one vertical limb ; the other equal and similar limb is a hypothetical column of cold air, and the furnace or boiler with the interconnecting flues forms the lower, horizontal portion of the U-tube. For all practical purposes the density of air and flue gas are equal, and will here be taken as 0·076 lb. per cubic foot at 60° F. at atmospheric pressure.*

If the chimney is 100 feet high, the weight of a column of cold air at 60° F. 1 square foot in section and 100 feet high is 7·6 lb.

Taking the average temperature in the chimney as 350° F., the weight of a column of waste gases of the same dimensions is inversely as the relative volumes at 60° F. and 350° F. and (cf. Chapter IV) is given by :—

$$7{\cdot}6 \times \frac{460 + 60}{460 + 350} = 4{\cdot}9 \text{ lb.}$$

* This is virtually true for high-volatile coals burnt with 50–100 per cent. excess air. For coal and coke generally the figure is 0·077–0·082, being higher for the lower-volatile fuels, and lower as the amount of excess air increases. For town gas the density of flue gas is generally between 0·073 and 0·075. These figures refer to the gas containing its full content of water vapour as leaving the combustion chamber but calculated to 60° F. and assuming that the water is not condensed.

Since the contents of the limb of the U-tube containing atmospheric air are heavier (and are continuously maintained heavier through the operation of the furnace) than the contents of the other limb, a continuous motion will be set up, causing the heavier atmospheric air to descend and the lighter hot flue gas in the chimney limb to rise.

The motive power is thus a pressure of 7·6 — 4·9 or 2·7 lb. per square foot. As a column of water 1 square foot area and 1 inch high weighs 5·2 lb., the static draught is thus calculated as being (2·7/5·2 =) 0·52 inches w.g. The measured draught with the gases moving up the chimney will be less than

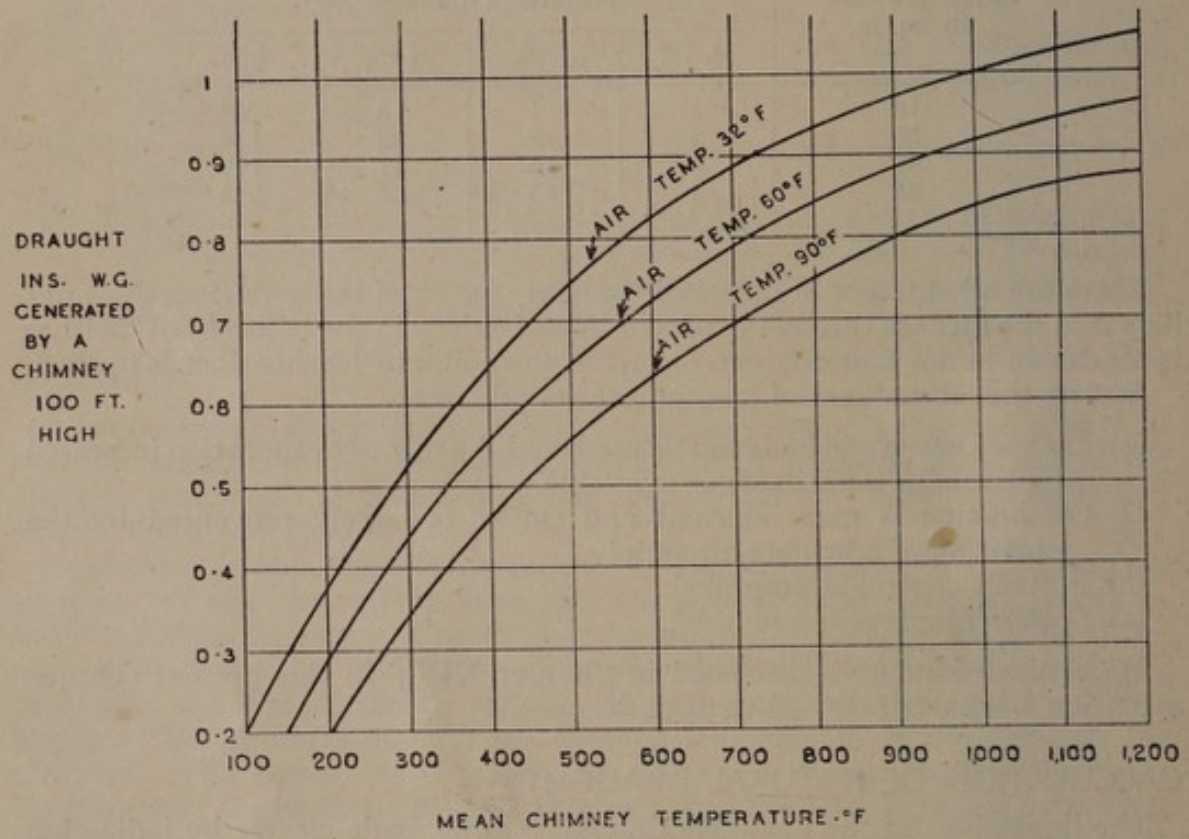

FIG. 21. Effects of gas and air temperature on static draught.

this since movement converts part of the pressure energy into energy of motion, as is described at the end of this chapter.

The following table gives the results of a calculation on similar lines for a chimney 100 feet high (*a*) with various average chimney temperatures and (*b*) with air temperatures representing a cold day and a hot summer day. The effect on the draught of these various conditions is striking (cf. Fig. 21).

	(*a*)			(*b*)	
Outside air temp.—°F	60	60	60	32	90
Mean chimney temp.—°F.	200	350	500	350	350
Draught—inches w.g.	0·31	0·52	0·67	0·59	0·44

The chimney height depends on the draught required, and may also be calculated from the formula :—

$$H = D / \left(\frac{7{\cdot}6}{t_1 + 460} - \frac{7{\cdot}9}{t_2 + 460} \right)$$

where H = height of chimney in feet.
D = draught in inches w.g.
t_1 = outer air temperature, e.g. 60° F.
t_2 = mean temperature in the chimney in °F.
7·6 & 7·9 = weights in lb. of 100 cubic feet of air and flue gas at 60° F.

For a chimney of given height and having a known mean temperature this formula can also be used to ascertain the draught that should be available.

The draught required depends on the conditions as is described at the end of this chapter.

The mean temperature in the chimney can be deduced from the temperature at the chimney base by subtracting 2° F. from the recorded chimney base temperature for every 3 feet of height.

The gas velocity that can be set up in a system of flues varies as the square root of the draught. For practical purposes the area of a chimney should be computed on the basis of a flue and chimney velocity of 15 feet per second measured at the working temperature. The gross area of chimney and flues are calculated on this basis from the probable quantity of gaseous products of combustion deduced from Table 15 allowing the proper quantity of excess air for the fuel concerned.

To the radius of the chimney as thus determined an additional 2 inches is added to allow for the diminished velocity against the walls.

PRIMARY AND SECONDARY AIR

Air admitted directly to or with the fuel is known as primary air. With solid fuels primary air passes through the fuel bed. With gas, oil and powdered coal it is admitted through the burner and therein mixed with the fuel before combustion.

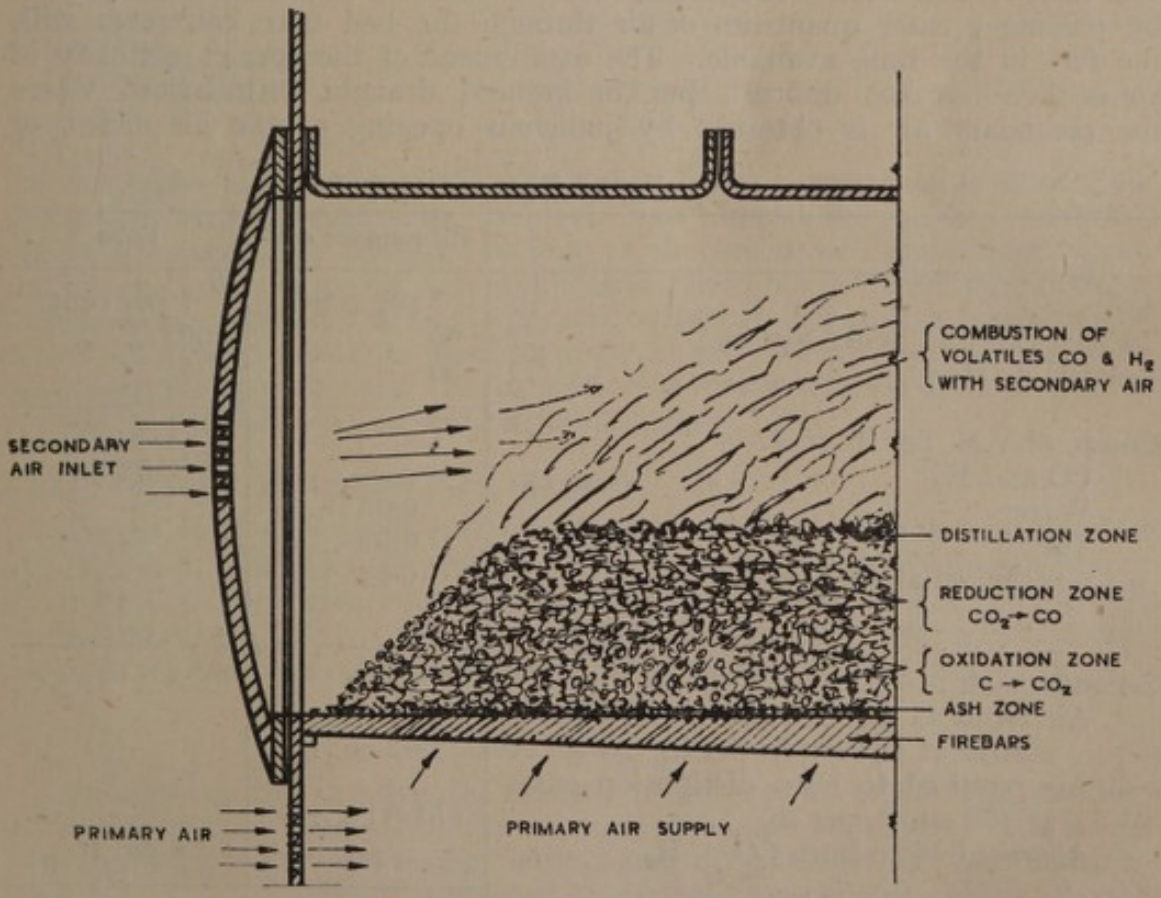

FIG. 22. Firebed zones in over-feed firing.

For gaseous and liquid fuels and powdered coal the remainder of the air necessary for combustion is admitted at a port placed (*a*) alongside the fuel burner or (*b*) farther along the furnace ; for solid fuels which evolve combustible volatile matter a further air supply must be admitted above the fuel bed. This is known as secondary air. Some secondary air may find its way up the side or back of the fuel bed or even through thin places in the fuel bed. Combustion thus takes place in two stages. A third stage, tertiary air, is occasionally used.

The firebed may be divided into zones, as in Fig. 22. At the bottom lies the bed of ash. Above this is the oxidation zone in which the reaction between carbon and the oxygen of the air forms carbon dioxide directly :—

$$C + O_2 = CO_2$$

If the firebed is deep enough, the carbon dioxide is reduced by passage over further hot carbon forming carbon monoxide :—

$$CO_2 + C = 2CO$$

Since carbon monoxide is a combustible gas, secondary air must be provided to burn it. In a shallow fuel bed very little carbon monoxide is formed.

Above this is the distillation zone. Raw coal thrown on to the hot fuel bed is carbonised or distilled, giving off its tarry gases into the combustion space above the fuel bed, where they must be burnt with the secondary air. For low-volatile fuels such as coke there is no distillation zone.

Combustion is thus brought about by the use of both primary and secondary air, and efficient combustion depends on the correct proportioning of the two. The secondary air must be so controlled and mixed with the gases leaving the fuel bed as to complete combustion within the appointed space and with as little excess air as possible.

If the fuel bed is sufficiently thin, the total air may sometimes be admitted by passing greater quantities of air through the bed than can react with the fuel in the time available. The attainment of the correct quantity of air is then not too difficult, but in induced draught installations where the secondary air is obtained by judicious opening of the air slides, or

TABLE 21A

	Bituminous coal	Coke
Ash	7 per cent.	7 per cent.
" Fixed carbon " (coke residue)	63 ,, ,,	90 ,, ,,
Volatile matter	30	3
Comp. of V.M. per lb. fuel :—		
CO and H_2	—	0·03 lb.
Water	0·06 lb.	—
Tar	0·06 ,,	—
Gas (taken as coal gas)	0·18 ,,	—
Secondary air :—		
Air required to burn tar in the volatile matter at 154 cu. ft. per lb.	9·2 cu. ft.	—
Air required to burn distillation gases at 172 cu. ft. per lb.	31·0 cu. ft.	—
Air required to burn CO + H_2	—	4 cu. ft.
	40·2 cu. ft.	4 cu. ft.

often from adventitious leakage round the grate, the fireman is faced with a far more difficult task.

Generally, too much excess secondary air is derived from leakages and the most that the fireman can do is to see that these are reduced and that his fire is in good condition. The effect of leakage is most pronounced when the burning rate per square foot is low. One of the advantages of a high burning rate is that the volume of combustible gases released is much greater, and is therefore better able to utilise the excess air arising from leakages around the firing zone and grate.

The calculation in Table 21A indicates the importance of correctly adjusting the primary and secondary air streams where the system of air admission requires that this should be done. For this purpose two dissimilar dry solid fuels are considered, taken from Table 15 (p. 54).

This calculation thus shows that with a low-volatile solid fuel such as coke, anthracite, steam coal, etc., by far the greater proportion of the air must be primary ; with the higher-volatile coals, a great deal of secondary air must be admitted to burn the volatile matter. This calculation does not complete the picture, because the whole of the excess air—say, 50 per cent. for the coal, and 25 per cent. for the coke—will be admitted as secondary air ; the complete picture is thus :—

		Bituminous coal		Coke
Primary air		94·6 cu. ft.		135 cu. ft.
Secondary air :—				
Air for combustion ..	40·2		4·0	
Excess air..	67·4		34·7	
Total secondary air ..		107·6 cu. ft.		38·7 cu. ft.
Total air per 1 lb. dry fuel		202·2 cu. ft.		173·7 cu. ft.

A secondary air inlet correctly adjusted will offer a resistance to flow of air of the same order as that of the fuel bed. But whereas the fuel bed resistance will be approximately constant so long as the fire is well maintained, clean and without holes, as the air slide is opened its resistance decreases very rapidly. Unless the secondary air slides are properly adjusted, therefore, a given suction above the firebed will thus tend to draw in air through the secondary air openings rather than through the fuel bed. Some consideration of the resistance of a fuel bed to the flow of gases will be of interest.

FLOW OF AIR THROUGH FUEL BEDS

The flow of air through a fuel bed is complicated because both streamline and turbulent flow can exist side by side. In streamline flow which occurs at low velocities, the pressure required to overcome the resistance of the fuel bed is directly proportional to the velocity (cf. Chapter IX). In turbulent flow the pressure loss is directly proportional to the square of the velocity, v,

i.e. for streamline flow : p varies as v
for turbulent flow : p varies as v^2

In a fuel bed the passages for the flow of air are dependent upon the size of the fuel and the character of the changes which are occurring, which it will be appreciated, must vary in turn with change of velocity. Further, the combustion of the air in being converted to carbon monoxide or carbon dioxide and steam takes place with a change of volume according to the proportions

of these various gases being produced. These vary throughout the fuel bed according to conditions. Again the volume of the gases is affected by the temperature in the fuel, so that the whole course of the flow becomes complicated and cannot be related to any simple statement of principles.

There is experimental evidence that the flow of air through a fuel bed is partly streamline, but is primarily turbulent, and that

$$p \text{ varies as } v^{1.72}$$

For practical purposes it can be taken that the flow is substantially turbulent. This means that to double the rate of combustion (which involves doubling the air supplied to the furnace) the draught must be quadrupled, the " draught " here meaning the pressure difference between top and bottom of the fuel bed.

EXCESS AIR

The relative ease with which secondary air is drawn into a furnace directs attention to the importance of control of the quantity of excess air used for combustion. The methods of calculation of flue gas analysis given in Chapter IV enable the amount of excess air used and the total quantity of flue gases produced to be deduced from the flue gas analysis. The exact figures will depend on the analysis of the fuel, but guidance can be obtained from Table 29.

TABLE 29. EXCESS AIR, CO_2 IN FLUE GASES AND TOTAL VOLUME OF GASES

Excess air per cent.	CO_2 content of flue gases—per cent.			Total volume of gases (approximate) cu. ft./lb. coal (somewhat less for coke and anthracite)
	Bituminous coal 40 per cent. V.M.	Steam coal 11 per cent. V.M.	Coke	
0	18·9	19·6	20·5	150
30	14·6	14·9	15·8	200
50	12·5	12·9	13·6	220
100	9·4	9·7	10·2	300
150	7·5	7·7	8·2	375
200	6·2	6·4	6·8	450
300	4·7	4·8	5·1	600

The presence of unburned gas, e.g. carbon monoxide, in the flue gases is an important source of heat loss should it be allowed to occur. In boiler practice good results are obtained when the percentage of CO_2 is between 10 and 14 per cent. using coal, the figure depending on the character of the fuel and the type of equipment.

The proportion of excess air necessary to give complete combustion depends on the temperature in the combustion chamber. As a general rule, with anything over 14 per cent. CO_2 it is likely that unburned gases such as carbon monoxide and hydrogen will be present, which will cause a loss of efficiency greater than anything gained by a higher percentage of CO_2. The presence of 1 per cent. of CO in the gases causes a loss of heat equivalent to between 4 and 5 per cent. of the coal burned ; 0·1 per cent. of CO is equivalent to a heat loss of 0·47 per cent.; 0·1 per cent. of hydrogen is equivalent to a heat loss of 0·48 per cent. ; 0·1 per cent. of methane (CH_4) is equivalent to a heat loss of 1·52 per cent. For the efficient working of Lancashire boilers and similar types of installation, the CO_2 in the flue gases should generally be in the neighbourhood of 12 per cent.

Whilst a certain amount of excess air is for practical reasons unavoidable,

the waste resulting from too much excess air must be emphasised. The heat carried away in the waste gases depends upon their heat capacity which in turn depends mainly on their quantity and temperature. The whole of the useful heat cannot be abstracted by a boiler and its auxiliaries for example ; economisers or air heaters may cool the gases to about 300° F.—generally to between 250° and 400° F. Accordingly if the volume of the waste gases is increased by excess air, so is the irreducible chimney loss for a given rate of combustion of coal.

For furnace work, as will be shown in Chapter XVIII, it is often necessary to ensure a reducing atmosphere within the furnace. For this purpose insufficient air is added for complete combustion, and the amount is adjusted to retain a little (up to 2 per cent.) of carbon monoxide in the flue gases, which is accompanied, when burning coal gas, by 2–4 per cent. of hydrogen.

The quantity of excess air that it is essential to use when firing with coal is generally not more than 50 per cent., and may be as low as 30 per cent. under specially favourable conditions. Coal gas and producer gas can be used with less than 10 per cent. of excess air, oil fuel with less than 15 per cent. and powdered fuel with less than 20 per cent., the essential factor being the uniformity with which fuel and air are fed into the combustion chamber. With gaseous fuel and almost to the same extent with liquid fuel, regulated streams of fuel are admitted at a uniform rate. This uniformity can be secured with coal to a larger extent by means of mechanical stokers and especially by firing powdered fuel ; it cannot be secured with hand firing. Thus the amount of excess air admitted to the combustion chamber must depend upon the method of firing. Not infrequently 100 per cent. or more excess air is found to be used with hand fired boilers.

GRAPHICAL CALCULATION OF HEAT LOSSES IN FLUE GAS

Detailed methods of calculating flue gas losses have been given in Chapter V. To simplify these methods in practice charts may be constructed similar to Figs. 14 and 15, and the heat content of the flue gases may then be directly read off the charts when the CO_2 content of the gases (or the amount of excess air used) and the flue gas temperatures are known. Since the maximum CO_2 content varies for different types of fuel, depending on the carbon and hydrogen content, charts must be calculated for the particular fuel used.

To avoid the labour of calculating complete sets of curves for each fuel, the more comprehensive method of Figs. 23 and 24 can be used.

Fig. 23 as here reproduced is drawn for four fuels, namely pure carbon, a bituminous coal, a fuel oil and a coal gas. It lends itself, however, to other compositions of the same fuels. It cannot be used for two-stage gases such as producer gas and blast furnace gas, for which the more elaborate curves previously described must be drawn, or the calculations made from first principles (Chapter V).

To construct Fig. 23, the straight line diagonally across the chart from the point (0, 0) to the point (21, 21) is taken as the basis. This line represents the theoretical CO_2 content of the flue gas from any fuel when no excess air is present. On this line a point is marked corresponding to the theoretical CO_2 content for the fuel used ; thus for the fuels illustrated in the chart the points selected are :—

	Per cent. CO_2
Coke oven gas and town gas	11·5
Fuel oil	15·5
Bituminous coal	18·5
Pure carbon	21·0

$$\text{MAXIMUM } CO_2 \text{ CONTENT (DRY)} = \frac{21C}{C + 2{\cdot}37H}\ \%$$

WHERE

C = % CARBON CONTENT OF FUEL

H = % AVAILABLE HYDROGEN CONTENT OF FUEL

$= \left(H - \frac{O}{8}\right)$

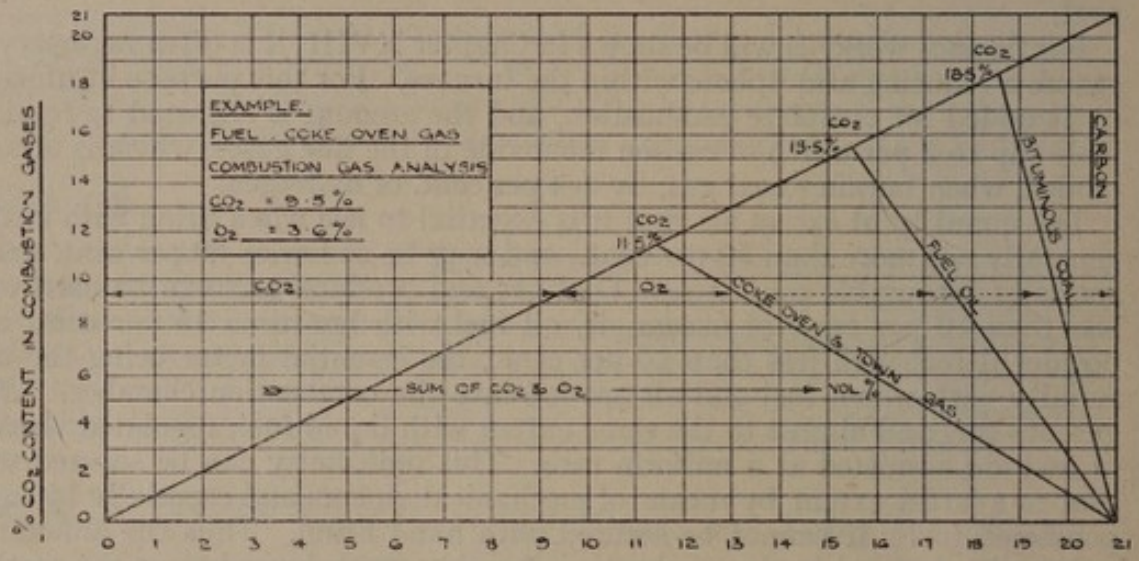

FIG. 23. Relation between CO_2 and O_2 in flue gases from coal gas, oil and bituminous coal.

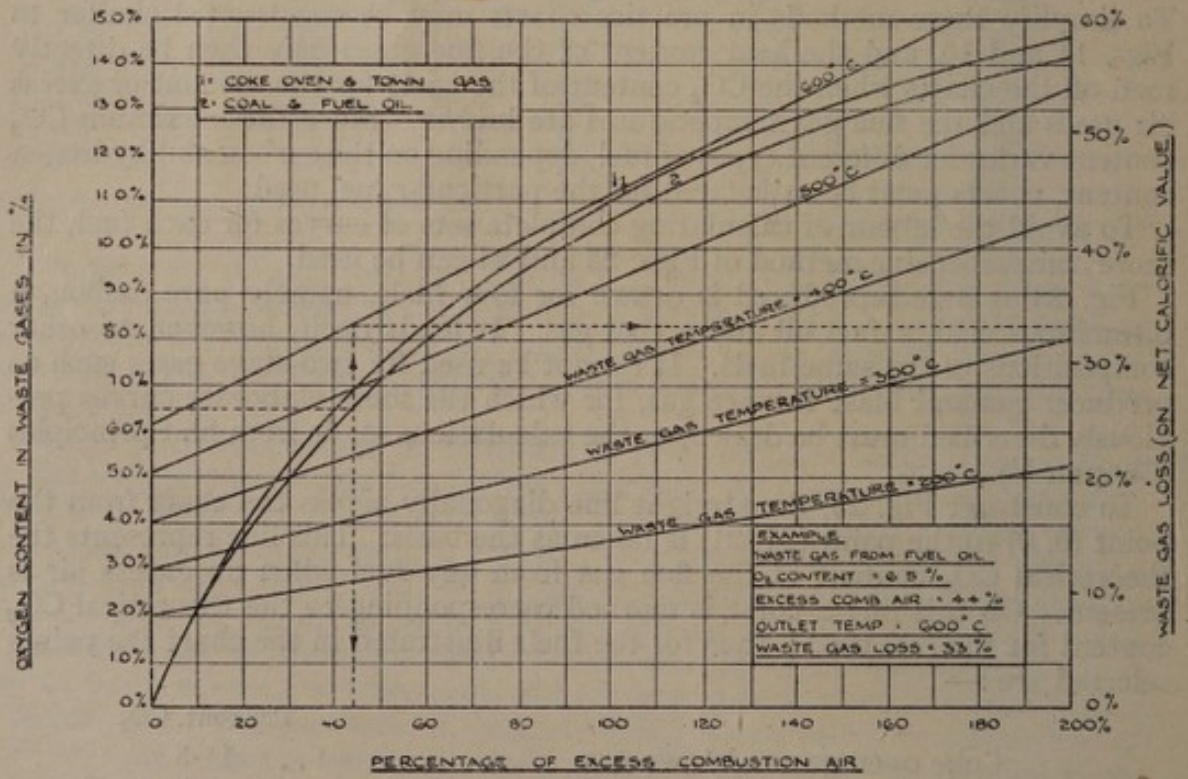

FIG. 24. Chart relating flue gas loss with oxygen content of the gas.*

* The waste gas loss in Fig. 24 does not include the latent heat of the water vapour formed by the combustion of the fuel.

If coke were used the point 20·5 would be selected. Some town and coke oven gases have a theoretical flue gas CO_2 figure of 10·5 per cent. and some may be as high as 13 per cent. Whatever the figure may be for the fuel used, the position of the maximum CO_2 content by analysis (i.e. with the water vapour condensed out of the gas) is marked on the basic diagonal line. This point is then joined to the point (0, 21) in the bottom right-hand corner of the chart, as has been done for the four fuels taken as typical.

The method of using Figs. 23 and 24 will be best explained by an example. Let it be assumed that oil fuel is used of such a composition that the theoretical flue gas contains 15·5 per cent. CO_2 by analysis and that the flue gas temperature at the plant outlet is 572° F. (300° C.). The actual CO_2 content of the gases is 10 per cent. From Fig. 23 it is found by measuring horizontally from 10 per cent. CO_2 on the extreme left-hand side that with 10 per cent. CO_2 the flue gases will contain 7·5 per cent. O_2. From Fig. 24, Curve 2, it is seen that 7·5 per cent. O_2 indicates 55 per cent. excess combustion air. Then from the point where the waste gas temperature curve for 300° C. cuts the vertical line from 55 per cent. excess air, and reading horizontally on the extreme right, it is found that the stack loss is 16·5 per cent. of the net C.V. of the oil.

If an allowance of 5 per cent. be made for radiation and similar losses, the thermal efficiency of heat utilisation would be $100 - 16{\cdot}5 - 5 = 78{\cdot}5$ per cent.

On the gross C.V. of the fuel the stack and radiation loss would be

$$\frac{(16{\cdot}5 + 5) \times \text{net C.V.}}{\text{gross C.V.}}$$

THE TIME FACTOR IN COMBUSTION

Having introduced the proper quantity of air into the furnace, divided when necessary in the right proportions of primary and secondary air, the further factors of time, temperature and turbulence may be considered.

All chemical reactions require a definite time for their completion, this being due either to the inherent nature of the reaction or to the difficulty of bringing the reactants in contact sufficiently quickly. Consequently a certain time must be allowed for combustion both in a solid fuel bed and in the gaseous combustion of the volatile matter or of a gas. With solid fuel, a bed which is too thin (or thin in relation to the size of the fuel used) enables air to pass through unburnt because there is not adequate contact between air and carbon. In gaseous combustion the volume of the combustion space must be adjusted to the conditions. For rapid combustion the air and gas (or powdered coal in pulverised fuel firing) must be as intimately mixed as possible immediately upon entry into the furnace. If the mixing is not intimate the flame may be very long and may extend outside the furnace proper into the flues.

TURBULENCE

Turbulent flow as opposed to streamline flow is the means whereby air and gas may be mixed rapidly. Parallel streams of gas and air will diffuse slowly into one another and cause slow combustion. If the flow is turbulent, however, the air and gas are thoroughly mixed in a small volume so that the opportunity for bringing the oxygen and gas into contact is increased.

In boiler furnaces the volatile matter is evolved from the coal and the secondary air may tend to pass over the top of the volatiles as shown in Fig. 25. When this happens turbulent flow may be induced by a jet of steam or air directed downwards at an angle. One such arrangement which makes use of preheated air for the purpose is shown in Fig. 26.

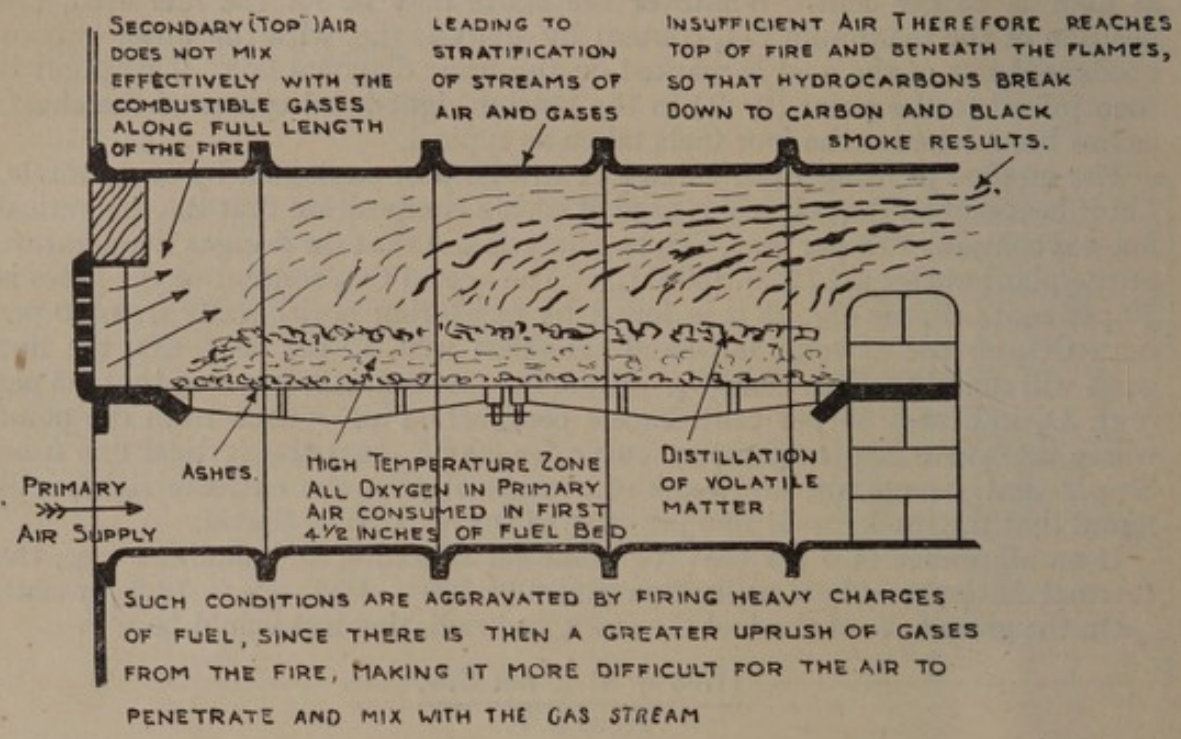

FIG. 25. Stratification effects in boiler furnaces.

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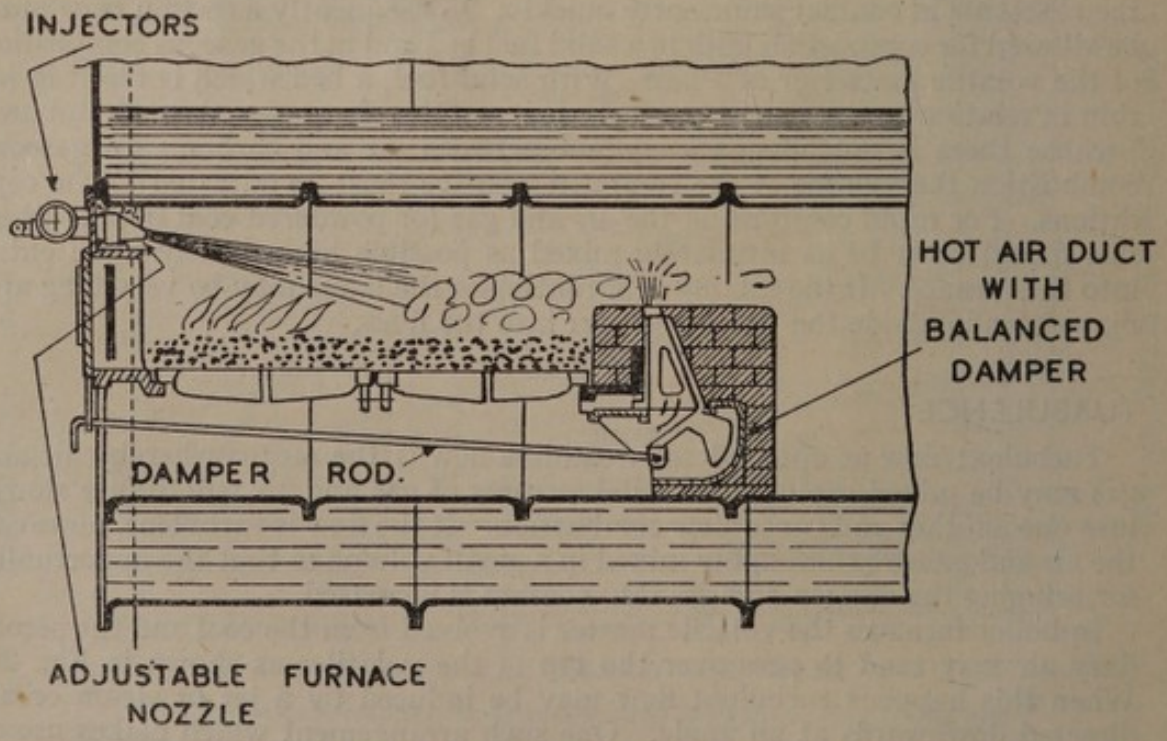

FIG. 26. A method of increasing turbulence in boiler furnaces.

TEMPERATURE

Combustion will not take place unless the temperature is sufficiently high. The temperature of ignition of materials has been dealt with in a preceding section. Unless this temperature of ignition is exceeded combustion cannot take place. It is particularly necessary to mix the secondary air and the combustible gases as soon and as well as possible when there are cold surfaces in the proximity of the grate. It is in this respect that preheated secondary air has an advantage. This early mixing must be avoided in certain furnaces and kilns where long flames are required.

The speed of reaction increases and is commonly doubled or trebled for each 10° C. rise in temperature. Consequently a high temperature, which may be promoted by preheating air for example, will assist in obtaining more rapid combustion. It has been computed that at furnace temperatures, hydrocarbons whether contained in coal gas or the volatile matter from coal are decomposed completely into carbon and hydrogen in one-thirtieth of a second.

If production of black smoke is to be avoided from liquid fuels or from the volatile matter of coal this carbon must be burnt. An important influence upon this is the chilling effect of cold surfaces. The flames from burning coal are approximately at 2,000° F. (1,094° C.) or higher according to the temperature in the fuel bed and the distance the flame has travelled from the hot incandescent fuel, whereas boiler metal, for example, is no higher than the saturated steam temperature, generally below 450° F. (230° C.). In a furnace the stock will generally be charged cold. If the flames impinge on these cool surfaces combustion may be arrested and carbon monoxide and hydrogen may go forward into the flue gases. In extreme conditions carbon will remain unburned and will form smoke.

The chilling effect of cold surfaces on flames may go farther than this. If a basin of cold water is held in a luminous flame of coal gas the basin will quickly be covered with a film of carbon. In the same way soot may be deposited on metal surfaces. This carbon represents loss of fuel, as has just been indicated, and in addition it has a low thermal conductivity so that the heat transmission from the flame to the vessel being heated is reduced.

Chilling of furnace flames can arise from :—

(*a*) Heavy infrequent charges.
(*b*) Excessive air supply, e.g. by prolonged opening of fire doors.
(*c*) A long flame which impinges on the metal, e.g. a flame from very high volatile coals.
(*d*) Thermal features which force the burning gases into intimate contact with cooler metals, e.g. too high a fire-bridge in boilers.
(*e*) The design of the plant, e.g. the circular section of shell boiler furnaces makes them more liable to this type of chilling than other types of boiler furnace.

PHYSICAL CHARACTERISTICS OF THE FUEL BED

The importance of the condition of the fuel bed has been emphasised when dealing with the flow of air. Above the fuel bed there is a lower pressure than exists below the bed. This pressure difference, aided by the buoyancy of the hot gases within the fuel bed, causes air to flow through it. The amount of fuel burnt depends upon the amount of air that flows through the bed, provided that the air interacts with the fuel (cf. Table 30).

Clearly the ideal conditions are that

(1) the fuel bed shall offer uniform resistance to the flow of gases at all parts where there is combustible fuel to be burnt,

(2) the fuel shall be of such a size that it will react rapidly and uniformly with the incoming air.

The practical characteristics of the fuel bed will now be discussed.

Size of Coal. In order to make the fullest use of primary air it is essential to have a uniform fuel bed. A small-sized, graded fuel gives the best air distribution. A small percentage of fines through $\frac{1}{8}$ inch mesh may not be a disadvantage, but the quantities of fines found in slack increase the fuel bed resistance and often cause uneven air distribution. This is largely due to segregation, i.e. separation of larger material from the fines. Some parts of the fuel bed will therefore burn out more quickly than others, leaving bare patches, through which unburned primary air rushes uncontrolled.

This difficulty can be obviated to a great extent, especially in chain-grate stokers, by thoroughly wetting the coal. The amount of water to be added depends on the percentage of fines under $\frac{1}{8}$ inch mesh. For a normal slack containing 20–40 per cent. fines, 5–7$\frac{1}{2}$ per cent. of water is required, rising to 12$\frac{1}{2}$ per cent. for fine slacks with 80 per cent. through $\frac{1}{8}$ inch mesh. This water is in addition to the inherent moisture that remains in the coal when air-dried. For really fine coals the water content may require to be over 20 per cent.

It is essential for firing on chain-grate stokers that slack should be uniformly wetted. Since this is a difficult operation to perform in practice, there is an advantage in using washed slack, already evenly wetted, instead of dry slack. The addition of water results in a small lowering of the efficiency, to the extent of approximately 0·1 per cent. for each 1 per cent. of water added, but this is far more than fully recovered by improved combustion conditions. The objection to *buying* wet slack is that the water is paid for at the price of coal.

Coking Properties. The coking properties of fuel affect the physical character of a fuel bed to a marked degree. Plastic coal will not permit the passage of air and in addition a coking coal tends to give an open, irregular fuel bed, made up of large masses of coke separated by wide fissures, which give bad contact between air and fuel. As a result, a greater depth is required for complete primary combustion and the amount of CO_2 reduced to CO is small.

A non-caking, free-burning coal, on the other hand, gives a uniform fuel bed, made up of particles in close contact, so that both primary combustion and secondary reduction proceed rapidly and efficiently. Under certain conditions, fuels may form a crust on top of the bed and stop air penetration ; this crust must be carefully broken up.

Thickness of Fire. A thick fuel bed offers more resistance to the flow of air than a thin one, and therefore with the same available draught less primary air passes through a thick bed. If difficulty is experienced in maintaining steam in a boiler with a thick fuel bed, it may be possible to overcome the difficulty by carrying a thinner one, but it is, of course, rather more difficult to avoid holes in a thin bed. Some mechanical stokers operate with a thin fuel bed (cf. Chapter XI).

With bituminous coals, the firebed thickness is generally from 3 to 5 inches with chain-grate stokers and up to 8 inches with hand firing.

THE COMBUSTION OF SOUTH WALES SLACK

South Wales low-volatile slack is difficult to burn with natural draught, but burns satisfactorily with forced draught. The coal has a high calorific value, on the dry ash-free basis, of the order of 15,500–15,650 B.Th.U. per lb. as compared with 13,850 for low rank Midland coals, and it also has a low inherent moisture content. On the same ash basis, therefore, the weight of South Wales coal which must be burned to give the same heat release is three-quarters of that of the low rank coal.

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The same amount of total air is required per unit of heat generated, but a larger proportion of this must be primary air because the volatile content of the coal is lower ; moreover, the reactivity of the fuel is less with the net result that the gases leaving the fuel bed contain less CO and hydrogen, and therefore require less secondary air to complete combustion.

THE COMBUSTION OF COKE

In the preceding sections, the effect of the most important factors governing the combustion of coal, its size, reactivity and volatile content and the depth of fuel bed, was considered. Coke yields so little volatile matter that the influence of volatiles on combustion can be neglected, but the other factors are as important as in the combustion of coal.

Depth of Fuel Bed. Table 30 shows the composition of the gas at different levels in a coke fuel bed. It will be seen from the figures that it is possible to distinguish the same two zones of combustion, as those described in the coal fuel bed.

CO_2 is formed in the bottom zone. Higher up, when all the oxygen is used it is reduced to CO. For example, with the $1\frac{1}{2} \times \frac{1}{2}$ inch fuel, the oxygen is practically all used up $4\frac{1}{2}$ inches from the bottom of the bed, and increasing quantities of CO are being formed.

At the top of the bed, almost all the CO_2 has been reduced and the larger quantities of CO in the gas require secondary air to complete combustion.

Size of Fuel. The effect of size on combustion can be judged from the table by comparing the combustion of the two sizes of coke, $3\frac{1}{2} \times 3$ inches and $1\frac{1}{2} \times 1$ inch. With the larger size, there is less intimate contact between air and coke, and the combustion zone is therefore much deeper. At a level $4\frac{1}{2}$ inches from the bottom less CO_2 has been formed, and the oxygen has not been used up to the same extent. Even at 6 inches, little CO has been formed.

At the top of the bed, there is much more CO_2 and much less CO than with the smaller size, so that less secondary air will be required to complete combustion. The lack of contact between the air and the fuel results in a much lower burning rate for large coke than for the smaller size.

TABLE 30. GAS COMPOSITION IN A COKE FUEL BED

Distance from bottom of fuel bed	Composition by volume—per cent.					
	Size of coke $3\frac{1}{2} \times 3$ inches			Size of coke $1\frac{1}{2} \times 1$ inch.		
	CO_2	O_2	CO	CO_2	O_2	CO
12 inches	14·0	0·5	6·4	5·4	0·4	24·9
6 ,, ...	16·0	4·0	0·5	15·1	1·1	7·0
$4\frac{1}{2}$,, ...	12·0	7·5	1·0	16·3	1·4	5·3
3 ,, ...	—	—	—	16·8	2·9	2·7
$1\frac{1}{2}$,, ...	5·0	13·5	1·0	11·5	8·0	0·8

If the size of the pieces of fuel is sufficiently great, some air may pass through the bed without contact ; this is known as channelling. When this happens, the unused air functions as secondary air.

The smaller sizes of coke have the disadvantage of tending to form objectionable clinker. This is partly due to the fact that the ash is evenly distributed through the pieces of coke and partly to their small size. Both lead to close contact, which is a primary cause of clinker formation. Difficulty with clinker formation may be lessened by using large fuel. This is also true of anthracite.

Reactivity and Combustibility of Coke. A characteristic of coke is that different types react at different rates with oxygen and CO_2. The results recorded in Table 30 relate to oven coke. With gas coke the results might be somewhat different in that more rapid combustion in the lowest zone and more rapid reduction in the next zone would probably occur.

There are also variations in the reactivity of the coke formed during the combustion of coal, due to variations of conditions within the fire. Generally it seems that the coke formed at high rates of combustion is more reactive than that formed at low rates.

Coke burns more slowly than coal and requires more draught because a much greater proportion of primary air is needed. In boiler practice it is frequently found that sufficient secondary air is derived from air passed through or round the grate or entering through imperfections in the fitting of the doors, and that the regular secondary air slides should be closed. This applies also to anthracite. The composition of the combustion gases from coke differs somewhat from those from coal (see Table 29).

SMOKE

Smoke is formed from the tarry distillation products of coal, which escape before combustion is complete. When the amount of volatile matter in the coal becomes small, the quantity of tar evolved from the coal becomes much less, and the less is the liability to smoke ; this is the reason for the smokeless characteristics of Welsh smokeless coal, anthracite and coke.

There are two forms of smoke : (1) brown smoke, which is mostly tarry matter given off at relatively low temperatures when fresh coal is fired on to a hot surface and which is relatively easy to consume, and (2) black smoke, which consists largely of particles of carbon derived from the cracking of hydrocarbon gases at a high temperature, and is probably formed when the fuel bed is stirred. Excess of air even if preheated is not sufficient to burn this black smoke ; it must also be subjected to a high temperature when in contact with air. The two smokes may, of course, occur simultaneously.

To consume smoke there must be sufficient air, adequate mixing of air and volatiles, and the temperature must be sufficiently high to enable combustion to take place rapidly and before the gases come into contact with cold surfaces.

The amount of carbon in dense smoke amounts to not more than 1 per cent. of the carbon in the coal ; the real danger lies in the possible presence of other unconsumed combustible gases. On the other hand, attempts to avoid smoke by using considerable quantities of excess air may cause much greater losses. A chimney serving a furnace working on bituminous coal should be neither clear nor smoky ; the gases should be just visible at the top and a wisp of brown smoke, in boiler practice for example, may be indicative of the most efficient conditions. A plant that never shows a trace of smoke is not burning coal efficiently.

THE COMBUSTION PROCESS IN OVERFEED FIRING

The process of combustion varies, to some extent, from one form of appliance to another and the following examples give an indication of how variations in the firebed conditions affect the proportions of primary and secondary air required.

With a thick fuel bed (cf. Fig. 22), where the raw coal is fed on to the top of the hot fuel most of the volatile matter may be driven off before the carbonaceous residue descends to the lower levels of the bed where an excess of oxygen is present. Under such conditions the volatile matter distilled unburned requires a supply of secondary air over the fuel bed for its combustion.

With a thin fuel bed, of the order of 3 inches in depth as on a sprinkler

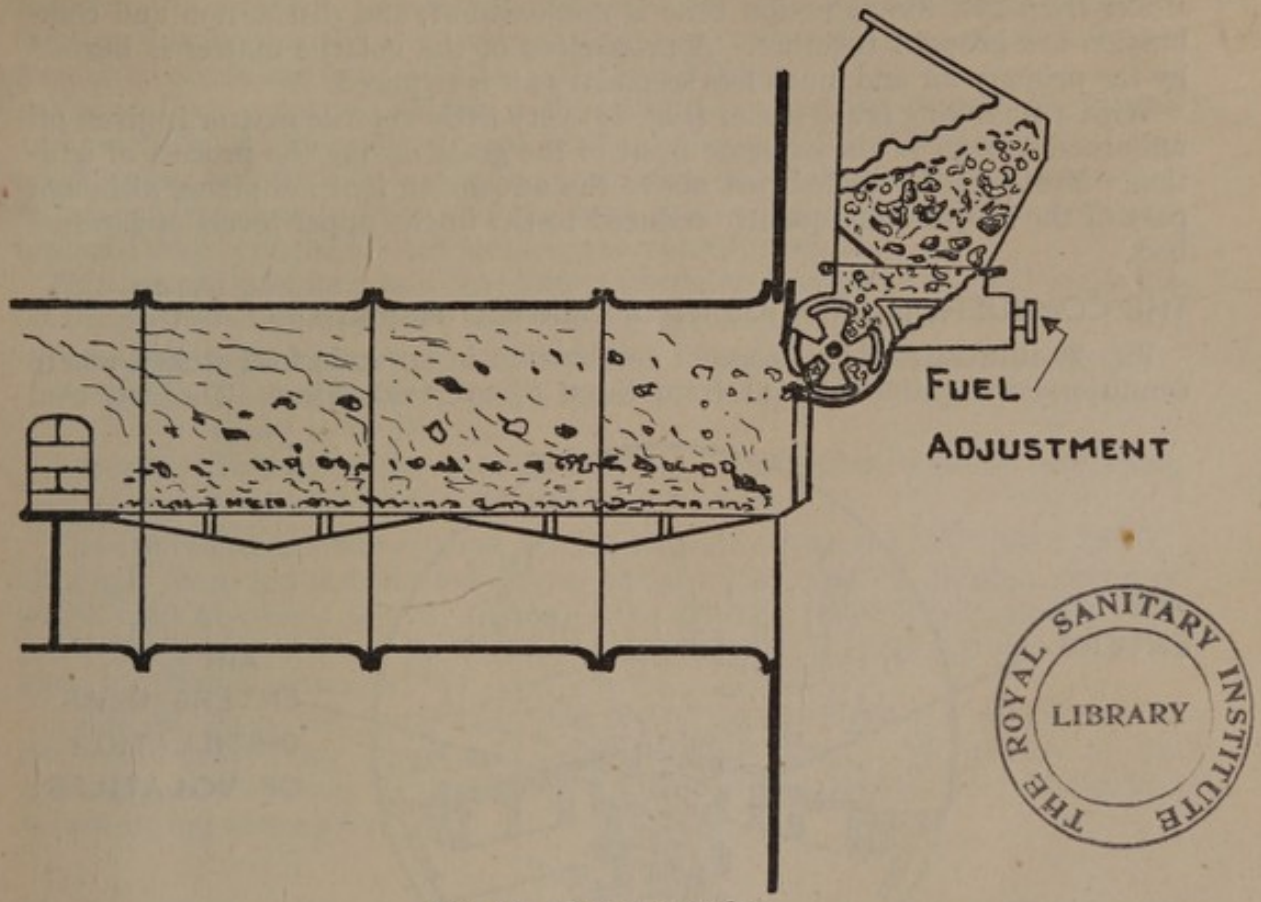

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FIG. 27. Sprinkler stoker.

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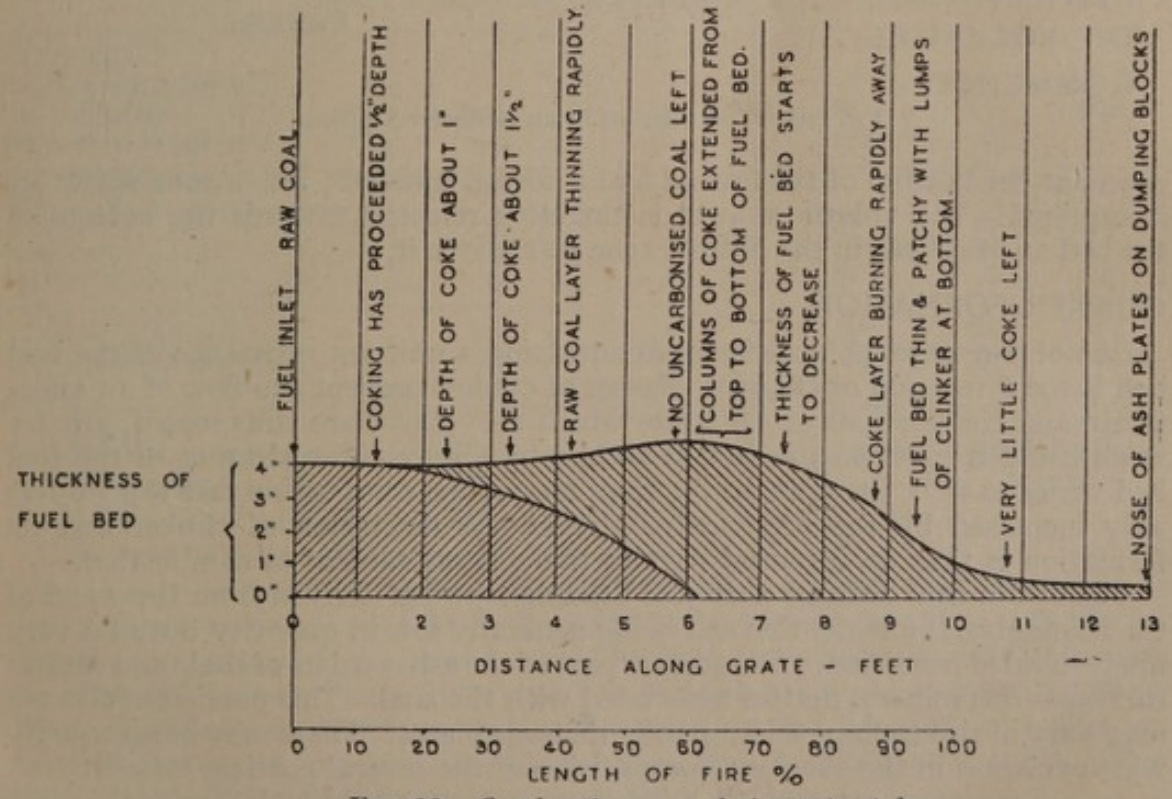

FIG. 28. Combustion on a chain-grate stoker.

(B. M. Thornton, *J. Inst. Fuel*, VIII, 13.)

stoker (Fig. 27), the reduction zone is non-existent, and distillation and combustion are brought together. A proportion of the volatile matter is burned by the primary air and much less secondary air is required.

With a travelling grate stoker (Fig. 28) very little volatile matter is given off unburned, except in the extreme front of the grate during the process of ignition. Most of it is burned just above the advancing ignition plane, although part of the CO_2 is subsequently reduced to CO in the upper levels of the fuel bed.

THE COMBUSTION PROCESS IN UNDERFEED FIRING

Fig. 29 illustrates the process of combustion in an underfeed stoker, where conditions are quite different from those of overfeed firing. The raw coal

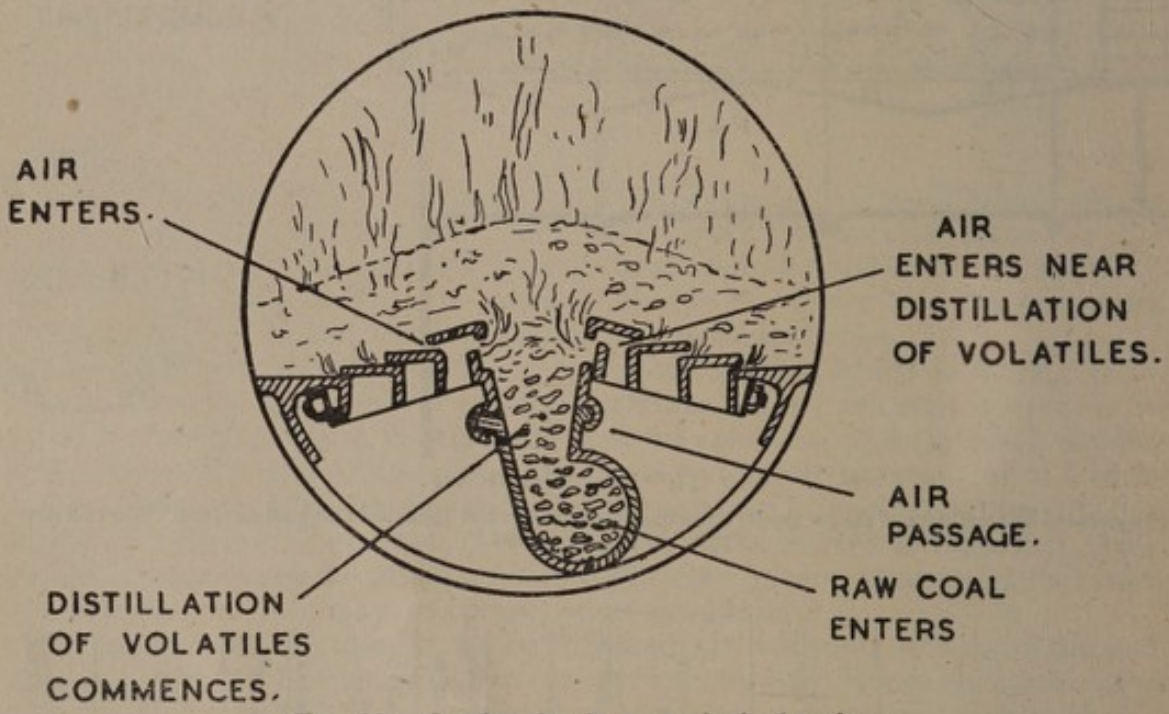

FIG. 29. Combustion in an underfeed stoker.

enters at the bottom of the firebed and is forced upwards, into a zone where air is supplied. The volatile matter is therefore released towards the bottom of the bed and is burnt in the hottest zone just above it.

CLINKER FORMATION

One of the major difficulties in maintaining a uniform resistance of the fuel bed is the formation of clinker. Masses of clinker prevent the flow of air effectively and the portions of the grate which they cover are thus sealed, greater work being thrown on the rest of the grate. Since the resistance of the fuel bed varies as $v^{1.72}$, the draught required for a given combustion rate is considerably increased by the presence of clinker. A discussion of clinker and its formation is thus an essential part of the general subject of combustion.

The ash in coal consists partly of the inherent ash derived from the wood of the original coal forests, this ash being generally low in quantity but in a very finely divided condition. The greater part of the ash consists of shale and similar carbonaceous mineral matter associated with the coal. This portion of the ash may vary in size from powder up to quite large pieces. There may be an equally wide variation in the chemical composition of the several particles of ash.

An ash may thus consist of a mixture of comparatively fusible with infusible substances, and clinker formation may occur by the fusible substances binding

together the infusible. Iron oxide which under reducing conditions can form very fusible ferrous silicate, has probably the predominating influence in the formation of clinker from solid fuel (Chapter XXII).

The melting point of coal ash is determined by grinding the ash to powder and determining the melting point of the mixture. Unless this mixture gives a low value, it cannot be said to be a criterion of liability to clinker formation because it takes no account of the fact that a small proportion of fusible material may bind the whole together into masses of clinker.

As a general guidance, and based upon the determination of ash melting point on the powdered mixture, the following figures have been given :—

Group 1. Melting point, 1,425°–1,710° C. Clinkering troubles absent.
Group 2. Melting point, 1,200°–1,425° C. Clinker may form, but is not likely to be serious.
Group 3. Melting point, 1,040°–1,200° C. Clinkering troubles excessive, unless adequate precautions are taken.

The ash melting points of most British coals lie between 1,050° and 1,350° C. although there are notable exceptions such as the ashes of durains, many of which melt at over 1,450° C. Some of the ash constituents will usually melt below the temperature of the fuel bed (1,300°–1,500° C.) and will thus afford a basis for clinker formation.

Table 31 gives the results of analyses of nearly 800 coals which, whilst not necessarily being completely representative, give a fair picture of the melting points of ashes of British coals. These melting points were determined in a reducing atmosphere.

TABLE 31

	Number of coals with ash melting points between stated limits					Total No. of coals tested
° C. ° F.	1,100 2,012	1,100–1,200 2,012–2,192	1,200–1,300 2,192–2,372	1,300–1,400 2,372–2,552	1,400 2,552	
SECTION 1						
English coals (excluding Durham)	10	298	288	71	17	684
Expressed as per cent.	1½	43½	42	10½	2½	—
SECTION 2						
Durham	—	6	5	14	15	40
Scotland	—	3	7	6	12	28
South Wales	—	5	4	19	7	35
Total of Section 2 ..	—	14	16	39	34	103
Expressed as percentage	—	13½	15½	38½	33	—

Whether a given ash will form clinker or not will depend upon several circumstances :—

(*a*) The distribution of the mineral matter in the coal.

Clinker may form through complete fusion of the ash, which will lead to large sizes of clinker ; or by the cementing together of pieces of shale which are themselves relatively infusible, in the manner previously described. It can be shown that particular fractions can be obtained from coals, by a combination of sizing and float-and-sink tests, which melt much below the average temperature. Fusible and infusible constituents are thus not blended in a uniform manner in the coal, but may be distributed differently in different parts of the seam and in different laminations in each lump of coal.

(*b*) The atmosphere to which the ash is exposed.

A coal ash will melt at a lower temperature in a reducing atmosphere than in an oxidising atmosphere. Experiments suggest that for coals having a fusion point above 1,200° C. the difference is 30°–50° C. Below this the difference is greater as will be seen from the following figures :—

Melting range °C.	Temperature difference between oxidising and reducing conditions °C.	°F.
1,150–1,200	80	144
1,100–1,150	175	315
1,050–1,100	200	360

In ashes from boilers and furnaces the iron is almost completely reduced to the ferrous state so that the conditions are the worst possible in respect of clinker formation. The decrease in fusion temperature is due to the formation of complex ferrous silicates. A thin fuel bed is the more likely to produce an oxidised ash and so to reduce clinker formation.

(*c*) The extent to which the mineral matter becomes mixed with the fuel in the hottest zone of the fuel bed.

Disturbance of the fuel bed during combustion tends to raise mineral matter from the grate (where it is cooled by air) into the hottest part of the fire. Slicing and poking should be kept to a minimum. In under-feed stoking the fuel bed is stirred up thus assisting the formation of clinker, whereas on chain grates, where the fuel bed is moved bodily forward, disturbance is a minimum.

Allied to this is the contact between the several particles of coal. This will be greater as the coal decreases in size, the smaller the coal the greater the likelihood of clinker formation. It has been suggested that the very finely divided inherent ash may have a greater effect on clinker formation than the rough associated dirt.

In those zones where the coal is burning most actively the temperature will be the highest. Much of the ash will then be within the coal, and ash particles may not be so readily brought into contact with one another as in the last stage, when most of the carbon has disappeared ; in this last stage, however, the combustion is less intense and the temperature will fall. This view is confirmed by published observations (*J. Inst. Fuel*, **14**, 223) showing that both with coal and coke clinker formation occurs more than 4 inches above the grate, and as it sinks through the fuel bed, the clinker solidifies below the 4-inch level. This illustrates the next point also.

(*d*) The time to which the ash is subjected to any high temperature.

Chemical reactions may take place with the formation of mixtures of low melting point and unless there is sufficient time given these low fusible constituents may not be formed if they do not exist already in the coal.*

(*e*) For clinker to form in any quantity the conditions must be such that a sufficient amount of slag shall be produced which is fluid enough to permit it either to run together in masses or to percolate through other masses of infusible mineral matter so as to bind them into aggregates.

(*f*) Preheated air will raise the fuel bed temperature and thus favour clinker formation.

Further information on clinker formation may be found in Chapter XXII, under " The Action of Slags."

FIREBED COOLING

Steam is sometimes used under fire-grates (in addition to its use in forced draught injectors)

* Some ashes will form mixtures of minimum melting point, in which the melting point is less than that of the individual constituents. Quite frequently, however, the melting point of the mixture will be above that of the individual constituents.

(*a*) To reduce the grate temperature and prevent material damage.
(*b*) To reduce the firebed temperature to diminish or prevent clinker formation.

Jets of steam are blown into the fuel bed from below somewhat in the manner indicated in Fig. 30. This use of steam is quite distinct from the use of steam jets to promote draught or pressure and the steam acts wholly as a cooling medium.

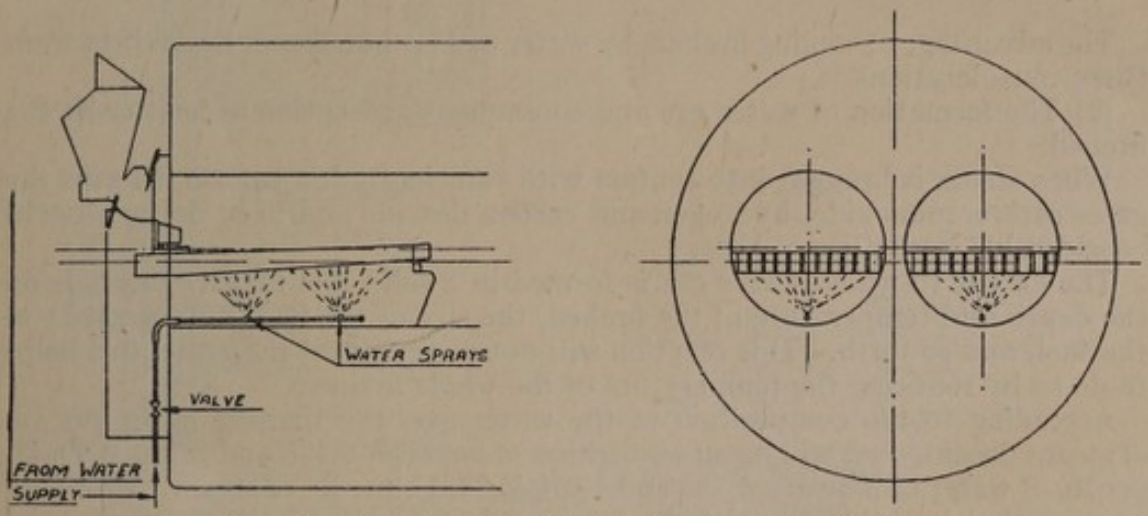

FIG. 30. Arrangements of water jets for cooling firebars.

The quantities of steam involved may, unknown to the operator, attain such dimensions as to cause a heavy strain on the available steam supplies. Published records suggest that 6 per cent. of the total steam production of the plant is not an unusual figure, though this is at least three times more than should be necessary. Wear on the nozzles and inattention whereby steam is left on unnecessarily, are responsible for much waste.

Steam used for cooling fire-grates operates in two ways :—

(1) Mechanical contact with the grate and the firebed results in the steam being heated to a high temperature and thus abstracting heat.

In considering this effect, three factors must be taken into account.

(*a*) If saturated steam is contained in the primary air stream under the grate, the steam has a total heat of 1,150 B.Th.U. per lb. If in passing through the grate and lower portion of the firebed it is then superheated to 1,000° C. (1,832° F.), its total heat becomes 1,994 B.Th.U., representing an abstraction of (1,994–1,150 =) 844 B.Th.U. per lb. of steam used. Clearly it would be advantageous if water could be used instead of steam, thus permitting the additional 1,150 B.Th.U. per lb. to be applied for direct cooling, instead of representing coal wasted through additional generation of steam in the boiler.

(*b*) This 844 B.Th.U. of heat is not necessarily used where it is most needed. The steam is raised in temperature primarily in the combustion zone, and is reasonably efficacious in cooling this part of the firebed. Its direct effect in cooling the grate is small. If water were used, on the other hand, a considerable proportion of the 1,150 B.Th.U. mentioned above would be available for directly cooling the grate or the clinker immediately above it.

(*c*) Following the argument just developed in (*b*) it is instructive to observe the relative cooling effect of the primary combustion air and the steam or water admitted with it. A coal may require about 100 cubic feet of primary air per lb. of coal, and when cooling with steam it is customary,

as previously indicated, to use about 0·3 lb. steam per lb. of coal. If the air and steam are heated to 500° F. (260° C.) when passing through the grate and the ashes at the immediate base of the fire, it can be shown that :—

100 cubic feet of primary air absorbs	860	B.Th.U.
0·3 lb. steam absorbs	41	,,
0·3 lb. water absorbs	386	,,

The advantage of cooling firebars by water rather than steam is manifest from these considerations.

(2) The formation of water gas and consequent absorption of heat cools the firebed.

When steam is brought into contact with sufficiently hot carbon it forms the gases carbon monoxide, hydrogen and carbon dioxide and in so doing absorbs considerable quantities of heat.

The extent to which water gas is formed in a solid fuel furnace depends on the depth and temperature of the firebed, the size of the fuel, the reactivity of the fuel, and so forth. This reaction will not directly cool the grate, but helps to do so by reducing the temperature of the whole firebed.

According to the composition of the water gas, the thermal effect per lb. of steam decomposed will be an absorption of between 4,170 and 2,120 B.Th.U. per lb. of water or between 3,120 and 1,070 B.Th.U. per lb. of steam. Probably no more than half the steam will be decomposed, and if 0·3 lb. of steam are used per lb. of coal, and if the gases and undecomposed steam leave the fuel bed at 900° C. (1,652° F.), the total cooling effects per lb. of coal would then be :—

Water spray ..	between 1,104 and 774	B.Th.U. per lb. coal.
Steam admission ..	between 772 and 443	,, ,, ,,

There is experimental evidence that the effect of saturating at 50°–60° C. the air blast of a gas producer with steam, involving the use of some 0·3–0·4 lb. steam per lb. of coal gasified, lowers the hot zone temperature by some 300° C. (540° F.), i.e. from 1,400° to 1,100° C. (2,550°–2,010° F.). It has been found (also experimentally) that in a travelling grate stoker it requires 1 per cent. of the steam generated in the boiler to lower the link temperature by 30° C. (53° F.) with a caking coal and 50° C. (90° F.) with a non-caking coal.

Danger to the firebars occurs principally after periods on high load. During high load periods the firebars are cooled by incoming primary air. The firebed meanwhile increases considerably in temperature. If a period of moderate load follows, the cooling effect of the incoming air is diminished, since its quantity is decreased, but the high firebed temperature increases the temperature of the grate. When a boiler is continuously on a moderate load it is generally possible to operate without artificially cooling the grates.

The considerations just advanced have led to investigations on the possibility of using water sprays instead of steam jets for cooling purposes. Fig. 30 shows the general arrangement and Fig. 31 the design of a nozzle that has been found satisfactory for the purpose (cf. B. M. Thornton, *Engineering*, September 4th, 1942, p. 142). It is essential that the spray of water should be very finely atomised, and provision should be made for draining any drip back from the firebars on to the flue.

The quantity of water required for cooling has been found to be only one-half to one-third that of the steam needed. Figures published by B. M. Thornton (*loc. cit.*) obtained on an Economic boiler of 15,000 lb. per hour are as follows :—

Gross steam generated per hour	15,000 lb.
Evaporation per lb. of coal	7·5 lb.

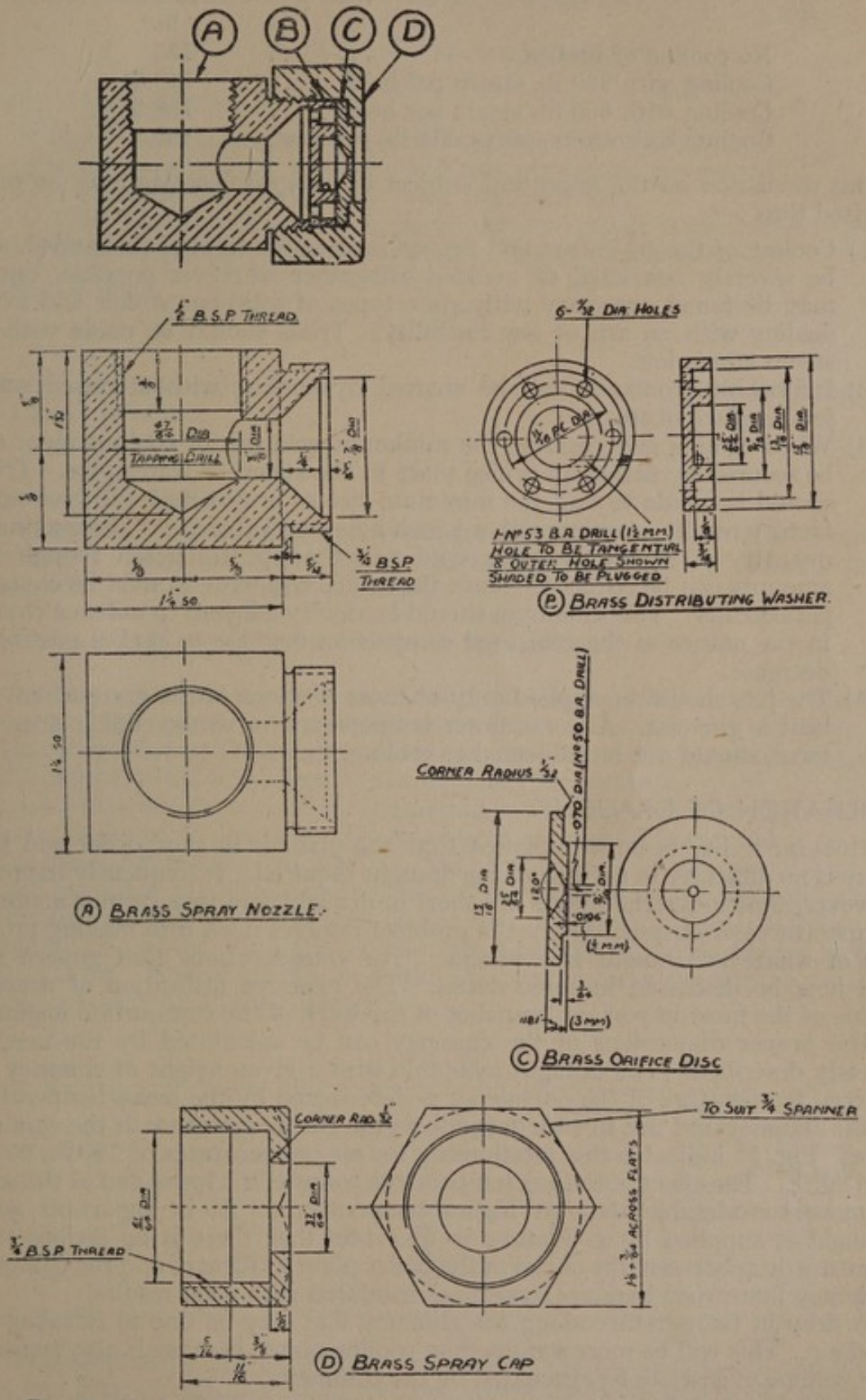

FIG. 31. Design of ½-in. spray jet suitable for cooling firebars by water sprays (B. M. Thornton).

	Coal required per 1,000 lb. steam lb.
No cooling of firebed	133
Cooling with 860 lb. steam per hour ..	140·5
Cooling with 600 lb. steam per hour ..	138·5
Cooling with water sprays, 270 lb. per hour	135

This discussion on the important subject of fire-grate cooling may be summarised thus :—

(1) Cooling of the grate bars and firebed is often wasteful of fuel and should be severely restricted or avoided altogether wherever possible, but it may be found necessary with some types of solid-bar stoker and when dealing with an ash of low fusibility. Trials should be made without steam admission.
(2) Important economies can be secured by cooling with atomised water jets in place of steam.
(3) When cooling is necessary, the minimum quantity of cooling agent can be secured by fitting an orifice plate in the steam or water pipe. Trials should be made to find the minimum size of orifice that will give satisfactory results, starting with a $\frac{1}{8}$-inch hole (see Table 28). The maximum quantity of steam or water used will then be restricted though the amounts can be reduced. Since the orifice will wear it must be changed periodically. This minimum should be rigidly adhered to unless a change in the nature of the coal used compels an increase or makes possible a decrease.
(4) The jets should be applied only to those portions of the grate where the heat is greatest. Areas of lower temperature, or where clinker does not form, should not be subjected to cooling.

UTILISATION OF DRAUGHT

Thus far it has been presupposed that the plant is in good order and that there is no difficulty in getting all the draught required. It frequently happens, however, that industrial plants are short of draught and thus cannot maintain temperatures or output. Often the problem resolves itself into making proper use of whatever draught is available. The considerations that govern this will now be discussed in some detail. The effective utilisation of draught is one of the most important branches of the work of the combustion engineer.

The proper dimensions of the chimney can be calculated by the method already described. The draught available with a given height of chimney for a given temperature of the outside air is determined by the mean temperature in the chimney and this in turn depends on the gas temperature at the chimney base. Fig. 21 indicates this relationship for air temperatures of 32° F., 60° F. and 90° F. The effect of the gas temperature towards the lower end of the scale is more considerable than at higher temperatures. It is important when draught is supplied by a chimney to ascertain that there is no serious temperature drop between the outlet of the boiler plant or furnace and the chimney. Chimney flues when long are sometimes insulated with good results.

A drop in temperature along the chimney flue may be due to inleakage of cold air. This will be more serious than cooling alone, because it also increases the volume of gases to be evacuated by the chimney.

The volume of flue gases is an important factor, and this reverts back to control of excess air in combustion. Chimney area is calculated on the principle that a given velocity, say 15 feet per second at the prevailing temperature, is permissible. Velocities much higher than this will cause a high friction loss

in the chimney and consequently will reduce the draught. The pressure loss (P) in the chimney can be shown to be (cf. Chapter IX) :—

$$P = \left(1 + f\frac{4H}{D}\right)\frac{wv^2}{2g} \text{ lb. per square foot}$$

where f is the coefficient of friction,

H and D are the height and diameter of the chimney in feet.
g is the gravitational constant, 32·2 feet per second per second.
w is the density of the flue gas in lb. per cubic foot.
v is the linear velocity of the gas in feet per second, both w and v being taken at the mean temperature prevailing in the chimney.

The first term,

$$1 \times \frac{wv^2}{2g}$$

is the loss of pressure due to the velocity of escape from the top of the chimney and the second term,

$$f \cdot \frac{4H}{D} \cdot \frac{wv^2}{2g}$$

is the loss by friction.

If w_o and v_o are the density and velocity respectively under standard conditions of 60° F. and 30 inches mercury pressure, and the mean chimney temperature is t° F., the expression becomes

$$P = \left(1 + f\frac{4H}{D}\right) 0{\cdot}00000574\ w_o\ v_o^2\ (460 + t) \text{ inches w.g.}$$

From this expression it will be seen that the total pressure loss in the chimney increases as the *square* of the velocity of the gases. The quantity of excess air used in combustion will thus have an important bearing on the available draught, in addition to its adverse effect on the chimney temperature.

As an example, there will be considered a chimney 100 feet high, 6 feet diameter designed to pass 254 cubic feet of gas per second (measured at 60° F.) with a mean temperature of 400° F. This corresponds to a value of v of 15 feet per second or v_o of 9 feet per second. It is further assumed that with these conditions, coal is being burnt with 50 per cent. excess air. Table 32 indicates the effect of excess air, if the gas temperature is assumed to be unaffected by the additional air. Excess air overloads the chimney in the same way as putting too many boilers on to the chimney.

TABLE 32

Percentage excess air	Percentage CO_2 in flue gas	Total gas flow cu. ft./sec.	v_o ft./sec.	Loss of pressure in chimney, inches w.g.
50	12·5	254	9	0·04
100	9·5	350	12·4	0·07
200	6·4	520	18·5	0·16
300	4·7	700	25	0·30

If the excess air is due to inleakage of cold air the temperature of the gases will fall. Thus, if the 254 cubic feet per second properly produced from the boiler or furnace and at a temperature of 400° F. were diluted with 266 cubic feet of cold air at 60° F., bringing the CO_2 content down to 6·4 per cent., the temperature of the gases would fall to 225° F. The loss of chimney pressure

would then be 0·13 inches w.g. instead of 0·16 inch as calculated in Table 32 ; but the draught as shown from Fig. 21 would fall from 0·55 inch at 400° F. to 0·33 inch at 225° F. The net result is a diminution of available draught of 0·55–0·33 + 0·13–0·04 or 0·31 inch w.g. Cold air infiltration is thus even more detrimental to draught than is the admission of excess air through the furnace.

This calculation indicates, of course, that a boiler plant which is just able to burn a given quantity of coal under good combustion conditions can no longer burn that quantity under bad combustion conditions with high excess air. The output of the plant is therefore reduced.

Bad combustion conditions may not reduce output in this way with mechanical draught, but the power required to drive the fan is in direct proportion to the volume of gases to be handled. Thus if the steam required for a properly operated installation be taken as 1½ per cent. of the steam generated with 50 per cent. excess air, if the excess air rises to 200 per cent., the steam used is doubled.

Having determined that the maximum of draught is available at the chimney, the next step is to make sure that there is no obstruction in the flues or plant causing undue resistance. A rough rule is to allow 0·2 inch w.g. for each 100 feet of straight flue, and 0·1 inch for each bend, but these figures will depend upon the quantity of gas flowing, and other factors (see Chapter IX). The loss of pressure due to an economiser is in general not higher than 0·25 inch w.g., and this should be checked from time to time by pressure measurements at various points in the system to ensure that accumulations of soot and flue dust are not causing undue resistance to the flow of gas. Occasionally water has been known to seep into the flues and to cause blockages.

The importance of preventing inleakage of air through faulty brickwork has been emphasised. In addition to its adverse effect on the draught available, to which reference has just been made, inleakage uses up the draught by drawing in air at places where it cannot be used for combustion, and thus starves the furnace of the combustion air there needed. It must not be forgotten that in addition to cracks, often too small to be apparent, brickwork is permeable to gases. All cracks should be carefully pointed, and the outside of the walls of boiler settings, flues and furnaces, should be coated with tar paint.

One method of searching for air leaks is to run a duck-lamp over all doubtful places and note where the flame is drawn in. Another method, applicable to certain fuels only, is to close the damper and note where smoke issues. When this method is used with coal, fresh bituminous coal would be put on to the fire immediately before lowering the damper.

The draught required depends on the nature of the fuel, on the depth of fire maintained, on the rate of combustion desired, on the design of the boiler and flues and on the resistance of ancillary plant, such as economisers, air-heaters, regenerators or recuperators, waste heat boilers and so forth.

In view of these possible divergencies in practice it is difficult to give recommended figures. The following may be taken as fairly representing average practice for an internal-flued boiler, e.g. a Lancashire boiler :—

Chimney height feet	Draught Chimney base	Draught Bridge	Combustion rate coal/sq. ft./hr.
100–120	0·65″ w.g.	0·3″ w.g.	20 lb.
120–150	0·9″ ,,	0·4″ ,,	25 ,,
150–200	1·1″ ,,	0·5″ ,,	30 ,,

Some coals require more draught than others. Coke, anthracite and low-volatile coals require a strong draught. Much depends upon the resistance of the firebed. With low-volatile solid fuels nearly all the air for combustion must be drawn through the firebed. With higher-volatile fuels, a considerable proportion of the combustible material is distilled off and is burnt with secondary air. Since secondary air is drawn in through openings which offer virtually no resistance to flow, the draught required to provide secondary air is a good deal less than that needed for primary air.

Low rank coals require a high draught, since the volume of products of combustion per unit of heat generated is generally greater than for the higher rank coals.

The resistance of the fuel bed has much to do with the effective utilisation of draught. The ideal is that air shall be uniformly drawn through the fire, at the same rate over each square foot of the grate, though this may be modified in chain-grate stokers where less air is needed at the extreme end of the travel.

Any segregation of the fuel on the bed will cause more air to pass through where the coal is larger, and will thus result in uneven combustion and possibly in unconsumed air passing through some portions of the grate owing to its high velocity. In effect, this will also reduce the output from the plant, since those parts of the grate where the resistance is high are not being fully utilised.

Uneven thickness of the fire will have the same effect as segregation, and if the unevenness is so great as to cause holes and bare spots, much excess air will flow through these places, there being little or no resistance to flow.

Special care should be taken when the combustion space is under suction that the fire-doors are not opened more than is necessary and that the secondary air ports or dampers are correctly adjusted. The lack of resistance to the flow of secondary air has already been mentioned ; it is probable that more inefficiency occurs through lack of correct proportioning of primary and secondary air than through any other single factor in boiler plants.

Accumulations of ash, and particularly the formation of large masses of clinker, adversely affect the distribution of air, and increase the resistance of the firebed. Caking of coals on the grate also has a similar effect. Large masses of coke formed on combustion should be broken up.

A high fire-bridge will cause unnecessary resistance. The fire-bridge should be no higher than is required to keep the coal on the grate and should be streamlined, back and front.

The salient features of effective utilisation of draught may be summarised thus :—

(1) The available draught is seriously diminished by the introduction of unnecessary excess air. If this excess air leaks in through the brickwork and is cold, the adverse effect *on the draught* is even worse than when it has passed through the furnace and is hot.

(2) Resistances to the flow of gas in the furnace and boiler setting should be watched carefully to ensure that they do not rise beyond reasonable limits.

(3) The objective of draught is to bring the combustion air to the right place, at the right time, and in the proper quantity. The resistance of the fuel bed has a vital bearing on this. Precautions should be taken to avoid the introduction of air at places where it cannot take part in combustion, and to maintain the resistance of the fuel bed uniform over the whole grate area.

(4) The earlier in the system that excess air leaks in (excluding excess air which passes through the fire), the worse is its effect *on heat transmission* ; the cooler gases transmit heat much less readily than the hotter gases obtained with less excess air.

CHAPTER VII

PROPERTIES AND PRINCIPLES OF UTILISATION OF STEAM

Pressure—Formation of steam—Work done by steam—Selection of boiler pressures—Superheated steam for power production and for process work—Use of steam in process work—Pipe lines—Return of condensate.

THE greater proportion of the coal mined in this country is used for the purpose of heating water. For purposes such as central heating and many process operations the hot water is used directly. For many other purposes the water is converted into steam, which is used either for the production of power in prime movers or for the transference of heat for process work. The steam used in process work may either be indirect steam which does not come into contact with the processed liquid, or it may be direct steam blown through the liquid.

PRESSURE

A distinction must be made between absolute and gauge pressure. A pressure gauge is a means of comparing the pressure in a vessel or pipe with the pressure of the surrounding atmosphere. A gauge freely exposed on all sides to the atmosphere reads zero. If the gauge is attached to a vessel and reads, say, 30 lb. per square inch, the pressure recorded as existing in the vessel, i.e. the " gauge pressure," is 30 lb. per square inch above that of the atmosphere.

The atmospheric pressure is measured by a barometer which gives the height of a column of mercury producing the same pressure as the atmosphere. Since the density of mercury at 0° C. is 0·49 lb. per cubic inch the barometric height in inches multiplied by 0·49 gives the corresponding pressure in lb. per square inch.

The barometric height for " normal " atmospheric pressure is 760 mm. (29·92 inches) of mercury which is equivalent to 14·7 lb. per square inch. Unless the barometer reading differs widely from " normal," or a high degree of accuracy in pressure is required, the absolute pressure can be obtained by adding 14·7 to the gauge pressure in lb. per square inch.

Zero on the scale of absolute pressure is a perfect vacuum. Normal atmospheric pressure is 14·7 lb. per square inch abs. Thus in lb. per square inch :—

Abs. pressure	0	10	14·7	100	114·7	214·7
Gauge pressure	29·92" vac.	9·5" vac.	0	85·3	100	200

STEAM FORMATION

Steam is produced by supplying heat to water and raising its temperature until it reaches the boiling point at which temperature the water gradually changes from the liquid state to vapour in the process known as boiling. The temperature at which boiling takes place depends upon the pressure under which the steam is made. If this pressure be " normal " atmospheric, 14·7 lb. per square inch absolute, or 29·92 inches mercury barometer, the temperature is 212° F. ; if the pressure is above normal atmospheric the temperature will be above 212° F., and if the pressure is lower than normal atmospheric, as at high altitudes or under a vacuum, the water will boil at a temperature below 212° F.

The water is fed to the boiler at a temperature determined by local circumstances, but it is clear that in general the feed temperature cannot be less than 32° F. at which temperature water solidifies (at atmospheric pressure) and

therefore by general consent all heat quantities contained in the steam are based on water at 32° F. which is taken as the starting point, i.e. the heat content of water at 32° F. is arbitrarily regarded as zero.

The heat required to make steam obviously depends on the quantity of steam to be made and the unit quantity in this country is the pound. If the British Thermal Unit is chosen for the unit of heat, it follows that all heat quantities in the steam will be expressed as B.Th.U. per lb.

The formation of steam consists of three operations :—

(1) Forcing the water into the boiler against the boiler pressure.
(2) Warming the water until it reaches the temperature at which it boils.
(3) Evaporation, by which the water is changed into steam.

These three operations are essential for the production of steam of any quality, but often there is a fourth operation in which the steam is withdrawn from the boiler and heated further in order to superheat it.

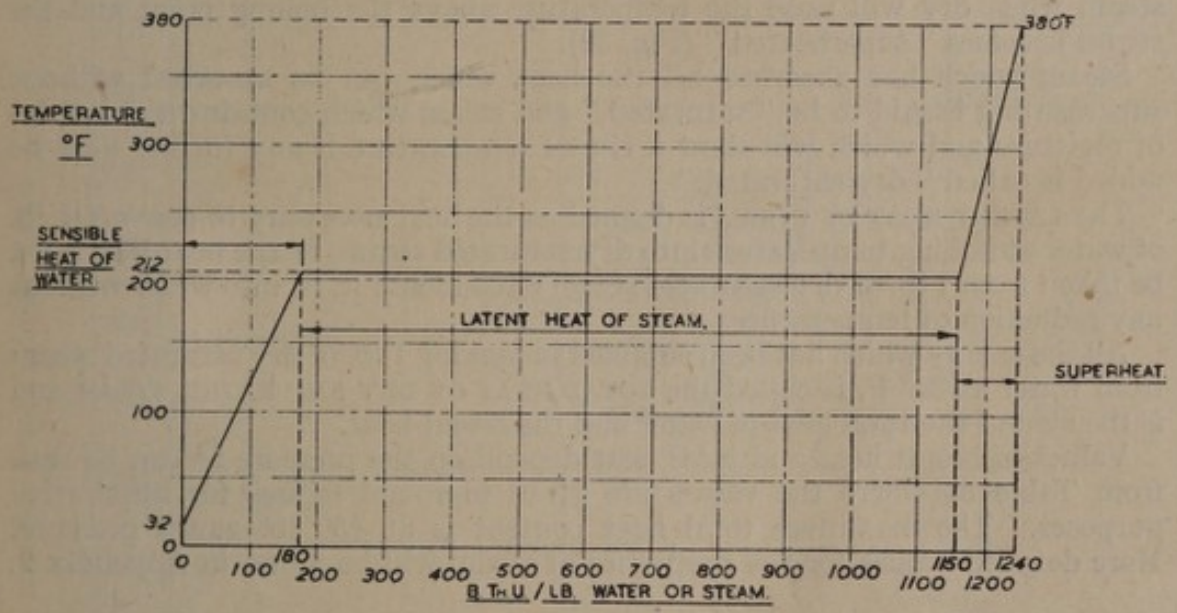

FIG. 32. Heat quantities involved in the formation of steam at atmospheric pressure.

The first operation is performed by the feed pump or injector. The energy required for this is small (the volume of 1 lb. of water being only about 0·016 cubic foot), and although in the aggregate it may be considerable, it is generally less than 1 B.Th.U. per lb.

The second and third operations occur within the boiler proper. Starting from the agreed datum temperature of 32° F., the steady application of heat produces a steady rise in the temperature of the water. If the water is mixed so that the heat supplied is uniformly distributed throughout the pound of water and the temperature is uniform, the absorption of 1 B.Th.U. raises the temperature by 1° F., the specific heat of water being unity for moderate ranges of temperature.

The temperature continues to rise with the absorption of heat until the water reaches the temperature at which it boils ; this temperature, as has been mentioned above, depends upon the pressure in the boiler. If the pressure be atmospheric this temperature is 212° F. The 180 B.Th.U. which have been supplied will have produced the 180° rise in temperature from 32° to 212° F. (see Fig. 32). This heat is apparent to our senses—to our sense of warmth if we put our hand in the water, to our sense of sight if we use a thermometer. It is therefore called " SENSIBLE HEAT."

The energy supplied by the pump added to the sensible heat gives the total

amount of energy which has so far been given to the water. This quantity is called the " TOTAL HEAT OF WATER." Values of the total heat of water for different pressures and also the temperatures of boiling are given in Table 33.

When the water has reached the boiling temperature a further absorption of heat does not cause a further rise in temperature, but the water is gradually changed into steam (Fig. 32). This change in physical state from liquid to vapour may require the absorption of many heat units, but it takes place at constant temperature and therefore all this heat is hidden in the steam without producing any rise in temperature. Heat thus utilised is called " LATENT."

The heat thus rendered latent at normal atmospheric pressure is 970 B.Th.U. per lb., but the latent heat varies considerably with the pressure as shown in Table 33.

The vapour may contain drops of liquid in suspension in which case the steam is called " wet." When the last particle of liquid in the steam is converted into vapour the steam is called " dry." Any further addition of heat to the steam when dry will raise the temperature above the boiling point and the steam becomes " superheated " (Fig. 32).

Steam which has absorbed all the heat which can be absorbed without superheating is said to be " saturated " and steam which contains no particles of moisture and which will show a rise of temperature if any further heat be added is called " dry saturated."

The LATENT HEAT OF STEAM is defined as the heat necessary to convert 1 lb. of water at boiling temperature into dry saturated steam, or the heat that must be taken from 1 lb. of dry saturated steam to condense it all into water without any reduction of temperature.

All the energy which has been supplied in making 1 lb. of dry saturated steam from water at 32° F. is called the TOTAL HEAT OF DRY SATURATED STEAM and is the sum of the total heat of water and the latent heat.

Values of latent heat and total heat depend on the pressure as can be seen from Table 33 where the values are given in round figures for illustrative purposes. The maximum total heat content is at 430 lb. gauge pressure. More detailed values for the properties of steam will be found in Appendix 2.

TABLE 33. PROPERTIES OF DRY SATURATED STEAM

Pressure ins. vacuum and lb./sq. in. gauge	Temperature of boiling ° F.	Heat in water B.Th.U./lb.	Latent heat of steam B.Th.U./lb.	Total heat B.Th.U./lb.
29·0″	77	45	1,050	1,095
27·0″	114	82	1,029	1,111
25·0″	133	101	1,018	1,119
20·0″	161	129	1,002	1,131
15·0″	179	147	991	1,138
10·0″	192	160	983	1,143
5·0″	203	171	976	1,147
Atmos.	212	180	970	1,150
100 lb. sq. in. ..	338	309	880	1,190
150 ..	366	338	857	1,196
200 ..	388	362	837	1,199
250 ..	406	382	820	1,202
430 ..	455	436	769	1,205
500 ..	470	453	751	1,204
1,000 ..	546	545	646	1,191
1,500 ..	597	614	553	1,167
2,000 ..	636	673	461	1,134
3,191·5 ..	705·4	902·7	0	902·7

The formation of steam under various pressures may be represented diagrammatically as in Fig. 33, in which heat in B.Th.U. is plotted against temperature in °F.

Suppose it is desired to make a pound of steam at atmospheric pressure (14·7 lb. per square inch absolute). Starting from the origin at 32° F. and zero heat supplied, the energy required to force 1 lb. of water into the boiler is too small to show in the diagram and may be neglected. The supply of 180 B.Th.U. raises the temperature of the water to the boiling point (212° F.) as represented by the line Oa. The change in physical state from water at boiling point to dry saturated steam requires 970 B.Th.U. This operation takes place at 212° F. and is depicted by a line aA which represents 970 B.Th.U. in length. The point A is therefore at 212° F. and 1,150 B.Th.U. The point a gives the total heat of

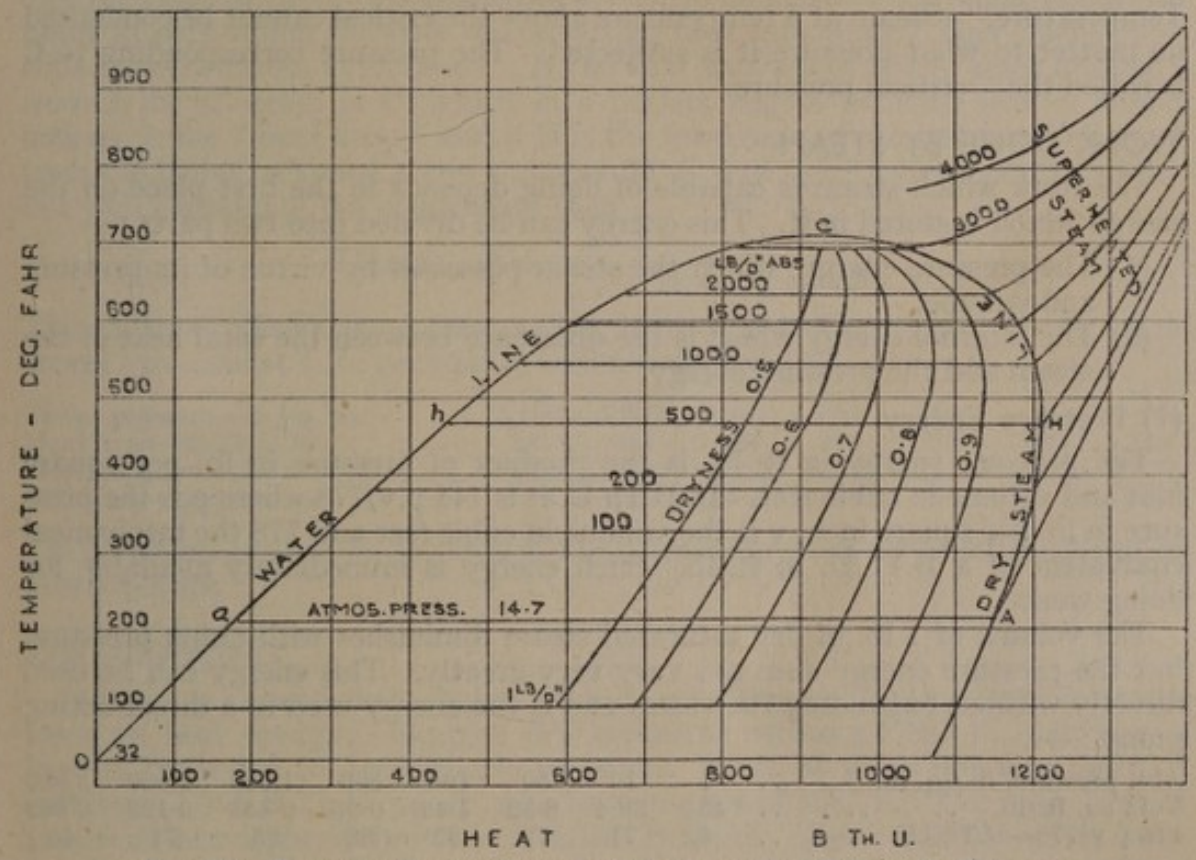

FIG. 33. Pressure, temperature, heat per pound and dryness of steam.

water, the point A the total heat of dry saturated steam and the length aA to scale gives the latent heat, all three for atmospheric pressure.

Next suppose it is desired to make steam at 500 lb. per square inch absolute pressure for which the boiling temperature is 467° F. The energy necessary to push the pound of water into the boiler is less than 1½ B.Th.U. and can hardly be represented on the diagram. The warming operation therefore starts from O and passes up through the point a to the point h, which has a value of 450 B.Th.U. at 467° F. The latent heat at this temperature is 755 B.Th.U., giving the line hH and the point H which is at 1,204 B.Th.U. If this process be performed for the whole range of pressures given in Table 33 the complete curve OahCHA is obtained.

Reference has been made to wet steam which contains particles of liquid in suspension. The amount of moisture is found from the DRYNESS FRACTION of the steam which is defined as the fraction of the steam which is "dry." Thus 1 lb. of steam having a dryness fraction q consists of q lb. of dry steam

and 1—q lb. of liquid in suspension. Steam at 500 lb. per square inch and dryness 0·5 is represented by a point midway between h and H. Points at the same dryness can be joined together giving dryness lines as in the diagram.

SUPERHEATED STEAM is steam which has absorbed more heat than is necessary to make it dry saturated and which therefore has a temperature above the boiling point. The specific heat of steam depends on pressure and temperature, rising with increase of pressure and falling with increase of temperature. At pressures near atmospheric the specific heat is practically constant at 0·48 but at 500 lb. per square inch absolute the variation is from 0·74 at 480° F. to 0·51 at 1,000° F. Lines representing superheated steam are shown on Fig. 33.

It will be observed from the diagram that the water line OahC and the dry steam line AHC run together at the point C. At this point the latent heat of steam vanishes and the temperature of the point C is called the " Critical Temperature." Steam at a temperature above the critical cannot be condensed no matter to what pressure it is subjected. The pressure corresponding to C is called the " critical pressure."

WORK DONE BY STEAM

The work which steam is capable of doing depends in the first place on the energy which is stored in it. This energy can be divided into two parts :—

(1) The pressure energy, which the steam possesses by virtue of its pressure and volume.
(2) The internal energy which is the difference between the total heat of the steam and the pressure energy.

(1) Pressure Energy

The pressure energy in ft.-lb. is the product of pressure in lb. per square foot and volume in cubic feet. In B.Th.U. it is 144 p.v/778 where p is the pressure in lb. per square inch, v is the volume in cubic feet and 778 the mechanical equivalent of a B.Th.U. in ft.-lb. Such energy is immediately available for doing work.

The volume of 1 lb. of dry saturated steam diminishes with rising pressure, but the pressure energy does not vary very greatly. This energy can be used directly without expanding the steam and is the energy used in a direct-acting pump :—

Abs. pressure, lb./sq. in. ..	1	10	50	100	500	1,000	2,000	3,000
Vol., cu. ft./lb.	334	38·4	8·52	4·43	0·93	0·458	0·193	0·083
144 p.v/778—B.Th.U.	62	71	79	82	86	85	71	46

(2) Expansion Energy

By far the greater part of the energy is stored as internal energy. Much of this energy can be utilised by allowing the pressure of the steam to drop and the total amount of work which can be obtained theoretically by dropping the pressure of the steam is called HEAT DROP. It is beyond the scope of this publication to explain how such heat drops can be determined, but the following figures show how the heat drop depends on pressure drop. If dry saturated steam at 500 lb. per square inch absolute is expanded in a perfect engine to p lb. per square inch absolute the heat drops are as follows :—

p	500	100	50	14·7	10	3	1	
Heat drop.. ..	0	129	175	253	272	339	391	B.Th.U. per pound

If dry saturated steam at pressure p lb. per square inch absolute is expanded to 1 lb. per square inch absolute the heat drops in B.Th.U. per lb. (cf. Appendix 3) are :—

p	3,000	1,500	1,000	500	100	50	10	5	1
Heat drop.. ..	381	431	423	391	293	248	140	99	0

The work done by 1 lb. of steam is thus seen to depend on the pressures of supply and exhaust.

The number of B.Th.U. required per lb. of steam reckoned from water at 32° F. has already been considered and values have been given in Table 33, but if the steam is exhausted at atmospheric pressure its temperature will be 212° F., and it should be possible, theoretically at any rate, to return to the boiler the lb. of condensate at or near this temperature and thus reduce the number of B.Th.U. required per lb. of steam by 180. If the plant were a condensing plant with a vacuum of 28 inches mercury for which the temperature of condensation is 101° F. the reduction would be 101 − 32 = 69 B.Th.U. per lb.

Two important quantities in a power plant are the amount of work obtained per lb. of steam and the number of heat units required to make that lb. of steam, and the ratio of these two quantities gives the measure of the efficiency of the plant in performing the task of converting heat into work. This efficiency is called the THERMAL EFFICIENCY. If the heat equivalent of the work obtained from 1 lb. of steam is U, which in a perfect engine without heat or steam leakage is the " heat drop," and if H is the total heat of the steam as delivered from the boiler reckoned above water at 32° F. and t_w is the temperature of the hot well containing the condensate then

$$\text{Thermal efficiency} = \frac{U}{H - (t_w - 32)}$$

The thermal efficiencies corresponding to the heat drops given above when steam exhausts at 1 lb. per square inch absolute are :—

Boiler pressure—lb./sq. in. ..	3,000	1,500	1,000	500	100	50	10
Heat drop—B.Th.U./lb.	381	431	423	391	293	248	140
Heat supplied—B.Th.U./lb. ..	970	1,113	1,135	1,134	1,117	1,107	1,073
Efficiency—percentage	39·3	38·7	37·3	34·6	26·3	22·4	13·1

These figures explain in part the modern tendency to raise boiler pressures in power plants.

WIREDRAWING

If steam is expanded through a valve or orifice it does no work and therefore loses no heat energy. Suppose dry saturated steam at 100 lb. per square inch is expanded to atmospheric pressure. Table 33 gives the total heat at 100 lb. as 1,190 B.Th.U. per lb. at a temperature of 338° F. The total heat of steam at atmospheric pressure is 1,150 B.Th.U. There will therefore be a surplus heat of 39 B.Th.U. in the expanded steam, which can only superheat the steam. At atmospheric pressure and moderate temperature, the specific heat of steam may be taken as 0·49, so that there will be 80° F. of superheat temperature. The expanded steam will thus have a temperature of 292° F. Expansion through a valve or orifice when no work is done is called " throttling " or " wiredrawing."

UTILISATION OF STEAM

Steam is required for three primary purposes :—

(1) For the production of power.
(2) For use as a medium for transmitting heat for process work.
(3) As a physical or chemical agent in industrial operations.

The same steam may be used in sequence for more than one of these operations. Thus, it may be generated under high pressure and used first for the production of power in pass-out turbines, the exhaust steam at a suitable temperature being then used for heating or for chemical purposes.

As examples of the third use of steam may be cited the use of direct steam in distillation (a physical use) and the use of steam in gas producers or in the production of hydrogen by the iron-steam process (chemical uses). Steam used for cooling fire-grates or preventing clinker formation comes under this third category.

SELECTION OF BOILER PRESSURE

In any use of steam a decision must be taken as to the pressure at which the steam should be generated, the pressure at which it is to be used and whether it should be saturated or superheated.

(*a*) *For Power Production.* Some indication has been given that the modern tendency in large power generating plants is to increase the boiler pressure but, as efficiency depends upon other factors as well as boiler pressure, it is impossible to give any simple rule for the selection of boiler pressure for a particular plant. Power plants of over 50,000 kW capacity, both in this country and in America, work on pressures varying between 1,900 and 265 lb. per square inch. In almost every case there is a high degree of superheat, the temperature of the steam being between 650° and 950° F. For the higher pressure plants some degree of reheating the steam after partial expansion may be necessary, otherwise the steam becomes too wet during expansion.

The degree of superheating, reheating and feed heating all affect the efficiency of the plant and make any suggestion of optimum pressure misleading.

Higher pressures can be used in turbine plants than in reciprocating engines because turbine plants do not limit the ratio of expansion which can be usefully employed and, furthermore, are specially adapted for using large volumes of steam such as must be dealt with if the steam is expanded to low pressures which is an essential condition for high efficiency. Reciprocating engines, even if compounded in two or three cylinders, impose a limit on expansion which turbines do not. It is therefore not possible to work over so wide a pressure drop in a reciprocating engine as in a turbine.

The type of prime mover employed may modify the form of heating surface required to utilise the heat in the products of combustion, e.g. from a boiler. Thus in the modern high efficiency power plant a form of feed water regeneration is incorporated, using steam bled from the main turbine.

In such a plant the feed temperature may nearly approach the temperature of the water in the boiler and the boiler heating surface proper has then only to transmit the latent heat of steam to the heated water. Much of the heat that would have been absorbed from the hot gases by a feed water economiser can now be utilised for preheating the air supplied to the furnace.

This necessitates the provision of air-heating equipment, but since the upper limiting temperature of the air supplied to a mechanical stoker (if operating troubles are to be avoided), is about 350° F., it is easy to conceive circumstances in which the air temperature must be allowed to exceed this, or a high gas outlet temperature be permitted. Pulverised fuel burners, however, can use hot air up to temperatures of 400°–500° F., and when power plant design conditions arise as depicted above pulverised fuel firing is a rational solution of the problem.

(*b*) *For Process Work.* For process purposes, steam should not be generated at a higher pressure than is dictated by the temperature at which it is required to give up its heat. Additional costs are involved in operating at higher pressures, both in the design of the boiler and in increased plant to recover heat from the flue gases which of necessity must leave the boiler at a higher temperature. It is thus desirable to select the lowest boiler pressure for process work consistent with securing an adequate steam temperature.

Advantage can, however, be taken of higher steam pressures and tempera-

tures to provide power as a by-product. If the steam is generated at elevated pressures (200 lb. per square inch, for example) the excess pressure can be used to generate power, leaving the exhaust, which may still be at 30 lb. pressure, for use in process work.

SUPERHEATED STEAM FOR POWER PURPOSES

For reciprocating engines, one reason for superheating steam is to reduce condensation in the cylinder. The temperature of the steam supplied to a reciprocating engine should therefore be as high as is permitted by the practical consideration of lubrication. Modern lubrication technique has removed in a very large measure the essential difficulty of the use of superheated steam in reciprocating engines.

When the exhaust steam from engines is led to a process for which it is essential that the steam must be clean, the steam must reach the engine in the saturated state, since any superheat, however small, demands lubrication.

In turbines it is possible to allow the steam to expand more than is practicable in a reciprocating engine. This means that the exhaust temperature will be lower and if saturated steam had been used much of the steam would have condensed. It has been found by many years of experience that 13 per cent. wetness at exhaust is the greatest amount of condensation that is permissible without undue blade erosion. Whatever pressure is selected, sufficient superheat should be added to prevent the steam leaving the last row of turbine blades more than 13 per cent. wet.

Expert advice on this subject can be obtained from the manufacturers of power station boilers and of turbines. Much useful information is also available in the Proceedings or Transactions of the Technical Institutions.

As a general statement the saving in fuel in prime movers due to the first 100° F. of superheat may be of the order of 6 or 7 per cent.; the saving due to the next 100° F. of superheat will be somewhat less, say 4 to 5 per cent. and so on.

The maximum superheat temperature needed by those industrial plants exhausting from turbine to process is about 750°–800° F., though much higher temperatures have been used in power stations.

The primary saving in fuel by the use of superheated steam is quite considerable when the steam is used for power generation. The quantity of steam for a given output of power is considerably reduced. There is, therefore, less work to be done by the feed pump, by the condenser and by the cooling water circulating pump.

Relative average steam consumptions when using saturated steam and superheats of 100° F. and 200° F. are given in Table 34.

TABLE 34

Type of engine	Steam consumption—lb. per I.H.P./hour.		
	Saturated steam	100° Superheat	200° Superheat
Simple non-condensing	29–45	20–38	18–35
Simple non-condensing Corliss ..	26–35	18–30	—
Compound non-condensing	19–28	15–25	13–22
Compound condensing	13–22	10–20	9–17
Simple duplex steam pumps ..	120–200	80–160	—
Turbines non-condensing (kWh.) ..	28–60	24–54	21–48
Turbines condensing (kWh.) ..	12–42	10–38	9–34
Simple non-condensing locomotive..	27	23½	19·5

Reciprocating engines and turbines show a reduction in steam consumption of about 1 per cent. for every 10°–15° F. of superheat for a given power developed. This is not an equivalent gain in overall thermal efficiency (i.e. it is not an equivalent saving in fuel) because superheated steam requires more heat to be put in it initially.

For power generation substantial advantages have thus been shown to be effected by the use of superheated steam.

The superheater may also be a means of increasing boiler efficiency in various ways: firstly by reducing the stack temperature due to the abstraction of heat in the superheater and secondly by increasing the prime mover efficiency, and thereby reducing the work to be done by the boiler plant for a given output of power. This second effect will be particularly valuable if the boiler is overloaded when operated without a superheater. Thirdly, the boiler auxiliaries will have less work to do for a given output.

THE USE OF STEAM IN PROCESS WORK

The object of steam-heated process plant is to transfer a definite quantity of heat from the steam to the material being processed. It is usually essential that the temperature of the processed material be accurately maintained within close limits.

Process heating can be divided into the following broad groups :—

(1) Evaporation.
(2) Raising the temperature of a processed substance :
  (*a*) By means of heating surface.
  (*b*) By direct contact.
(3) Maintaining the temperature of a processed substance by compensating for the heat lost by radiation, etc., and absorbed by endothermic reactions :
  (*a*) By means of heating surface.
  (*b*) By direct contact.
(4) Distillation.

(1) Evaporation

Evaporation for the purpose of concentrating a solution is a widely used industrial process requiring much steam. It differs from distillation in that its goal is the production of a concentrated solution, whereas the aim in distillation is the production of condensed distillate.

The steam is required to supply heat to the process. This heat consists of :—

(1) Heat required for warming the liquid up to the temperature of evaporation, and
(2) Heat required for evaporation.

The liquid may be warmed either by live steam or by exhaust from the evaporator, but the live steam applied to the evaporator will have to provide not only the latent heat of the liquid evaporated but heat lost by radiation, etc., from the outer surface of the plant.

From the principles of heat transmission (Chapter VIII) it will be evident that there must be some temperature difference between the (indirect) heating steam and the liquid heated, if there is to be an adequate rate of heat transfer. The temperature of the steam must, therefore, always be a few degrees (say 5°–20° F.) higher than that of the boiling liquid. If this liquid is water, it follows that the pressure of the steam must be greater than the pressure at which evaporation is taking place.

The total heat of the condensate is above the total heat of water at atmo-

spheric pressure, and therefore some of the water will change to steam when the pressure on the condensate is reduced to atmospheric.

Whether the steam used for evaporation, or for any form of indirect heating, should be saturated or superheated depends on the principles of heat transmission. When saturated steam unmixed with any permanent gas is fed into the heating jacket or tubes of a vessel, the steam comes in direct contact with the heating surface. It is immediately chilled, loses its latent heat and is condensed. The film of water runs down the surface and more steam immediately flows in to take the place of that which has been condensed. Thus the steam is condensing in contact with the water film adhering to the heating surface and heat transfer is rapid.

If the steam is mixed with permanent gas, there is set up a gaseous film in streamline motion between the film of condensed water and the main body of the steam. The steam must diffuse through this gas layer to reach the surface (a very slow process) or heat must be conducted through the gas layer ; gases are bad conductors of heat. The presence of gas therefore slows down the rate of heat transfer seriously.

If the vessel is heated by a permanent gas the streamline gaseous layer is still present (see Chapters VIII and IX) and heat must be transmitted through it—a slow and inefficient process. Superheated steam behaves as a permanent gas and is thus undesirable for most operations in which steam is used for indirect heating. This is discussed in more detail in Chapter XXIII.

The vapour which leaves an evaporator carries away its latent heat and in many plants this latent heat is lost. The vapour is condensed in a separate condenser and the heat in the vapour is transferred to the cooling water or to the surrounding air if the condenser is air cooled. It is possible sometimes to conserve this heat by using a second evaporator as a condenser to the first, a third as condenser to the second, and so on until the heat has been reduced to such a low temperature that it is no longer economically useful. This is known as multiple effect evaporation.

The essential of multiple effect evaporation is that each evaporator must evaporate at a temperature lower than that of the heat supplied to it. This may be illustrated by assuming that each in a series of evaporators is evaporating water. The necessary drop in temperature through the series then demands a corresponding drop in pressure.

To illustrate the principle, let it be assumed that steam is supplied to an evaporator at 100 lb. absolute pressure ; it then has a temperature of 328° F. and a latent heat (available for heating in the evaporator) of 889 B.Th.U.

If the evaporator operates at 312° F. under a pressure of 80 lb., the latent heat of the water to be evaporated is 901 B.Th.U. per lb.

Weight of steam at 100 lb. required to provide 901 B.Th.U.

$$= 901/889 = 1{\cdot}013 \text{ lb.}$$

Heat contained in the vapour leaving the evaporator is 901 B.Th.U. per lb.

Thus the expenditure of $1{\cdot}013 \times 889$ B.Th.U. has resulted in driving off 1 lb. of vapour having a latent heat content of 901 B.Th.U.

This pound of vapour can be passed forward as the heating agent in a second evaporator. If this second evaporator works at 40 lb. pressure, the conditions are :—

pressure	40 lb.
temperature	267° F.
latent heat	934 B.Th.U.

The available heat, namely 901 B.Th.U., can then evaporate in the second evaporator, 901/934, or 0·965 lb. of water. This has been effected without any additional heat, so that the heat in the original steam has virtually been used twice over.

The 901 B.Th.U. now in the form of steam at 40 lb. pressure can then be passed to a third evaporator, and so on.

In this illustration all losses have been neglected and it has been assumed that the heat supplied in evaporation is fully returned in the subsequent condensation and can be used again without loss. Actually small losses will occur, but this principle of multiple effect evaporation, well known in chemical engineering, may be very valuable.

If the circulation of the process material is sluggish and there is a considerable depth of liquid in an evaporator, it is possible for the lower layers of material to be raised to a temperature considerably in excess of the upper layers. This is due to the hydrostatic head of the liquid exerting a pressure on the liquid in the bottom of the vessel.

It might be thought that a high temperature on the steam side of the heating surface would increase this danger, but often this is not so. With a high temperature on the steam side the higher rate of heat transfer increases the circulation of the boiling liquid due first to convection currents and second to the mechanical drag of the bubbles of vapour acting on the liquid.

This increased circulation has beneficial effects. It greatly minimises the danger of bottom overheating because with violent circulation local stagnation is unlikely. By increasing the rate of flow over the heating surface the rate of heat transfer is much increased. The fact that the liquid is full of bubbles clearly reduces its mean specific gravity. A given depth of violently boiling liquid will therefore exert a lower hydrostatic pressure than the same depth which is boiling sedately.

(2) Raising the Temperature of a Processed Substance

(*a*) *By means of a Heating Surface.* The temperature of a substance can be raised by means of a heated surface with which it is in contact. The discussion relating to evaporators applies here equally with an important addition. By the use of saturated steam a limit can be set to the temperature to which the material can be raised if, by negligence, the steam is left on too long. With superheated steam the temperature of the heating surface is no higher than with saturated steam during the temperature raising process, but, whereas with saturated steam the heating will stop when the temperature has risen to the saturated steam temperature, when using superheated steam it can rise, possibly undesirably, to the temperature of the superheated steam. Superheated steam is here clearly not to be recommended.

(*b*) *By Direct Contact.* Many processes use steam injectors to raise the temperature of the process substance. There may sometimes be a legitimate use here for superheated steam. For a given quantity of water superheated steam will bring in more B.Th.U. and this may be important. On the other hand, some parts of the material will be in contact with and be temporarily raised to the temperature of the superheated steam, with possibly detrimental results.

Superheated steam blown into a liquid may have insufficient time to desuperheat itself and condense, so that some breaks the surface and is wasted. This is particularly likely to happen when medium pressure steam is blown into a shallow vat or tank. If dry, saturated steam at 30 lb. per square inch is blown into a vat containing 4 feet 6 inches of liquid, there will be a pressure drop in the blower of 28 lb., and the steam will be superheated by 38° F. It is very unlikely that 4 feet 6 inches of hot liquid can remove 38° of superheat and condense the steam during its passage through the liquor.

(3) Maintaining the Temperature of a Processed Material

The principles are the same as those just stated, except that the reasons for

not using superheated steam are even stronger. Maintenance of temperature means maintenance of temperature and a plant that can produce overheating by the use of superheated steam is clearly unsatisfactory.

From the foregoing it can be stated quite categorically that superheated steam for most process purposes is not only of no benefit but may be undesirable.

There is one possible case where superheated steam might usefully be applied. This is in a piece of process plant where only the superheat is used, the steam passing through the plant into the process main. This enables one piece of high temperature plant to operate in conjunction with a much larger amount of low temperature plant on a single low pressure steam supply.

(4) Distillation

(*a*) *By indirect Steam.* This is similar in general characteristics to evaporation. The heat provided by the steam must be sufficient to heat the liquid to its boiling point (if this has not been already done outside the still), to supply the latent heat of evaporation, and to compensate for any losses of heat that may occur.

The pressure of the steam should be just high enough to provide steam at the temperature at which the liquid boils, plus an additional 5°, 10°, or 20° F., according to circumstances to provide the necessary temperature gradient for the passage of heat (cf. Chapter VIII).

(*b*) *By Open Steam.* Direct steam is often used for distillation, steam being passed directly through a liquid so that the more volatile constituents therein are distilled at a temperature below their normal boiling point. The steam and the distillate vapour each exert pressure, and the mixture boils when the sum of the partial pressures equals the external pressure. Generally, indirect heat is also supplied by means of jackets or coils, or by preheating the feed up to its boiling point.

Substances of high boiling point may thus be distilled at relatively low temperature ; aniline, for example, boils at 180° C. (356° F.) at 760 mm., but distils in a current of steam at 98·5° C. (210° F.) at the same pressure.

The pressure may be lowered by working under a vacuum with further consequent reduction of the temperature at which distillation occurs.

Steam distillation yields a condensate consisting of water and the volatilised substance or substances in proportions dependent both on their respective vapour pressures and molecular weights.

$$\text{i.e. at } t^\circ \text{ C., } \frac{\text{Weight of steam in vapour}}{\text{Weight of volatile substances in vapour}}$$

$$= \frac{(\text{Vapour pressure of steam at } t^\circ \text{ C.}) \times \text{mol. wt. of steam}}{(\text{Vapour pressure of vol. subs. at } t^\circ \text{ C.}) \times \text{mol. wt. of vol. subs.}}$$

As general examples may be cited the stripping of crude benzole from wash oil in the carbonisation industry, the deodorisation of fatty oils for the food industry, the steam distillation of fatty acids, essential oils, aniline and other organic substances.

Steam distillation is particularly useful in processing materials subject to decomposition at temperatures approaching their normal boiling point. Excluding a special case dealt with in Chapter XXIII the substance to be steam distilled must be immiscible with water.

The steam performs a dual function.

I Its presence lowers the partial pressures of the substances distilled, thus reducing their effective boiling point and so facilitating distillation.

II The sensible heat in the steam above the effective boiling point of the mixture serves to supply some of the latent heat required to volatilise the constituents to be distilled. Normally a considerably greater amount of latent heat is required than is available from the open steam and, as indicated above, this is supplied indirectly.

Steam distillation operations may be divided into two classes :—

(1) Where the boiling mixture in the still contains liquid water.
(2) Where the steam is not allowed to condense in the still.

There is obviously no economic advantage in the use of superheated as opposed to saturated steam when liquid water exists in the still. There are, however, some instances when condensation of water is not permissible (class 2) where superheated steam may be advantageously employed as the superheat permits a higher temperature to be attained at the point of primary vaporisation.

Examples indicating the conditions and the lines upon which the economics of the use of direct steam for distillation may be considered are given under " Distillation " in Chapter XXIII.

PIPE LINES

Steam is conducted from the boiler to the place where it is used in pipe lines. Heat is lost from these pipes and in consequence superheated steam will lose part or all of its superheat, being ultimately converted into saturated steam, and saturated steam will become wet.

The radiation loss from steam pipe lines is mainly determined by the degree of insulation. The better the insulation the greater the economy due to reduction of heat loss. It is very important to insulate the flanges as well as the pipe itself.

If the steam be sufficiently superheated, it will reach the process plant dry. If initially it is dry and saturated, or wet, it will reach the process plant wet. For a given pipe the loss of heat is likely to be somewhat less with superheated steam than with saturated steam. There is thus some saving in superheating through reduced loss of heat in the transmission pipes, and there is a considerable saving in traps and their ancillary piping.

Two other advantages possessed by superheated steam are :—

(*a*) There is less wear and tear on valves and fittings generally.
(*b*) Speeds about 50 per cent. higher can be used in pipe lines with superheated steam than with saturated steam. In new plants this would mean smaller pipe lines ; in existing plants, the required quantity of heat can be conveyed through the pipe lines for a smaller drop in pressure.

CONDENSATE

The exhaust from prime movers or from process operations will consist of a mixture of steam and hot water at a temperature corresponding to the back pressure against which the exhaust is working.

If exhaust steam from prime movers is led directly into a water supply tank in order to heat the water, a back pressure is created which reduces the efficiency of the engine or turbine. Consequently, a condenser is frequently used for the purpose of condensing the steam and of reducing the back pressure. The quantity of heat abstracted in the condenser will depend upon the dryness fraction of the exhaust steam as well as upon its pressure.

If, for example, the exhaust from an engine is 70 per cent. dry at an exhaust pressure of 4 lb. per square inch absolute, the heat rejected will be in three forms * :—

* At 4 lb. abs. the boiling temperature of water is 153° F., the sensible heat in the water is 121 B.Th.U./lb. and the latent heat is 1,006 B.Th.U./lb.

Sensible heat in water at 153° F.—30 per cent. of 121 = 36 B.Th.U.
Latent heat in 70 per cent. steam—70 per cent. of 1,006 = 704 B.Th.U.
Sensible heat in water from the steam—70 per cent. of 121 = 85 B.Th.U.
Total = 825 B.Th.U. per lb. of exhaust

If the steam was generated without superheat at 100 lb. gauge pressure, it will originally have contained 1,190 B.Th.U. per lb. Nearly two-thirds of the heat in the steam would therefore be contained in the exhaust.

Similar considerations apply to the exhaust or hot water from process work. It is therefore of great importance that the heat contained in this exhaust shall be utilised.

When steam condenses in a pipe line or process vessel, the hot water must be removed without loss of steam. This is usually done by means of an automatic device called a steam trap.

The condensate and the heat it contains are valuable. It should, therefore, be returned to the boiler whenever possible. When it cannot be returned to the boiler, it should be used as hot process or washing water. If by any chance the condensate is corrosive, or if it contains oil, its use may be impossible, but even then the heat it contains can often be used by passing it through a heat exchanger.

FLASH

The hot water leaving steam traps may be at temperatures above 212° F., and at corresponding pressures above atmospheric. Thus, if steam is being used at a pressure of 100 lb. per square inch gauge, the water condensed from it at this pressure will be at 338° F. and will have a total heat above 32° F. of 309 B.Th.U. per lb. If it is to be run back into the boiler feed tank at atmospheric pressure its temperature cannot be above the boiling point of water at that pressure, namely 212° F. Heat must be abstracted from the water to cool it from 338° F. to 212° F. If the trap exhausts to atmosphere, i.e. discharges the water through a pipe direct to the boiler feed tank, part of the water will flash into steam and be lost. The heat thereby lost is as follows :—

Heat in 1 lb. of water at 338° F.	..	309 B.Th.U.
,, ,, ,, 212° F.	..	180 ,, ,,
Heat loss per lb. of exhaust	.. ..	129 ,, ,,

TABLE 35. FLASH STEAM

Condensate		Percentage condensate flashed off when pressure is reduced to :—					
Pressure lb/sq. in. gauge	Temp. ° F.	40 lb/ sq. in.	20 lb/ sq. in.	10 lb/ sq. in.	Atmos.	10" vac.	20" vac.
200	388	11·5	14·3	16·2	18·8	20·5	23·2
150	366	9·0	11·8	13·0	16·4	18·2	20·9
100	338	5·8	8·6	10·6	13·3	15·1	17·9
80	324	4·2	7·1	9·1	11·9	13·7	16·5
60	308	2·3	5·2	7·3	10·0	11·8	14·7
40	287	—	3·0	5·0	7·8	9·7	12·6
20	259	—	—	2·1	5·0	6·8	9·8
10	240	—	—	—	2·9	4·8	7·8
0	212	—	—	—	—	1·9	5·0

As the latent heat at 212° F. is 970 B.Th.U., a quantity of water will be evaporated as "flash" steam, amounting to 129/970 = 0·133 lb. per lb. of condensate. Some further data on this subject are contained in Table 35.

A loss from the evolution of flash steam arises wherever a steam trap is discharging above atmospheric pressure. The extent of the loss depends on the pressure and if, for example, the trap were working at 25 lb. gauge pressure, the loss would be 236 — 180 = 56 B.Th.U. per lb. of steam or water.

The loss can be avoided in various ways :—

(1) The water leaving the traps can be passed through a heat exchanger in counter-current with the make-up feed water.
(2) If low pressure steam can be utilised the water can be discharged to receivers connected to the low pressure steam supply, the flash steam being then utilised for the supply.
(3) The water may be collected in a tank under pressure in which the make-up feed is introduced to condense all flash steam.

The generation of flash steam by the reduction of pressure on hot water is used in steam accumulators (Chapter XXIII).

Another application of the release of heat by pressure reduction is in the cooling of liquids. If an aqueous liquor at, say, 200° F. is to be cooled to 150° F., this can be readily accomplished by spraying the hot liquor into an empty vessel attached to a condenser and vacuum pump—the vacuum being such as to correspond to a liquor boiling temperature of 150° F. At the same time as cooling there will be a small concentration which may save evaporation at a later stage in the process.

The practical utilisation of steam is discussed at greater length in Chapter XXIII.

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CHAPTER VIII

HEAT TRANSMISSION

Methods of heat transfer—Transmission of heat through a boiler plate—Conduction—Overall conduction coefficients—Conduction through composite walls and pipe coverings—Convection—Natural convection—Forced convection—Convection effects in banks of tubes and in single tubes—Heat transfer from superheated steam—Convection in liquids—Heat transfer from condensing vapours—Boiler circulation—Radiation—The laws of radiation—Radiation from a coke fire—Gas radiation, including methods of calculation—Radiation from luminous flames—Heat exchangers, regenerators and recuperators—Heat losses from external surfaces—Heat loss through a furnace wall—Practical aspects of heat transmission in boilers.

THE process of heating involves three stages. Firstly, the heat must be generated by efficient combustion as described in Chapters IV–VI. Secondly, the heat so generated must be transmitted as efficiently as possible to the heat-absorbing surfaces. Thirdly, the heat must be conserved in the heating appliance to do useful work, and its transmission to those parts of the appliance from which it may be dissipated must be reduced to a minimum. In addition to the loss of heat so occurring through the structure of the appliance and its foundations there are the combustion losses which have already been discussed.

MODES OF HEAT TRANSFER

Heat may be transferred in three different ways, by conduction, convection and radiation. All three methods depend upon a difference of temperature in the heat transmitting media, and heat is always transferred from the warmer to the cooler body.

Conduction. The simplest form of heat conduction takes place in a solid, say a boiler plate or a furnace wall. One surface may be hot and the other cooler. The quantity of heat conducted through the body in a given time is approximately directly increased by an increase in the difference of temperature between the hot and cold surfaces. It is further proportionately reduced by an increase in the thickness of the plate or wall through which the heat is being conducted. It is proportionately increased with the dimensions of the surface normal to the direction of flow, and finally the rate of heat conduction depends upon the nature of the material involved. The familiar reference to good conductors such as metals, and bad conductors such as firebrick, lagging compounds and boiler scale can be reduced to a form of expression suitable for use in heating problems, by stating the number of heat units (British Thermal Units) which can be conducted in one hour through a slab of the material 1 square foot in section and 1 foot thick when there is a difference of temperature of 1° F. between the hot and the cold face. The conduction must be effected when temperatures are steady, referred to as " in the steady state", and under these conditions the number of heat units conducted hourly is referred to as the " thermal conductivity " of the material. It is necessary to distinguish between the steady or uniform flow of heat and fluctuating flow, since in fluctuating flow heat may be given to, or taken from, the body of the slab. This arises from the fact that the material has a capacity for absorbing and storing heat, and the temperature gradient in the slab is accordingly disturbed by this capacity. In heating it absorbs heat ; in cooling heat is given up.

Heat transfer by conduction can take place in still liquids and gases, but this mode of heat transmission, apart from the stagnant film of gas collecting on solid surfaces, is of secondary importance in industrial practice.

Convection. In liquids and gases the moving particles transmit heat to one another by contact if they differ in temperature, and they transmit heat by

contact when they are brought against the surface of a solid. This transmission of heat by the motion of the fluid against a solid is referred to as convection transfer, since it is brought about by the flow or convection of the fluid. Where the motion of the fluid results from its natural buoyancy arising from heating, expansion and consequent fall of density, the phenomenon is described as free or natural convection. When the motion is maintained mechanically, as for example by a fan, it is referred to as forced convection.

Radiation. Conduction and convection represent the transfer of energy from particle to particle by direct contact. Energy is also emitted from a hot body and can be transmitted in the form of radiation to another separated from it by a suitable medium. The heat emitted from an open firebox door is a case in point. A hot solid body sends out radiation over a wide range of wave lengths from the smallest to the largest, but the intensity of the radiation differs greatly for the various wave lengths to which the human eye is sensitive. The wave length of violet light is about 4,000 Angstrom units, Å, and red light ends in the region of 7,000 Å, an Angstrom unit being 10^{-10} metres. Longer waves than 7,500 Å are known as the infra-red, and within 7,500 to 53,000 Å, falls the range of thermal radiation which is of practical importance in industrial heating. Radiation is also emitted by hot furnace gases.

If the phenomena of conduction and convection on the one hand are contrasted with thermal radiation on the other, it is found that the former are affected by temperature difference and very little by temperature level, whereas radiation increases rapidly with increase in temperature level. Accordingly at very low temperatures conduction and convection are the major contributors to the total heat transfer ; at very high temperatures radiation is the controlling factor.

Heat transmission by conduction occurs in the passage of heat through a boiler plate and its adherent resistant layer of soot and scale ; into the mass of the charge in a furnace ; and through the hearth, walls and crown of a furnace chamber or firebox.

Convection transfer is bound up ultimately with the mechanism of flow of fluids, which is discussed in detail in Chapter IX, and is an important factor in the heating of waste heat fire-tube boilers, and in any form of heat exchanger in which a fluid flows over the heat-exchanging surface, for example in superheaters, economisers and air heaters. The velocity of flow of the gases is of much significance in these conditions.

Radiation is universal in its occurrence, being operative in addition to natural convection from the relatively cool surface of a lagged steam pipe, as well as from the incandescent surface of a fuel bed or a refractory wall.

TRANSMISSION OF HEAT THROUGH A BOILER PLATE

It is now possible to construct a picture of what happens when a stream of hot combustion gases heats a boiler tube (Fig. 34). The boiler shell or tube does not consist merely of a plate of mild steel, but the surface adjacent to the hot gases is covered with a layer of soot and oxide of iron, and on the evaporation side is covered with a thin layer of boiler scale, which may consist mainly of mineral deposits from the boiler water.

The heat of the combustion gases is transmitted to the surface of the thin layer of soot and deposit by radiation and convection, since radiation is emitted by the hot gases, and the scrubbing action of the flowing gases further increases the total heat transfer. At the surface of the boiler tube the gases directly in contact with the layer of soot and scale are virtually stagnant by reason of the resistance to the flow of gas of the surface presented. There is thus formed a stagnant film of gas through which the heat must be transmitted by conduction. Since gases have a low thermal conductivity, this film offers a consider-

able resistance to the flow of heat, out of all proportion to its thickness (see Chapter IX).

Heat is again transmitted by conduction through the badly conducting fouling layer on the hot surface of the plate, then through the plate which is a good conductor and through the second fouling layer and finally to the water, where ebullition and evaporation of steam take place.

In boiling a liquid at a given pressure, as the temperature difference between heating surface and liquid is increased, the heat flux at first increases slowly, then more rapidly, goes through a maximum and subsequently decreases as the heat transfer surface becomes insulated with a film of vapour. The character of the heat transmission on the water side must accordingly be influenced by the convection conditions, and will thus depend on the character of the water

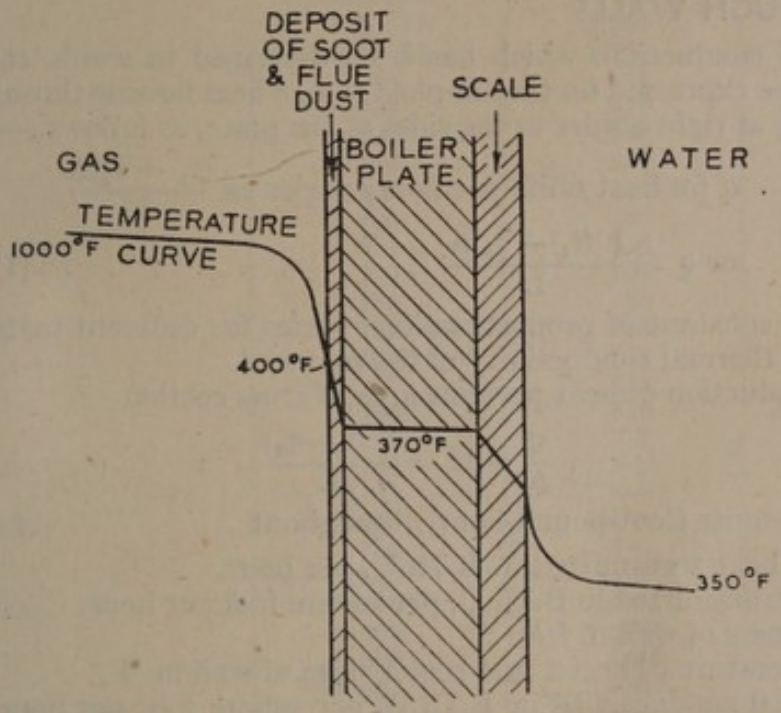

FIG. 34. Diagram illustrating heat resistances and their effects on temperature at a boiler heating surface. Scale may cause the temperature of the boiler plate to rise much above the figures stated on the diagram, particularly where the plate is subjected to direct radiation from the furnace.

circulation. Again a " film " factor on the water side of the plate becomes operative.

It will be appreciated in comparing the performance of various types of boilers and in investigating conditions in any one boiler that the individual layers of media more or less resistant to the heat flux cannot be considered independently, for the reason that the temperature differences which govern the heat transmission in the various layers cannot be known with any accuracy. Accordingly it has become the practice to use the concept of a " surface conductance " or " film " coefficient, which may be applied to include the stagnant layer of fluid and the fouling condition. The " surface conductance " then becomes a coefficient defining the rate of heat flux for each degree of temperature difference between the hot gases and the boiler plate on the one side and the boiler plate and the water in the boiler on the other. These surface conductances are combined with the conductances of the various solid layers so that an " overall surface conductance " or coefficient can be arrived at to define the the practical character of the plate in respect of heat transmission.

An allied conception is the " fouling factor " or " dirt factor " which is

equivalent to a decrease in the " film coefficient " or surface conductance to correspond to the increased resistance to heat flow. The use of this factor in computing heating surface has proved extremely valuable in practical work.

It is inevitable that the conservation of fuel should be bound up with the measurement of heat quantities. Accordingly in order to take the study of the subject of heat transmission beyond the elementary descriptive stage so far attempted, recourse to the methods of computation are necessary. The subject may be complicated and mathematical, but such a treatment is necessary, since no alternative is available.

CONDUCTION

APPLICATION OF THE LAW OF CONDUCTION TO TRANSMISSION OF HEAT THROUGH WALLS

The law of conduction, which has been described in words earlier in this chapter, can be expressed for the simplest case of heat flowing through a parallel plate (Fig. 35) at right angles to the sides of the plate, as follows :—

$$q \text{ (in heat units per hour) varies as } \frac{(t_1 - t_2)A}{L}$$

$$\text{or } q = \frac{k\,(t_1 - t_2)A}{L} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

where, k, the constant of proportionality, varies for different materials and is known as the thermal conductivity of the material.

For the conduction of heat per unit area of cross section

$$\frac{q}{A} = H = \frac{k\,(t_1 - t_2)}{L} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

Using British units (foot-pound-hour) throughout

q = total heat transmitted in B.Th.U. per hour.
H = heat transmitted in B.Th.U. per square foot per hour.
L = thickness of wall in feet.
t_1, t_2 = temperature of hotter and cooler sides of wall in °F.
k = thermal conductivity in B.Th.U. per square foot per hour per °F. per foot thickness (written for the sake of brevity, B.Th.U./sq.ft./hr./°F./ft.).
A = area of surface (square feet) normal to the direction of flow.

Often it is more convenient to express the thickness of the wall, L, in inches and to take k as B.Th.U./sq.ft./hr./°F./in. The formula $H = k(t_1 - t_2)/L$ is still applicable when both of these modifications are applied simultaneously.

THERMAL CONDUCTIVITIES

The nature of the variation of thermal conductivity of a number of common substances entering into industrial heat problems is shown in Table 36. Both the values for k just mentioned are given in Table 36.

The difference between metals and non-metals will be evident from Table 36. The thermal conductivity will also be seen to vary with the temperature and may increase or decrease with rise of temperature according to the material. Most metals conduct heat less well with rise of temperature, whereas the conductivity of most refractory materials increases with increasing temperature.

When calculating the rate of heat transmission through a substance as in Fig. 35, the thermal conductivity to be used is the average over the temperature range t_1–t_2. The value of the coefficient for aluminium at 400° C. is 1,240 B.Th.U./sq.ft./hr./°F./in., at 600° C. it is 1,050, and over the range 400–600° C. the average may be taken as 1,145.

TABLE 36

THERMAL CONDUCTIVITY OF SOME COMMON MATERIALS

Material	Temp. ° C.	Thermal conductivity *	
		B.Th.U./sq.ft./hr./° F./in.	B.Th.U./sq.ft./hr./° F./ft.
Aluminium	0	1,400	117
	200	1,490	124
	400	1,730	144
	500	1,860	155
Copper	100–200	2,916	243
	100–370	2,700	225
	100–541	2,620	218
	100–837	2,500	208
Wrought iron	18	418	35
	100	416	35
Cast iron	100	314	26
Steel	100	311	26
Nickel	100	400	33
	300	366	31
	600	256	21
	800	197	17
	1,200	168	14
Monel metal	100	203	17
	400	244	20
Brass	100	711	59
	400	810	68
Firebrick	300	7·3	0·61
	700	8·1	0·68
	1,100	8·4	0·70
Magnesite brick	300	60	5
	700	37·5	3·1
	1,100	27·5	2·3
Insulating bricks	0–200	0·73	0·06
	0–400	0·81	0·07
	0–600	0·90	0·08
	0–800	1·02	0·09
Magnesia pipe covering	50	0·38	0·03
	100	0·41	0·03
	200	0·46	0·04
	400	0·64	0·05

* The dimensions are strictly $\frac{\text{B.Th.U.}}{\text{ft. hr. ° F.}}$, but are stated in the form given for clarity.

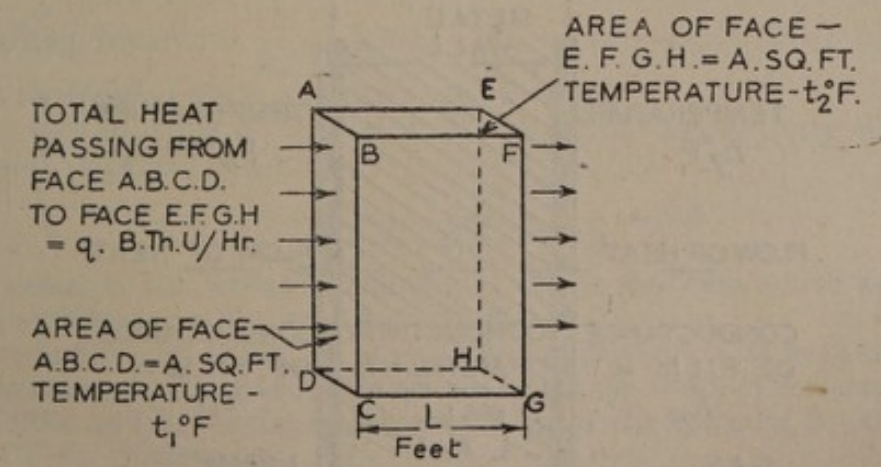

FIG. 35. Heat flow by conduction in a rectangular slab.

Examples. It is required to determine the rate of heat transmission through a refractory wall, 6 inches thick, when the flue temperature is 1,100° C. (2,010° F.) and the wall has been heated externally to 600° C. (1,130° F.). The mean value of the conductivity of the firebrick over the temperature range 600°–1,100° C. is known to be 8·2 B.Th.U./sq.ft./hr./°F./in.

$$H = \frac{8{\cdot}2 \times (2{,}010 - 1{,}130)}{6} = 1{,}200 \text{ B.Th.U./sq.ft./hr.}$$

Again, to illustrate the effect of differences in conductivity, suppose that in a furnace of the muffle type, the inner wall is to be maintained at 400° C. (752° F.), and it is required to calculate what must be the temperature of the flue if the muffle wall is (*a*) of cast iron 1 inch thick ; and (*b*) of firebrick 4½ inches thick, when the rate of heat transmission H = 700 B.Th.U./sq.ft./hr.

Here the amount of heat that is to be passed through the wall is the same in both instances, but the thermal conductivity of the materials is very different and is taken as 300 for C.I. and 8 for firebrick.

$$\text{For C.I. } 700 = \frac{300\ (t_1 - 752)}{1} \text{ whence, } t_1 = 754{\cdot}3° \text{ F. } (401{\cdot}3° \text{ C.})$$

$$\text{For firebrick } 700 = \frac{8(t_1 - 752)}{4\frac{1}{2}} \text{ whence } t_1 = 1{,}146° \text{ F. } (619° \text{ C.})$$

TRANSMITTANCE OR OVERALL HEAT TRANSFER COEFFICIENT

The calculation of the quantity of heat passing through a wall or a boiler tube lined on one side with soot and some oxide of iron or scale, and on the water side with " boiler scale " requires a knowledge of the surface temperatures of the wall. In practice these temperatures are often not known, but the temperatures of the media on each side of the wall are known. A simple example is a steam pipe. The temperature of the steam inside the pipe, and that of the air surrounding the pipe are known, but not the temperature of the pipe and the insulation. As already explained, to overcome this apparent difficulty it is convenient to use an overall coefficient of heat transfer, obtained in the following manner.

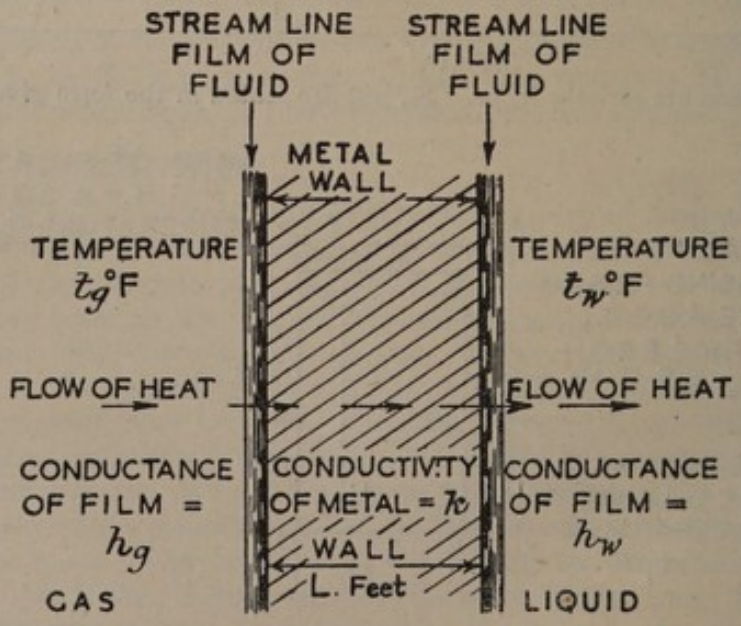

FIG. 36. Heat transmission through a plane wall separating two fluids.

The heat transmission through a simple plane wall separating two fluids g and w is (see Fig. 36) :—

$$q = \frac{A(t_g - t_w)}{\frac{1}{h_g} + \frac{L}{k} + \frac{1}{h_w}} \text{ B.Th.U. per hour} \quad . \quad . \quad . \quad (3)$$

where A = area of heating surface, square feet.
t_g = temperature of hot medium, °F. (the hot gas in a boiler flue)
t_w = ,, ,, cold ,, (the water in the boiler)
h_g = surface conductance of hot medium—B.Th.U./sq.ft./hr./°F./ft.
h_w = ,, ,, ,, cold ,, B.Th.U./sq.ft./hr./°F./ft.
L = thickness of the wall, feet.
k = thermal conductivity of the wall in B.Th.U./sq.ft./hr./°F./ft.

Defining the transmittance or overall heat transfer coefficient, h, as follows :—

$$q = Ah\,(t_g - t_w) \text{ B.Th.U.per hour} \quad . \quad . \quad . \quad . \quad . \quad (4)$$

and dividing equation (4) into equation (3) and transposing, it is found that

$$h = \frac{1}{\frac{1}{h_g} + \frac{L}{k} + \frac{1}{h_w}} \text{ B.Th.U./sq.ft./hr./°F} \quad . \quad . \quad . \quad . \quad (5)$$

THE RESISTANCE CONCEPT

The practical significance of these two equations is of supreme importance in heat transmission and is analogous to that of the application of Ohm's law in the transmission of electrical power. In comparison with Ohm's law, which states that the current = voltage/resistance, the temperature potential across the wall $(t_g - t_w)$ is equivalent to the electrical potential or voltage and the expression

$$\left(\frac{1}{h_g} + \frac{L}{k} + \frac{1}{h_w}\right)$$

is analogous to the resistance, the total resistance being the sum of the partial resistances. The flow of heat is analogous to the flow of current. Just as the reciprocal of the resistance in an electric wire measures the " conductance " of the wire for electricity and *vice versa*, so the reciprocal of the thermal conductance (e.g. $1/k_w$) measures the resistance to the flow of heat.

It is an important fact that if the partial resistances differ to any great degree in magnitude only the largest resistance will have a significant effect on heat transmission. Thus, in a boiler tube, taking a case quoted by Schack, the surface conductance on the water side, h_w, is of the magnitude of 1,000, and the corresponding resistance $\frac{1}{h_w} = 0{\cdot}001$. The surface conductance on the flue side, h_g is, however, only 6, and resistance = 0·167.

Putting into equation (5) the values of 27·5 for the conductivity of mild steel and a metal thickness of $\frac{3}{4}$ inch,

$$h = \frac{1}{0{\cdot}167 + 0{\cdot}0023 + 0{\cdot}001} = 5{\cdot}9$$

virtually equal to the surface conductance on the flue side, which accordingly governs the measure of the heat transmission.

In a condenser surface, on the other hand, conductances on both sides of the heat exchanging surface have high values, h_w on the water side being approximately 1,000, and h_g on the steam side being of the order of 2,000 B.Th.U./sq.ft./hr./°F./ft. thickness.

The conductivity of the metal wall can here be shown to be the governing

factor, and reduction of the wall thickness results in an appreciable improvement of heat transmission. It may further be shown * that, although the thermal conductivity of brass is more than twice that of steel, the transmittance across a 0·2 inch brass wall is only 19 per cent. greater than that across a steel wall of the same thickness. In general the largest partial resistance should first be considered in any attempt to improve heat transmission.

CONDUCTIVITY FORMULA FOR COMPOSITE WALLS

The standard conductivity formula requires modification when the wall is made of more than one kind of material. If, for example, there are three layers of materials in the wall through each of which the heat flows in turn, the thicknesses respectively being L_1, L_2 and L_3, and the conductivities k_1, k_2 and k_3 (Fig. 37) the formula becomes

$$H = \frac{t_1 - t_2}{\frac{L_1}{k_1} + \frac{L_2}{k_2} + \frac{L_3}{k_3}} \quad \text{B.Th.U./sq.ft./hr.} \quad . \quad . \quad . \quad (6)$$

Thus, if a wall consists of 9 inches of firebrick ($k_1 = 8{\cdot}1$) backed by 3 inches of

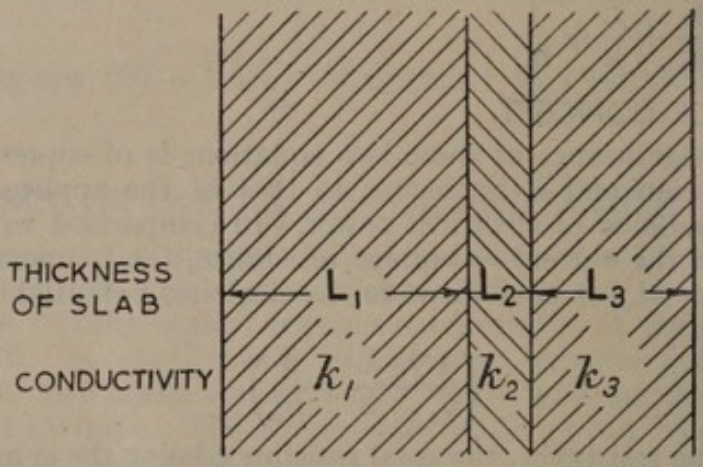

FIG. 37. Conduction through a composite wall.

insulating brick ($k_2 = 0{\cdot}8$) the inside temperature being 2,000° F., and the outer surface 300° F., the heat that will pass through is given by :—

$$H = \frac{2{,}000 - 300}{\frac{9}{8{\cdot}1} + \frac{3}{0{\cdot}8}} = 350 \text{ B.Th.U./sq.ft./hr.}$$

A further example applies to the water side of the boiler on which a layer of scale may have formed. Scale has a thermal conductivity of 12–18 B.Th.U./sq.ft./hr./°F./in. thickness., and is thus similar to a firebrick in this respect. Some scales have a much lower conductivity than this. If it is accepted that the safe maximum continuous temperature for boiler tubes is 900° F., a calculation of the thickness L enables an estimate to be made of the thickness of scale that might permit the plates to be dangerously overheated :—

In modern boilers heat input rates may be 70,000–100,000 B.Th.U./sq. ft./hr. If the boiler is at 100 lb. pressure (338° F.) the standard conductivity formula gives

$$70{,}000 = \frac{18(900 - 338)}{L}$$

whence $L = 1/7$ inch.

* " Industrial Heat Transfer," A. Schack. Trans. by H. Goldschmidt and E. P. Partridge (John Wiley, 1933).

Thus a scale only one-seventh of an inch thickness may set up a dangerous condition in a boiler. The direct effect of scale on thermal efficiency is probably of the order of 1½ to 6 per cent. loss, according to how much of the total boiler heating surface is coated with scale.

CONDUCTION THROUGH PIPE WALLS

Although the formulæ for conductivity through a flat surface just given are approximately correct for a curved surface of comparatively large diameter, provided the thickness is small compared with the diameter, they are incorrect when the diameter is small. For calculations of the flow of heat through pipe coverings other formulæ must be used, based on the same general principles.

If the diameter of a pipe is d_1 inches and the diameter of the outer surface of pipe covering is d_2 inches it can be shown that the heat flow is given by :—

$$q = \frac{2\pi k}{2{\cdot}3 \log \frac{d_2}{d_1}} \cdot (t_1 - t_2) \text{ B.Th.U./hr./ft. run of pipe} \quad . \quad . \quad . \quad . \quad . \quad . \quad (7)$$

where t_1 and t_2 are the temperatures at diameters d_1 and d_2 respectively.

This is perhaps the most useful form for practical purposes, but if the loss of heat per unit surface area of the pipe is required, it is given by :—

$$H = \frac{k}{2{\cdot}3r \log \frac{d_2}{d_1}} \cdot (t_1 - t_2) \text{ B.Th.U./sq.ft./hr.} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (8)$$

where r is the distance from the pipe axis ; when $r = \frac{1}{2}d_1$, this gives the heat loss per unit area of the metal pipe, and when $r = \frac{1}{2}d_2$, the basis is on the area of the external covering.

Thus if a 6-inch steam pipe is covered by 2½ inches of magnesia, experiment shows that the surface temperature of the magnesia in reasonably still air is 105° F. when conveying steam at 500° F. The thermal conductivity of the magnesia covering may be taken as 0·42 over this temperature range. The heat lost per foot run is then calculated :—

$$q = \frac{2\pi \times 0{\cdot}42\ (500 - 105)}{2{\cdot}3 \log\ (5\frac{1}{2}/3)} = 1{,}725 \text{ B.Th.U. per hour.}$$

The heat lost per square foot of surface area of the metal pipe is given by :—

$$H = \frac{0{\cdot}42\ (500 - 105)}{0{\cdot}25 \times 2{\cdot}3 \log\ (5\frac{1}{2}/3)} = 1{,}096 \text{ B.Th.U. per hour.}$$

For the more complete solution of such problems, taking into account the surface conductance inside and outside the pipe, the heat transmission per foot run of composite pipe wall is given by :—

$$q = \frac{\pi(t_s - t_a)}{\frac{1}{d_1 h_s} + \frac{1{\cdot}151}{k_1} \log \frac{d_2}{d_1} + \frac{1{\cdot}151}{k_2} \log \frac{d_3}{d_2} + \frac{1}{d_3 h_a}} \text{ B.Th.U./hr./ft. run} \quad . \quad . \quad . \quad (9)$$

where t_s, t_a are the temperatures of the steam and the surrounding atmosphere.
d_1 and d_2 = internal and external diameters of the metal pipe, feet.
d_2 and d_3 = ,, ,, ,, ,, ,, lagging, feet.
k_1 and k_2 = thermal conductivities of the metal of the pipe and the lagging material.
h_s and h_a = surface conductance of the internal pipe surface and external surface of the lagging.

The external conductance h_a, is in this expression made up of the convection and radiation coefficients, h_c and h_r explained later.

CONVECTION

Convection is dependent upon the laws of fluid flow (Chapter IX). Osborne Reynolds showed the difference between streamline or laminar flow and turbulent flow by introducing a fine stream of a red liquid into water flowing through a glass tube. Fluid friction increased instantaneously with the transition from streamline to turbulent motion, and the law governing the pressure drop through the tube changed. With streamline flow it was directly proportional to the quantity of fluid flowing through the tube, with turbulent flow to the square of this quantity.

In natural convection the motion of the fluid is caused by gravity due to differences in density between hotter and cooler parts ; in forced convection the fluid motion is caused by external forces, such as a chimney draught or a fan. In industrial work, the motion induced by a chimney is sometimes regarded as a form of natural convection, the term forced convection being reserved for motion set up by fans or other external means.

A hot surface, e.g. a steam pipe or a heated tank, over which the air flows freely, is losing heat to the air by natural convection. A nest of tubes in a condenser over which gas is blown by a blower or fan is absorbing heat from the gases by forced convection.

Progress in the study of convection in its bearing on heat transfer has been made by the application to the problem of dimensional analysis and the theory of similarity, a means whereby data obtained from one set of observations can be applied to another provided they are dimensionally compatible and bear geometrical similarity to one another. Thus results obtained from models can be applied to industrial plant. Further the method has been the instrument of correlating and establishing the validity of formulæ determined on widely differing types of apparatus, as well as the means of providing a technique for solving individual types of problems. For further information reference should be made to the modern standard works.

BASIS OF STUDY OF CONVECTION—FLOW IN PIPES

Reynolds showed in his studies of the motion of water * that certain relationships existed in regard to the velocity (v), the depth of the fluid stream (D), the density of the fluid (ρ) and the viscosity (η), which could be grouped together to give a criterion known as Reynolds's Number (Re), a fuller account of which will be found in Chapter IX. It is sufficient to state here that

$$\mathrm{Re} = \frac{\mathrm{vD}\rho}{\eta}$$

where v and ρ = velocity and density at the conditions of temperature and pressure ruling, and the viscosity, η, is defined in various ways using either unit force or unit mass. The value of the criterion Re is the same whatever consistent system of units be used. The change from streamline to turbulent flow is associated with a value of Re = about 2,300. It has been further shown that the characteristics of flow are associated with the heat transference from moving fluids, and that resistance to fluid flow and heat transfer are closely related (see Chapter IX).

The velocity distribution in two streams of fluids flowing in pipes will be similar if the Reynolds numbers are the same for both. The temperature distribution in the streams will be the same if the expression vD/a, where a is the thermal diffusivity, is the same for both ; further, for two fluid streams to have similar temperatures, the expression hD/k must be the same for both where the diffusivity = $\mathrm{k}/\mathrm{c}\rho$

* Ref. "On the Motion of Water." Trans. Roy. Soc., London, **174**, 935 (1885).

k = thermal conductivity
c = sp. ht. per unit weight
ρ = density
h = surface conductance
D = width of stream

Study of these groups has shown that for specific applications they are related by some simple value. Thus Nusselt developed the relationship shown by the equation

$$\frac{hD}{k} = f\left(\frac{VD}{a}\right) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (10)$$

the dimensionless group $\frac{hD}{k}$ being known as the Nusselt number Nu, and he found for heat transfer in a brass tube of inside diameter 0·87 inch that the surface conductance, h, increased with the 0·786 power of the velocity, V. That is to say

$$h = C \,.\, V^{0.786} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (11)$$

where C represents all the variables unchanged in the experiments.

It can be shown by the development of the above equations that the surface conductance is derivable from

$$h = \frac{k}{D} C' \,.\, \left(\frac{VD}{a}\right)^{0.786} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (12)$$

in which C′ is a general constant determined by the specific conditions. Thus for forced turbulent flow of gas in pipes

$$h = 15{\cdot}9 \frac{k_{wall}}{D^{0.214}} \left(\frac{V.C_p}{k}\right)^{0.786} \text{ B.Th.U./sq.ft./hr./°F.} \quad . \quad . \quad . \quad . \quad . \quad . \quad (13)$$

Certain transformations have been made and the values are

k_{wall} = thermal conductivity of the gas at the temperature of the pipe wall, B.Th.U./sq.ft./hr./°F./ft.
k = thermal conductivity of the gas at its mean temperature, B.Th.U./sq.ft./hr./°F./ft.
C_p = specific heat of the gas under the conditions of flow, B.Th.U./cu.ft./°F.
V = velocity of the gas flow, feet per second.
D = internal diameter of the pipe, feet.

It is also possible to derive expressions for the calculation of h by the use of the Reynolds' criterion, since the value of this is constant for similar streams. Generally the subject is complex and beyond the present aim to provide a basis for study and indicate the sources of the most essential formulæ. Simplifying the expression by applying the normal values of the constant for chimney gases and air the surface conductance by convection for turbulent flow in long smooth pipes at any pressure is given by

$$h = \frac{0{\cdot}32\, v_o^{0.8}}{\sqrt[4]{D}} \text{ B.Th.U./sq.ft./hr./°F.} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (14)$$

where v_o = fluid velocity in feet per second calculated to S.T.P.
D = diameter of the duct, feet.

For ducts which are not circular or square, the equivalent diameter is given by

(area of cross-section × 4)/perimeter.

For further reference to the literature of a wide field of experimental investigation in this class of problem which has been the basis of the useful empirical formulæ given in the remainder of this discussion, the reader should consult

more specialised works on the subject, e.g. McAdams' " Heat Transmission " (2nd Ed., 1942), Fishenden and Saunders' " The Calculation of Heat Transmission " and Schack's " Industrial Heat Transfer " (trans. by Goldschmidt and Partridge).

NATURAL CONVECTION

The transfer of heat by natural convection is governed by the area of the surface, the shape and position of the surface and the temperature difference between the surface and the gas. A useful empirical expression is,

$$H = C\,(t_1 - t_2)^{1.25}\ \text{B.Th.U./sq.ft./hr.} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (15)$$

where t_1 is the temperature of the surface, t_2 of the gas or air, and C is a constant depending on the shape and position of the surface.

The value of C as determined by experiment is as follows :—

C = 0·39 for a plane horizontal surface facing up and hotter than the gas ; or facing down and colder than the gas.

C = 0·3, for plane vertical surfaces and for large bodies of irregular shape but without re-entrant angles.

C = 0·2 for horizontal surfaces facing downwards and hotter than the gas.

For large horizontal cylinders (and this applies to pipes) over 6 inches diameter C = 0·35. As the cylinder diameter decreases (e.g. with small pipes) the value of C increases rapidly, so that for a wire of 0·01 inch diameter C = 3·5.

Some idea of the magnitude of natural convection effects can be seen from Fig. 38.

As a simple example of the use of convection formulæ, consider a bare hot water tank nearly filled with hot water at 180° F. placed in a workshop with air

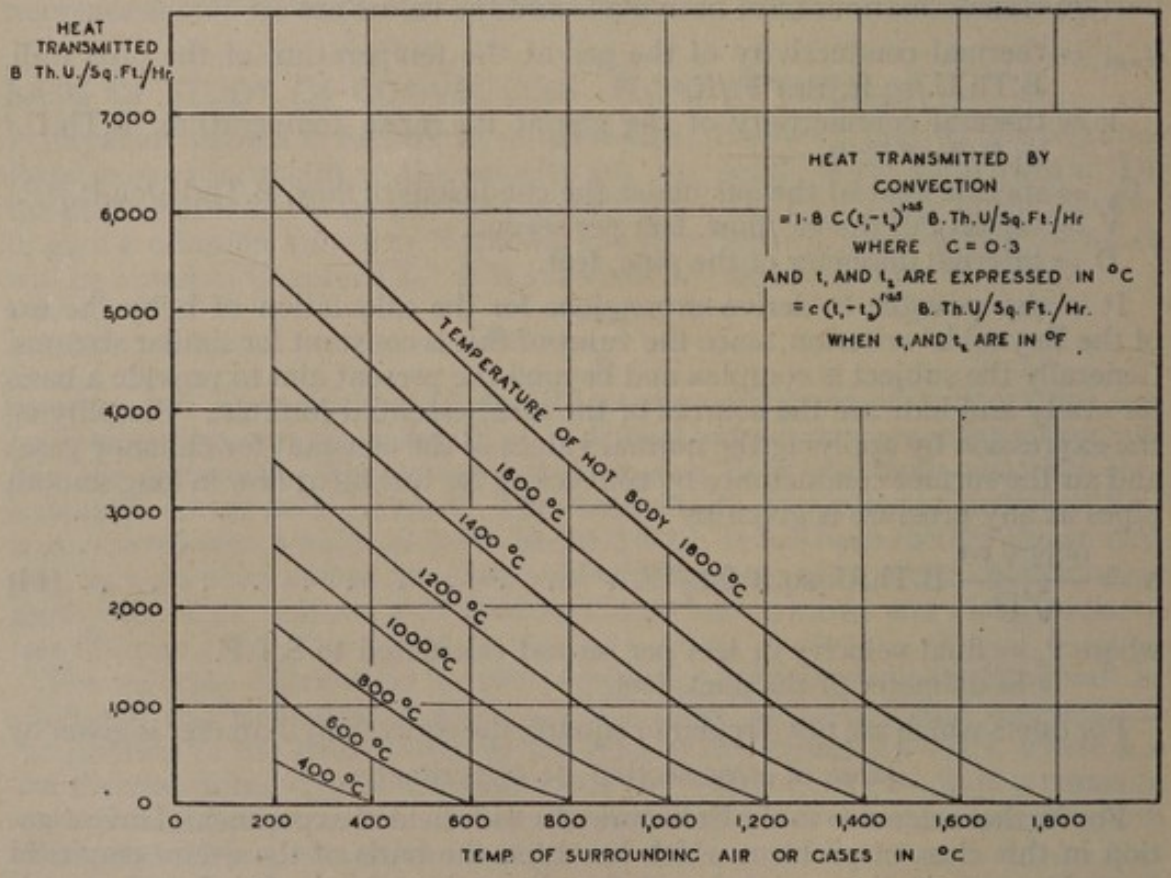

FIG. 38. Heat transfer by natural convection in still air

temperature of 70° F. The tank is supported on four columns so that its base is exposed. It measures 20 feet × 14 feet wide, and is 10 feet high. It is connected to an apparatus where the water is used by a pipe 8 inches diameter and 15 feet long. It is required to calculate the loss of heat per hour by convection.

	sq. ft.
Area of connecting pipe	31·4
Area of top of the tank	280
Area of bottom of the tank	280
Area of sides of the tank	680

Heat loss from connecting pipe $= 0{\cdot}35 \times 31{\cdot}4 \times (180 - 70)^{1{\cdot}25}$
$= 3{,}900$ B.Th.U. per hour.

Heat loss from top of tank $= 0{\cdot}39 \times 280 \times (180 - 70)^{1{\cdot}25}$
$= 40{,}000$ B.Th.U. per hour.

Heat loss from bottom of tank $= 0{\cdot}2 \times 280 \times (180 - 70)^{1{\cdot}25}$
$= 20{,}000$ B.Th.U. per hour.

Heat loss from sides of tank $= 0{\cdot}3 \times 680 \times (180 - 70)^{1{\cdot}25}$
$= 72{,}600$ B.Th.U. per hour.

$H = 3{,}900 + 40{,}000 + 20{,}000 + 72{,}600 = 136{,}500$ B.Th.U. per hour.

In addition to this there is loss by radiation which will be dealt with in a later section. Since the tank is unlagged it has been assumed in the foregoing example that the surfaces of the tank are at the same temperature as that of the water. The loss of heat can be reduced if the value of the temperature drop (180 — 70)° F. can be decreased. If the tank and pipe are lagged, the surface temperature should be reducible at least to 90° F. The loss of heat by convection will then be reduced to 16,200 B.Th.U. per hour—a striking commentary on the value of adequate insulation.

FORCED CONVECTION

In describing earlier the basis of the study of convection some indication of

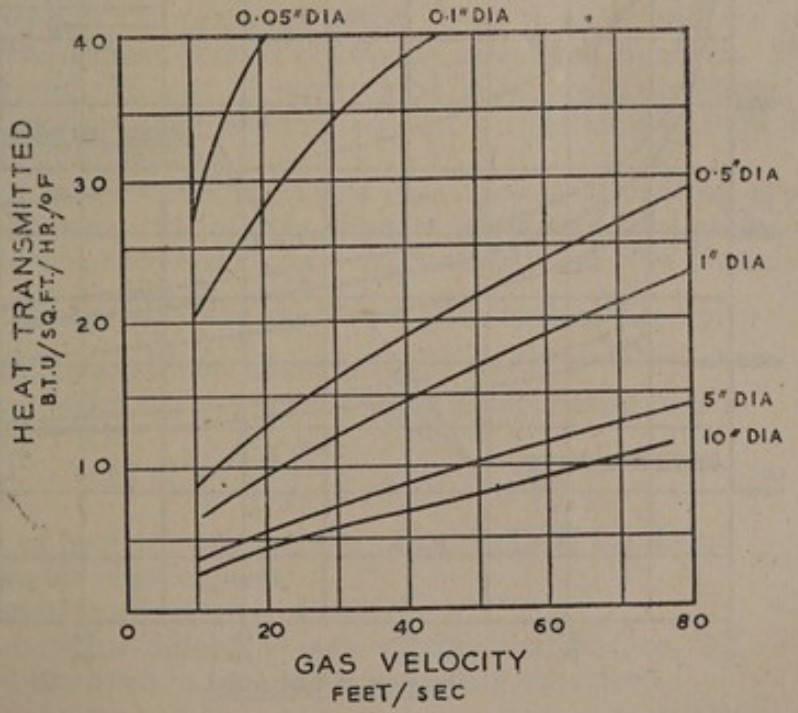

FIG. 39. Heat lost from cylinders by forced convection to air flowing at right angles to the cylinder.

(Based on "The Calculation of Heat Transmission," by M. Fishenden and O. A. Saunders, H.M. Stationery Office.)

the most useful formula has already been given, from which it will be noted that the rate of heat transfer depends directly upon the difference of temperature between the hot and cold elements of the heat exchanging surface, and not upon the $\frac{5}{4}$ power of the temperature difference, i.e.

$$H = h\,(t_1 - t_2) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (16)$$

where h is the "surface conductance," and is analogous to k, the thermal conductivity in the simplest case of conduction through a solid slab. The parallel is, however, far from being exact, for unlike k, which is a characteristic of the material, the value of h depends upon the linear velocity of the gas stream, the size of the body, and the physical properties of the gas.

The above formula is applicable to a wide range of conditions, the value of the surface conductance naturally varying with circumstances.

The following statement defines the values in certain specific applications.

Single tube in a transverse stream of air.

Fig. 39 shows the values found for h for tubes of various sizes. If the turbulence of the gas stream is artificially increased these values may be increased by up to 50 per cent. Increased turbulence means increased resistance, and therefore increased draught is needed, so that this method of increasing the heat flow must be used with caution. A Galloway tube is an example of this arrangement.

Single tube in longitudinal flow of air.

The mean value of h for any given conditions is about half that for the transverse stream, but can be doubled by creating artificial turbulence.

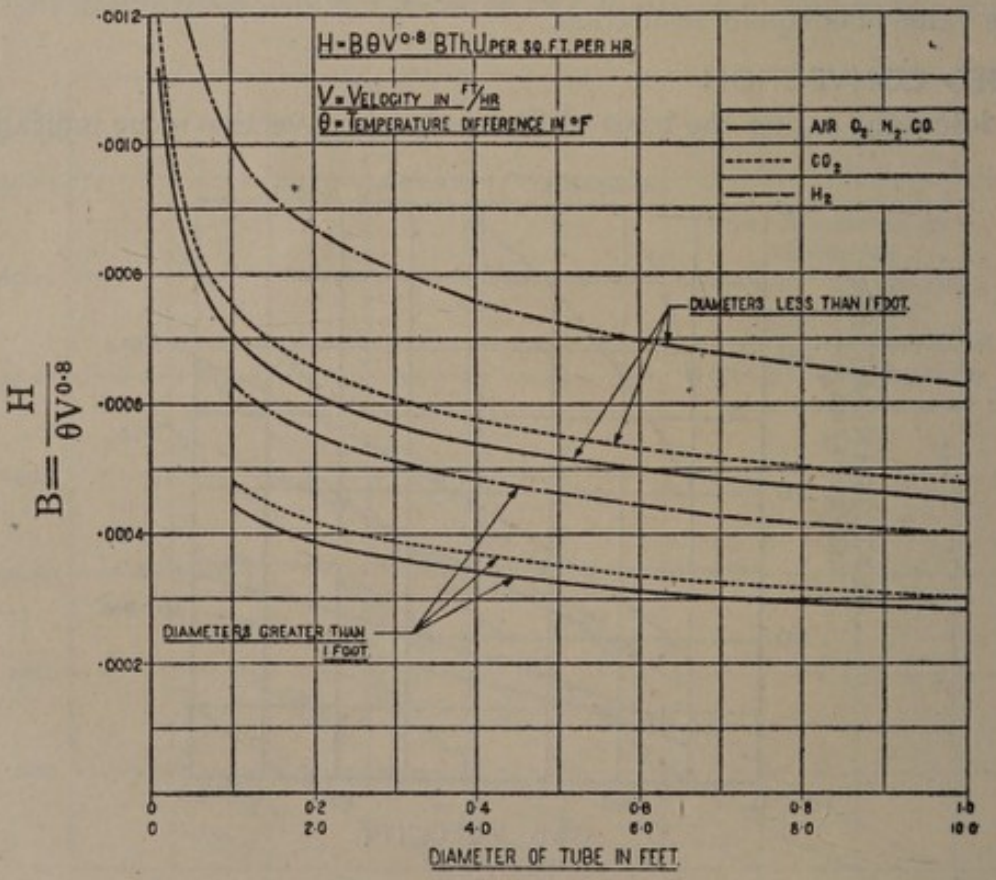

FIG. 40. Coefficients of forced convection for air or other gases flowing through tubes of varying diameter.

(From "The Calculation of Heat Transmission," by M. Fishenden and O. A. Saunders, H.M. Stationery Office.)

A bundle of tubes in a transverse stream of air.

This construction is that of an economiser or condenser. A staggered arrangement of the tubes through creating greater resistance is better than a straight-through flow.

The use of fins.

In air heaters, and other plant, the heat transfer by forced convection is increased by adding fins or other projections to increase the effective surface area. The heat transfer to fins is actually less than the heat transfer to the bare tube, but in view of the increased exposed surface, the overall heat transfer referred to the bare tube surface is increased. There is a limit to the useful height of the fins and their spacing. The larger and more numerous the fins the greater will be the heat transfer for a given base area ; but increase in height causes a reduction in the mean temperature of the exposed surface and if the fins are too close the flow of air between them is impeded. The fins must be of sufficient thickness to conduct the heat to or from the solid plate without too great a temperature gradient.

Flow through a single tube

The circular flues of a shell boiler, a main carrying hot producer gas, some forms of air heater, or any other arrangement in which the gas being heated flows through a tube, are examples of this class of problem. Here it is found that

$$H = B\, v^{0.8} (t_1 - t_2) \text{ B.Th.U. per square foot per hour} \quad . \quad . \quad (17)$$

v is the velocity of the gases or liquids passing through the tube in feet per hour. The value of B is given in Fig. 40. Another treatment of this arrangement on more general lines is that already described, equation (14).

HEAT TRANSFER AND PRESSURE DROP IN FLOW ACROSS BANKS OF TUBES

In discussing the contribution made by Osborne Reynolds to our knowledge of fluid flow, the importance of the resistance to flow introduced by fluid friction was indicated. In practical design, heat transfer and the resistance to flow through the appliance must always be considered together. Since the cost of equipment and the cost of power tend in opposite directions, there is an economic balance for some optimum value of resistance to fluid flow and the area of the heating surface.

In banks of tubes, occurring in so many forms of heating appliances and exchangers, the mean rate of heat transfer H per unit tube surface is given in all cases of forced convection by a relation between the Nusselt number (a dimensionless group, $\frac{Hd}{k\theta}$) and the Reynolds' number, Re, which depends upon the geometrical arrangement of the tubes.

The relation, which fits the most reliable experimental data is

$$\frac{Hd}{k\theta} = 0{\cdot}29\, C_H \left(\frac{vD\rho}{\eta}\right)^{0.61} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (18)$$

H = rate of heat transfer per unit area, B.Th.U./sq.ft./hr.
d = diameter of pipes, feet.
k = thermal conductivity of fluid.
θ = temperature difference between fluid and surface, °F.
$vD\rho/\eta$ = Reynolds' number as previously described.

and the constant C_H depends upon the tube arrangement. The velocity v is measured at the narrowest restriction. This may occur between adjacent tubes in line, or between two diagonally opposed tubes.

Frictional resistance to flow is given by the well-known D'Arcy formula (also known as the Fanning equation), which is discussed in detail in Chapter IX. According to this expression, if

h = pressure drop in inches w.g.
f = coefficient of friction.
g = gravitational constant
D = pipe diameter (feet)
ρ = gas density in lb. per cubic foot.
v = velocity in feet per second.

$$h = \frac{2f\rho v^2}{gD} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (19)$$

For flow through tubes this can be resolved into

$$h = 10^{-3}\, C_F n \rho v^2$$

where C_F is a coefficient of discharge depending on the tube arrangement and n is the number of rows of tubes.

Values of the constants C_H and C_F for various arrangements of tube banks, both in line and staggered are to be found in a recent paper by Lander.* The magnitude of the heat transfer is proportional to the corresponding value of the coefficient C_H in both in-line and staggered arrangements. For "in-line" arrangements with values of the Reynolds number above 20,000, C_H becomes unity for all kinds of spacings, from which it follows that the heat transfer is determined by the velocity. For staggered tubes it drops slightly. The mean value of the heat transfer at wide spacings in line is some 38 per cent. below that for a single tube, due to the shielding effect of one tube on another. For staggered formations the pressure drop appears to be quite independent of the longitudinal space, but rises as the lateral spacing is decreased.

In general, a heat exchanger becomes more economical as the pipe diameter is decreased. The installation cost decreases approximately with the fourth root of the pipe diameter, but practical conditions limit the reduction of cross section. Where heating surface is relatively expensive it will not be economical to use stack draught, since the velocity would probably be too far from the optimum value.

HEAT TRANSFER TO SINGLE PIPES

The best simple equation for highly turbulent flow of air against pipes is given by*:—

$$h = 0{\cdot}76\, \frac{v_o^{0{\cdot}56}}{D^{0{\cdot}44}} \text{ B.Th.U./sq.ft./hr./°F.} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (21)$$

v_o is the velocity in feet per second at N.T.P.; D is the pipe diameter in feet.

For undisturbed flow the constant becomes 0·70.

PIPE BUNDLES

A practical equation for conductances by convection from air to rows of evenly arranged or staggered pipe bundles has the following general form*:—

$$h = \frac{C \,.\, v_o^{\,x}}{D^y} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (22)$$

v_o is measured at N.T.P. as in (21).

The values for the various groupings are given in Table 37. They are valid for air at any temperature and pressure, and are approximately valid for flue gases.

* "A Review of Recent Progress in Heat Transfer," C. H. Lander, Journal, Institution of Mechanical Engineers, April 24th, 1943, pp. 95–98.

TABLE 37

Arrangement of pipe bundles		Even rows	Staggered rows
Values of indices	x	0·654	0·69
	y	0·346	0·31
No. of rows		C	C
2		0·615	0·70
3		0·635	0·79
4		0·65	0·86
5		0·66	0·91

HEAT TRANSFER FROM FLUIDS TO WALLS

In forced convection in a plane wall, as distinct from natural convection, the heat transfer has been found experimentally to depend not only upon whether the plate or wall is vertical or horizontal, but also on the character, particularly the roughness, of its surface. Data based mainly on experiments by Jürges is as follows (Table 38).

TABLE 38

FORCED CONVECTION IN PLANE WALLS

(Values of h in B.Th.U./sq.ft./hr./°F.)

For velocities smaller than 16·5 ft./sec.	For velocities larger than 16·5 ft./sec.
$h_{cs} = 0{\cdot}98 + 0{\cdot}20\ v$, for a smooth surface	$h_{cs} = 0{\cdot}50\ v^{0{\cdot}78}$
$h_{cr} = 1{\cdot}02 + 0{\cdot}20\ v$, for a rolled surface	$h_{cr} = 0{\cdot}50\ v^{0{\cdot}78}$
$h_{cr} = 1{\cdot}09 + 0{\cdot}22\ v$, for a rough surface	$h_{cr} = 0{\cdot}52\ v^{0{\cdot}78}$

The air velocity v is here referred to a basis of 70° F. If the temperature varies much from 70° F. the correction should be applied to bring the corresponding value of v to 70° F. Thus if the temperature is 200° F., and the actual velocity 20 feet per second, the corrected value of v becomes

$$v = 20 \times \frac{460 + 70}{460 + 200} = 16{\cdot}05 \text{ feet per second.}$$

For surfaces greater than $1\frac{1}{2}$ square feet h becomes decreased by as much as 40 per cent. for surfaces several feet in length.

HEAT TRANSFER FROM SUPERHEATED STEAM WITHIN PIPES

Whilst the fundamental investigations of heat transmission by convection have produced equations for practical use which are applicable to all fluids, condensing and boiling liquids introduce special conditions. It is doubted by some authorities if the equations for superheated steam are sufficiently satisfactory, and further investigation is probably needed. Schack gives the best available general equation for superheated steam in pipes, as follows :—

$$h = \left(0{\cdot}432 + 0{\cdot}151\ \frac{t}{1{,}000}\right) \frac{v_o^{0{\cdot}79}}{D^{0{\cdot}16}\ L^{0{\cdot}05}} \text{ B.Th.U./sq.ft./hr./°F.} \quad . \quad . \quad . \quad (23)$$

The inclusion of a term involving the length of the pipe is on account of decreased turbulence in the fluid flow with increase in pipe length, the value of the exponent 0·05 having been established by Nusselt.

CONVECTION IN LIQUIDS

It has been shown in reference to the work of Osborne Reynolds that the viscosity of a fluid is linked with the mechanics of its flow (see Chapter IX, Fig. 56). The layer of fluid in contact with the surface is virtually stationary ; the next layer flows over this stagnant layer with an imperceptible velocity, and so on, each succeeding layer moving with a higher speed. The greater the viscosity of the fluid the more is the resistance to motion.

The temperature drop through the film has been illustrated in Fig. 34. The significance of the velocity in influencing the thickness of the non-conducting stationary layer and so the surface conductance, and the reason for the use of the alternative term " film coefficient " is thus readily apparent. It is accordingly not surprising to find that for water flowing slowly in pipes the conductance has a remarkably low value. In practice the surface conductance from or to bodies in contact with water has values of the order of 40–200 according to the opportunity for the development of convection currents.

Fluid flow arising from the boiling of a liquid at a heating surface must be distinguished from free convection, since the ebullition introduces a measure of forced convection. The surface conductance for heat transmission to boiling water is of the order of 1,000–4,000 B.Th.U./sq.ft./hr./°F. according to the intensity of boiling and conditions of water circulation. The aggregation of steam bubbles over an intensely heated area resulting in overheating owing to the insulating effect of the layer of gas bubbles is a recognised occurrence in regions of poor circulation. The fact that the heat transfer is somewhat greater for vertical than for horizontal surfaces is thus also to be expected. It has also been established that the more viscous the liquid the lower is the surface conductance under conditions of ebullition.

The correlation of experimental results on the basis of similarity transfer by forced convection for liquids flowing in tubes in turbulent motion has been shown by Fishenden and Saunders (*loc. cit.*, p. 186) to be

$$\frac{Hd}{k\theta}\left(\frac{c\nu}{k}\right)^{0\cdot5} = 0\cdot03\left(\frac{vcd}{k}\right)^{0\cdot8} \quad . \quad . \quad . \quad . \quad . \quad . \quad (24)$$

where H = heat flow per unit area per unit time, B.Th.U./sq.ft./hr.
d = diameter, feet.
k = thermal conductivity, B.Th.U./sq.ft./hr./°F./ft.
θ = temperature difference, °F.
c = specific heat per unit volume at constant pressure, B.Th.U./cu. ft./°F.
ν = kinematic viscosity, sq.ft./sec. = η/ρ (see Chapter IX).
v = linear velocity, feet per second.

all values being in consistent units, either English or metric.

A simplified equation of the surface conductance of water with temperature t_w and with water flowing in pipes at an actual velocity of v feet per second, based on a density of 62·3 lb. per cubic foot is given by

$$H = 150\ (1 + 0\cdot011t_w)v^{0\cdot8}/d^{0\cdot2} \quad . \quad . \quad . \quad . \quad . \quad (25)$$

where d = internal pipe diameter in inches and t_w is in ° F.

Thus a flow of water of 9,000 lb. per hour passing through a pipe of 3 inches internal diameter will give a surface conductance at 200° F. of 328 B.Th.U./sq.ft./hr./°F. The drop of temperature through such a pipe freely exposed to the atmosphere is found under these conditions to be roughly of the order of 1° F. Further, this rate of loss of heat is just equal to the rate of loss of heat from a bare pipe by radiation and convection when there is an atmospheric temperature of 60° F. By calculating the drop of temperature in the water from a

knowledge of its rate of flow and heat capacity it can be shown that the water must traverse 30 feet of pipe to lose this 1° F. in temperature.

HEAT TRANSFER FROM CONDENSING VAPOURS

Heat transfer from saturated vapours differs fundamentally from heat transfer by convection from gases. As soon as a saturated vapour touches a surface which is colder than itself, condensation occurs, and there is given up to the cooling surface the latent heat of condensation of the liquid. At the same time a film of the condensed liquid is formed on the cooling surface and further heat transfer takes place through the film. Liquids are generally bad conductors and the thickness of the film of liquid must accordingly be of significance.

For this reason it is found that the character of the heat transfer differs in the horizontal and vertical positions and the conditions of drainage. Thus the height of a vertical condensing column, the viscosity of the liquid and the roughness of the surface, affect appreciably the resultant heat transfer.

The surface conductance of condensing saturated steam is given approximately for minimum values by the following :—

$$H_s = \frac{A + B\,(t_s + t_w)}{\sqrt[4]{L(t_s - t_w)}} \text{ B.Th.U./sq.ft./hr./°F.} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (26)$$

where the constants A and B have the values

	A	B
vertical pipe	1,590	4·07
horizontal pipe	1,225	3·15

t_s = steam temperature, °F.
t_w = wall temperature, °F.
L = length of vertical pipe and diameter of horizontal pipe.

Alternatively alignment charts by which problems falling into this class can be solved are given in " Heat Transmission " by W. H. McAdams (McGraw-Hill, 1942), pp. 268, 269.

As a rough guide only to the relative surface conductances or overall coefficients of heat transfer, Table 39 is submitted with the remark that calculations should be based on the pertinent equations available in the literature. The table of data merely serves to show for example that in comparison with the surface conductance of gases, the values for liquids are very much higher.

TABLE 39

Conditions	Overall coefficient of heat transfer, B.Th.U./sq. ft./hr./° F. temp. diff.
Steam to water being heated	80–140
Hot water to water being heated	70
Steam to boiling water	140
Steam to thick fruit product	32
Cooling water by cold water and brine	40–120
Hot oil to oil being heated	13–24
Hot oil to boiling water	30–40
Steam to water being heated to tubular heaters	100–160
Steam being condensed to water in tubular condenser jacket	140
Preheating sugar molasses by steam through copper coils	60–260
Enamelled apparatus with low-pressure steam condensing outside with water boiling inside	up to 200
Do., but with hot oil as the heating medium	may fall to 10 or 20

LIQUID CONVECTION CURRENTS—BOILER CIRCULATION

Since water is a very poor conductor of heat, the absorption of the heat transmitted from the fire to the water side of boiler heating surface must be effected by currents within the body of the liquid. In comparatively stagnant water, experiments tend to prove that the water film adjacent to the heated metal is heated by conduction, and subsequently breaks up into numerous small steam bubbles. These, unless removed by rising through the liquid or by the flow of water over the surface to which they are attached, form a heat-resisting blanket, and prevent the dispersal of heat by eddy currents throughout the main volume of water. This effect may be observed in boilers having water "legs" or regions of poor circulation, the aggregation of steam bubbles over an intensely heated area resulting in overheating and possibly deformation of the plate. Water circulation is therefore of vital importance for even moderate rates of heat transmission.

Unless otherwise constrained, steam bubbles tend to rise in a vertical direction, and since the greatest number is evolved in the region of the greatest heat transmission, the heating surface must be designed with free and adequate circulation on the water side. In an internally-fired boiler, for instance, the greater evaporation takes place over the furnaces. The water space above the furnaces, therefore, should be clear of obstructions, and when tube nests are provided the tubes must be as widely spaced as possible.

In practice it is usual to provide suitable "lanes" between tube nests, through which the ascending hot water and steam bubbles flow once circulation is established. An upward-moving current implies the existence of a downward flow to replace the water displaced, and similar provision must be made in this respect. In Lancashire boilers the colder water may flow down the space between the flues and shell, while in horizontal multitubular boilers a space of at least 3 inches must be left for this purpose between the outer tube peripheries and the boiler shell.

Neglect of adequate circulation lanes in shell boilers will result in erratic and violent ebullition, and when regions of the water space do not participate in the general circulation, heavy thermal stresses may be set up in the adjacent metal parts. This sometimes occurs in badly-designed boilers of the internally-fired type.

In water-tube boilers the necessity and conditions for good circulation on the water side are the same, but their achievement in practice is complicated by the high rates of heat transmission possible on this type of boiler. Natural circulation is a function of the ratio between the density of steam and that of water, or, in other words, the natural circulation of water implies the existence of a difference in temperature in the water spaces of the boiler. The more rapid the circulation the greater is the tendency for the temperature to be equalised throughout the boiler, so that each increase in rapidity of circulation reduces the force producing the circulation.

The trouble is further aggravated by the modern use of high feed water temperature and sometimes complicated water-wall system. Difficulty has also been experienced at times in arranging for an adequate flow of water to all the water tubes, the high rate of flow in the tubes nearest the fire tending to starve those tubes at the colder end of the boiler heating surface. Conditions may also arise in intermediate tubes which lead to a temporary stoppage or even a reversal of flow.

Some years ago it was thought that these and similar disadvantages would militate against the use of natural circulation in water-tube boilers for very high pressures and ratings, but their effect has been practically eliminated by careful design in the distribution of water flow to the various tube banks

and the provision of suitable downcomer surface. As a result, natural circulation in boilers is now considered to be suitable for steam pressures up to 2,500 lb. per square inch. Moreover, the design of modern high-pressure, high-duty boilers is such that the actual boiler heating surface is used mainly for steam generation and superheating; the sensible heat stage is effected by the use of steam bled from the main turbines to heat the feed water, and by the provision of separate convective surface for feed water and air heating. Even when the type of water-tube boiler employed necessitates the provision of boiler surface in regions of low gas temperatures, the gas temperature difference between the radiant and the convective zones is such that circulation troubles are largely smoothed out.

RADIATION

It has been stated that a hot solid body sends out radiation over a wide range of wave lengths. The manner in which the intensity of radiation varies with wave length and with the temperature of the hot body has an important bearing on heating practice. The most important information required, however, is the total energy that is transmitted to take part in the heating process.

A further significant circumstance arises also in regard to the capacity of a body to absorb radiation. A perfect absorber is known technically as a "black body." A perfect black body is said to have an "emissivity" of 1. "Black body" conditions are approached in a closed furnace chamber, where any radiation at first not absorbed is repeatedly reflected until it is finally absorbed. Even so good an absorber as lamp black is not a perfect black body; it absorbs to the extent of 0·97 of the black body absorption.

A good absorber is also a good radiator. Thus if lamp black, which is equivalent to particles of carbon in a flame, is heated, it will radiate to the extent of 0·97 of the maximum amount theoretically possible. Many bodies such as polished metals are poor absorbers and accordingly also poor radiators, their emissive power being below 0·1 of that of a "black body."

Stefan found experimentally that the total radiant energy emitted by a hot body varied as the fourth power of the absolute temperature of the body. Boltzmann developed the theoretical basis, and the result was the Stefan-Boltzmann law.

The Stefan-Boltzmann law states that when a body of area A square feet and having an emissivity, E, is at $T_1°$ Abs. ($= t + 460°$ F.) and its solid surroundings are at $T_2°$ Abs., the heat radiated, Q, is given by

$$Q = 17{\cdot}3 \times 10^{-10} \times EA(T_1^4 - T_2^4) \text{ B.Th.U. per hour} \quad . \quad . \quad . \quad . \quad . \quad (27)$$

A form convenient for calculation is

$$Q = 0{\cdot}173\ EA \left[\left(\frac{T_1}{100}\right)^4 - \left(\frac{T_2}{100}\right)^4\right] \text{ B.Th.U. per hour} \quad . \quad . \quad . \quad . \quad . \quad (27a)$$

DEFINITION OF SURFACE "A" AND THE CONSTANT "E"

If radiation takes place between two surfaces, one of which is totally enclosed by the other, then the smaller or enclosed surface is referred to as the area A, regardless of the direction in which heat is being radiated. Thus in a combustion chamber containing a fuel bed the area of the grate rather than that of the walls of the chamber is implied.

EFFECTIVE SURFACE AREA

The Stefan-Boltzmann radiation formula has been expressed in terms of B.Th.U. radiated (or absorbed) per square foot of area per hour. The area of the hotter and cooler surfaces involved may not be the same, and generally will not be the same. It is important to note that invariably the expression "rate

of heat transfer per square foot " applies to the area of the enclosed (i.e. the smaller) surface.

The heat transfer per square foot of the larger surface may be found by calculating the total heat exchange from the area of the smaller surface and dividing this by the area of the larger surface.

For a surface having no re-entrant angles, the total area, whether plane or curved, is the effective radiating surface.

For a surface having re-entrant angles, such as irregular-shaped stock in an annealing furnace, or many furnace structures, the increased area due to the concavity must not be included in the total area ; the area that must be taken is defined as the least surface that could be built across the concave portion.

The net radiation between two surfaces is dependent not only upon the radiation constant of the radiating surface, but also upon that of the surface receiving radiation. For two parallel mutually radiating surfaces, having constants respectively E_1 for the hotter and E_2 for the cooler, the net radiation constant is

$$E = \frac{E_1 E_2}{E_1 + E_2 - E_1 E_2} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (28)$$

The problem becomes more complicated in practice because three other laws have to be borne in mind, namely, Kirchoff's and Lambert's laws and the law of inverse squares.

In addition, the relative sizes of the surfaces influence the extent of radiation. Equation (28) gives the minimum value for the emissivity ; the maximum value is E_1. The maximum value represents the heat loss from a hot surface very small compared with its surroundings, under which condition the emissivity of the (cooler) surroundings E_2 has practically no effect on the heat loss. The minimum value represents the heat loss from a substance nearly as big as its surroundings.

KIRCHOFF'S RADIATION LAW

Kirchoff's law states that the radiating capacity of a given body, represented by the radiation constant E, is for a given temperature and a given wave length proportional to the absorbing capacity of the body. In other words a body at a certain temperature will radiate in proportion to its ability to absorb. Thus polished metals absorb only a small percentage of the radiation falling upon them and have a small relative blackness, whereas lamp black being nearly a perfect radiator approaches being a perfect absorber.

LAMBERT'S LAW

The radiation law of Lambert states that the radiation from a surface in a

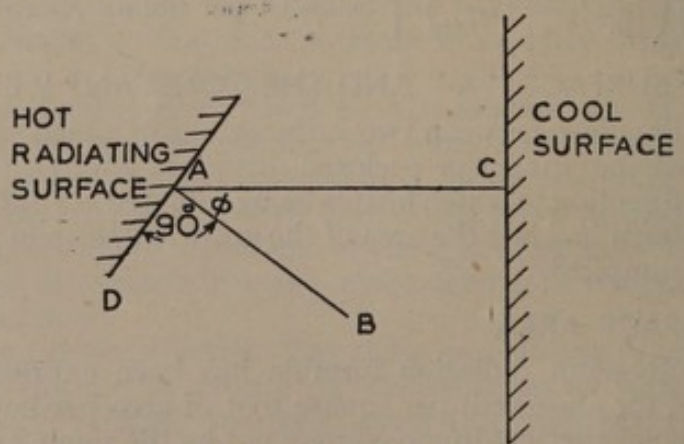

FIG. 41. Lambert's law of radiation

direction at an angle with the surface varies as the cosine of the angle between the direction of radiation and the normal to the surface (Fig. 41). The radiation in the given direction is

$$q = q_n \cos \phi \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (29)$$

where q_n is the radiation normal to the surface.

INVERSE SQUARE LAW

The intensity of radiation from a point source of radiant energy decreases with the square of the distance from the point source.

Another difficulty in the computation of radiation is that some bodies exhibit relative absorption of certain wave lengths.

For the full treatment of a somewhat complicated subject reference must be made to standard works on heat transmission. Actually in practice calculations on the basis of equation 27, using the emissivity constants given in Table 40 serve a useful purpose in many problems without recourse to the more exact treatment referred to above.

TABLE 40. EMISSIVITY OF SOLID SURFACES

Material	Surface conditions	Emissivity (Black body = 1)
Brass	Brightly polished	0·057
	Recently rubbed with emery	0·208
Copper	Polished	0·041
	Etched or scratched	0·094
	Oxidised black	0·788
Aluminium	Polished	0·053
	Rough	0·072
Lead	Grey oxidised	0·284
Iron	Polished nickel-plated	0·059
	Matt nickel-plated	0·114
	Fresh tinned	0·082
	Fresh zinced	0·230
	Grey zinced	0·280
	Freshly rubbed with emery	0·245
	Red rusted	0·694
	Wrought	0·664
	Coarse oxide	0·817
	Cast	0·817
Varnished enamel	Snow-white	0·919
Aluminium	Varnished	0·401
Paper		0·940
Plaster of Paris	½ mm. thick	0·915
Oak wood	Planed	0·941
Brick	Red	0·941
Porcelain	Glazed	0·936
Glass	Polished	0·950
Marble	Light grey, polished	0·943
Aluminium paint	On rough sheet iron	0·38–0·7 average 0·55
Metallic paints	All colours	0·9–0·95
Furnace interior	Closed chambers at uniform temperature—all surfaces	1
Refractory brick	Firebrick	0·75–0·8

The lower the value of the emissivity E, the less is the heat lost from a hot body, or the heat gained by a cool body. If under black body conditions the temperatures are such that H = 100 B.Th.U. per square foot per hour, the heat lost from a cast-iron surface under the same conditions would be 81·7

B.Th.U. per square foot per hour, and that lost from a surface of polished copper would be no more than 4·1 B.Th.U. per square foot per hour.

The experimental fact that the colour of a surface does not necessarily affect its emissivity often leads to confusion and seems contrary to known experience that in hot countries white clothes are found to be cool and dark clothes are uncomfortably hot. The explanation lies in the fact that the emissivity (and therefore the absorptivity) or substances varies with the wave length of the radiation. The low temperature absorptive power given in Table 21 often differs very considerably from the emissivity or absorptivity to solar radiation. Whereas terrestrial sources of radiation are generally between atmospheric temperature and 1,600° C., the solar surface is over 5,000° C.

EFFECT OF TEMPERATURE ON RADIATION

From the Stefan-Boltzmann law it will be seen that the rate of heat transmission by radiation varies as the fourth power of the absolute temperatures. Conduction, on the other hand, varies as the first power, and convection with the 1·25th power. It is thus evident that the higher the temperature difference between the hotter and cooler surfaces, the more important relatively is the transference of heat by radiation.

It is thus an established principle that for rapid transference of heat by radiation, the temperature of the radiating surface should be as high as possible. An application of this principle is the use of preheated air to increase flame temperatures (cf. Fig. 19, Chapter V).

To make this clear by a numerical example, let it be supposed that heat is to be transferred by radiation from a surface at high temperature to a surface

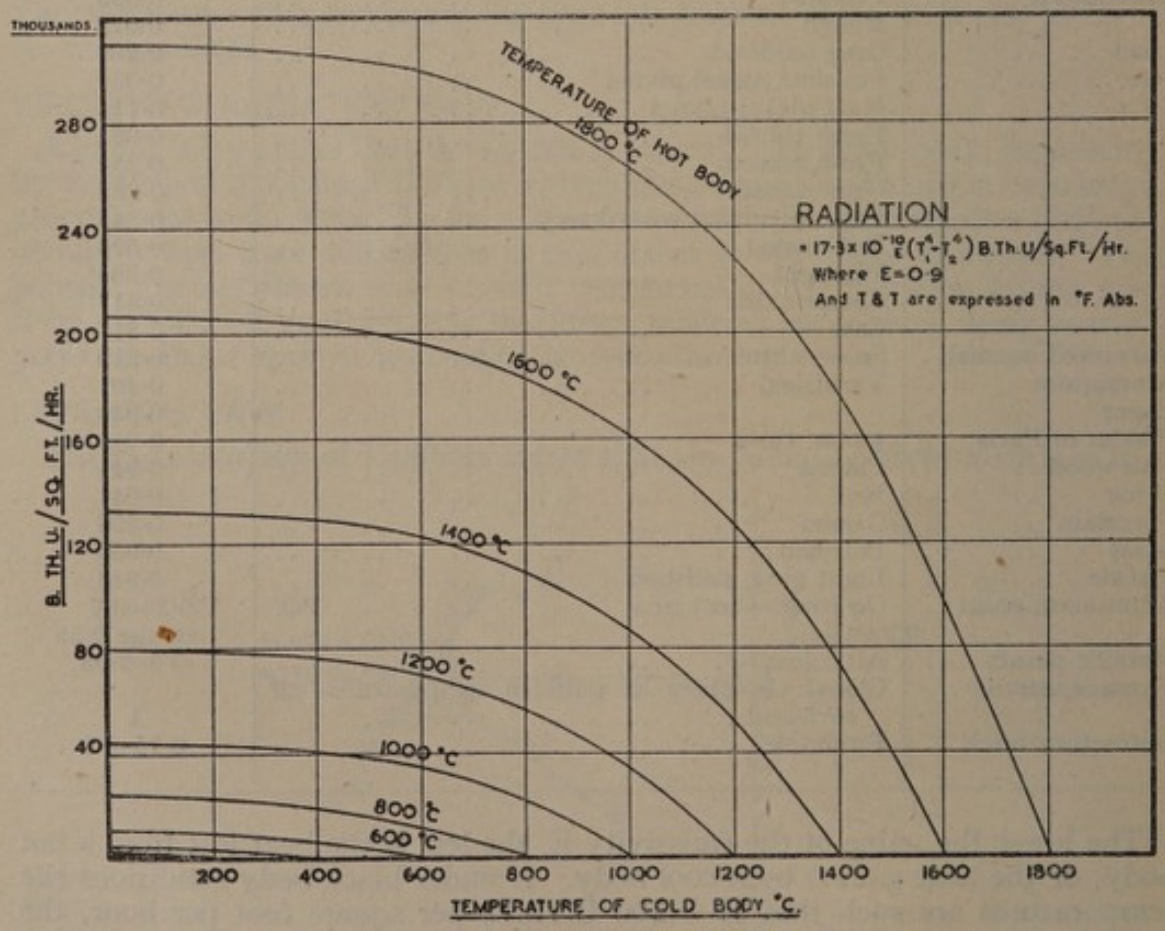

FIG. 42. Heat transfer by radiation from hot surfaces.

at 350° F., the value of E being 0·56. It can then be calculated that the heat transfer is as follows :—

Temperature of hot surface ..	°C.	800	1,000	1,200	1,500
	°F.	1,470	1,830	2,190	2,735
Heat transferred—H, B.Th.U./sq. ft./hr.		13,000	24,400	48,100	101,000
Relative values of H		1	1·88	3·7	7·8

It is thus evident that an increase in the hot surface temperature of less than double, i.e. 800°–1,500 C°., increases the rate of radiation nearly eightfold. Similarly an increase of 50 per cent. from 1,000° to 1,500° C. increases the rate fourfold.

These factors are further illustrated in Fig. 42.

The high rate of radiation from luminous flames is due to the radiating surface of the carbon in these flames, and clearly the higher the temperature to which the carbon is heated, the greater will be the radiating effect.

RADIATION FROM A COKE FIRE

An example of the heat transfer from the surface of an incandescent fuel bed of coke, which is frequently used for heating vessels such as evaporating pans is given in Table 41. From practical observations it has been established that temperatures of the surface of the fuel bed corresponding to rates of combustion of coke of 20 up to 106 lb. per square foot per hour are those given in the second line of the table, namely temperatures ranging from 1,232° to 1,509° C. The temperatures of the heat-absorbing surfaces range between 200° and 400° C. in such practice. The heat emission, according to the fourth power law, has been calculated for the two temperatures concerned by the use of equation 27 and an emissivity coefficient of 0·9. The total potential heat in the coke corresponding to the rates of combustion applying are also shown, together with the radiant heat emitted reckoned as a percentage of this total value. The calorific value of the coke has been taken as 12,500 B.Th.U. per lb. It is to be noted that at the lower rates of combustion the direct radiant heat is approximately 33 per cent. of the total available heat, but that at the higher rates of combustion the proportion of radiant heat approaches a much lower value, e.g. 12 per cent. This is due to the fact that at the higher rates of combustion the mass of the gas emerging from the fuel bed is increased, and carries with it a higher proportion of the total available heat as either potential or sensible heat in the hot gases.

TABLE 41. RADIATION FROM A COKE FIRE

Rate of combustion—lb./sq. ft./hr.	20	51	71	106
Temperature of coke bed—° C.	1,232	1,383	1,442	1,509
Heat emission by radiation to surface at 200° C. ..	83·3	122·3	140·9	164·3
at 400° C. ..	80·5	119·5	138·2	161·5
(Thousands B.Th.U./ sq. ft./hr.)				
Rate of total heat release from coke .. .. (Thousands B.Th.U./sq. ft./hr.)	250	637·5	887·5	1,325
Per cent. of heat radiated to total heat developed				
Cold surface at 200° C.	33·3	19·2	15·9	12·4
at 400° C.	32·2	18·8	15·6	12·2

GAS RADIATION

RADIATION OF CARBON DIOXIDE AND WATER VAPOUR IN FURNACE GASES

A further type of heat transference, which was ignored until relatively recent years is the direct radiation from hot furnace gases. It was found that in applying the ordinary methods of computation of heat exchange by convection to fire-tube boilers or to certain types of industrial furnaces, the surface conductances were of the order of magnitude of 10 (B.Th.U./sq.ft./hr./°F.). In practice surface conductances of a much higher order were observed. This led Schack to look for the cause of the discrepancy in the infra-red radiation of furnace gases, and the radiative capacity of carbon dioxide and water vapour always present were shown to account for the high practical values of the conductances.

Hot gases radiate heat even when no combustion is taking place. Good radiating gases include carbon dioxide, steam and hydrocarbons. Carbon monoxide has some radiating power, but is a relatively poor radiator. From a practical point of view hydrogen, oxygen and nitrogen are non-radiators. The radiation effect of a non-luminous flame is accordingly governed by its carbon dioxide and water vapour content.

The transfer of heat from the gas to the wall is an exchange process, since heat is also returned from the wall to the gas. On account of the thickness of the layer of gas some of the heat radiated by molecules of carbon dioxide and water are absorbed by other molecules before it can reach the wall. The course of the heat exchange in a combustion chamber containing the non-luminous products of combustion is as follows :—

(1) The gas radiation to the walls of the chamber depends upon the temperature of the gaseous atmosphere, and on the nature and amount of radiating gas between the walls, that is to say, on the concentration of the carbon dioxide and steam present, on the thickness of the gas layer and the pressure and temperature of the gas.

(2) Of the heat radiated by the gases only the fraction E, the emissivity of the material of the wall at the temperature ruling, is retained by the wall, the remainder being reflected back.

(3) At the same time there is heat exchange between the hot walls, radiating according to the normal Stefan-Boltzmann law, if temperature differences exist. Only a relatively low proportion of this heat is absorbed by the gases. If the wall is hotter than the gas the gas becomes heated; if the gas is hotter than the wall, the wall becomes heated by gas radiation.

The most convenient means of evaluating gas radiation is that due to H. C. Hottel.*

The method involves the evaluation of the gas emissivity coefficients and applying them to a fourth power law in a manner which corrects for the influence of the gas composition in regard to the content of carbon dioxide and water vapour, and for the thickness of the gas layer.

The nomenclature used in describing the method, which involves the use of the diagrams, Figs. 43–46, is as follows :—

q = rate of heat transfer due to gas radiation, B.Th.U. per hour.
A = area of surface exchanging radiation with the gas, square feet.
T_G = absolute temperature of the gas ($460 + t_g$ °F.).
T_S = ,, ,, ,, surface ($460 + t_s$ °F.).
E_G = emissivity of gas, dimensionless.
α_G = absorptivity of gas, ,,

* " Heat Transmission," W. H. Adams (McGraw-Hill, 1942), p. 64.

p_c = partial pressure of carbon dioxide, atmospheres.
p_w = ,, ,, ,, water vapour, atmospheres.
L = beam length of gas radiation, feet.
E_s = emissivity of heat exchanging surface.
E'_s = effective emissivity = $\frac{E_s + 1}{2}$

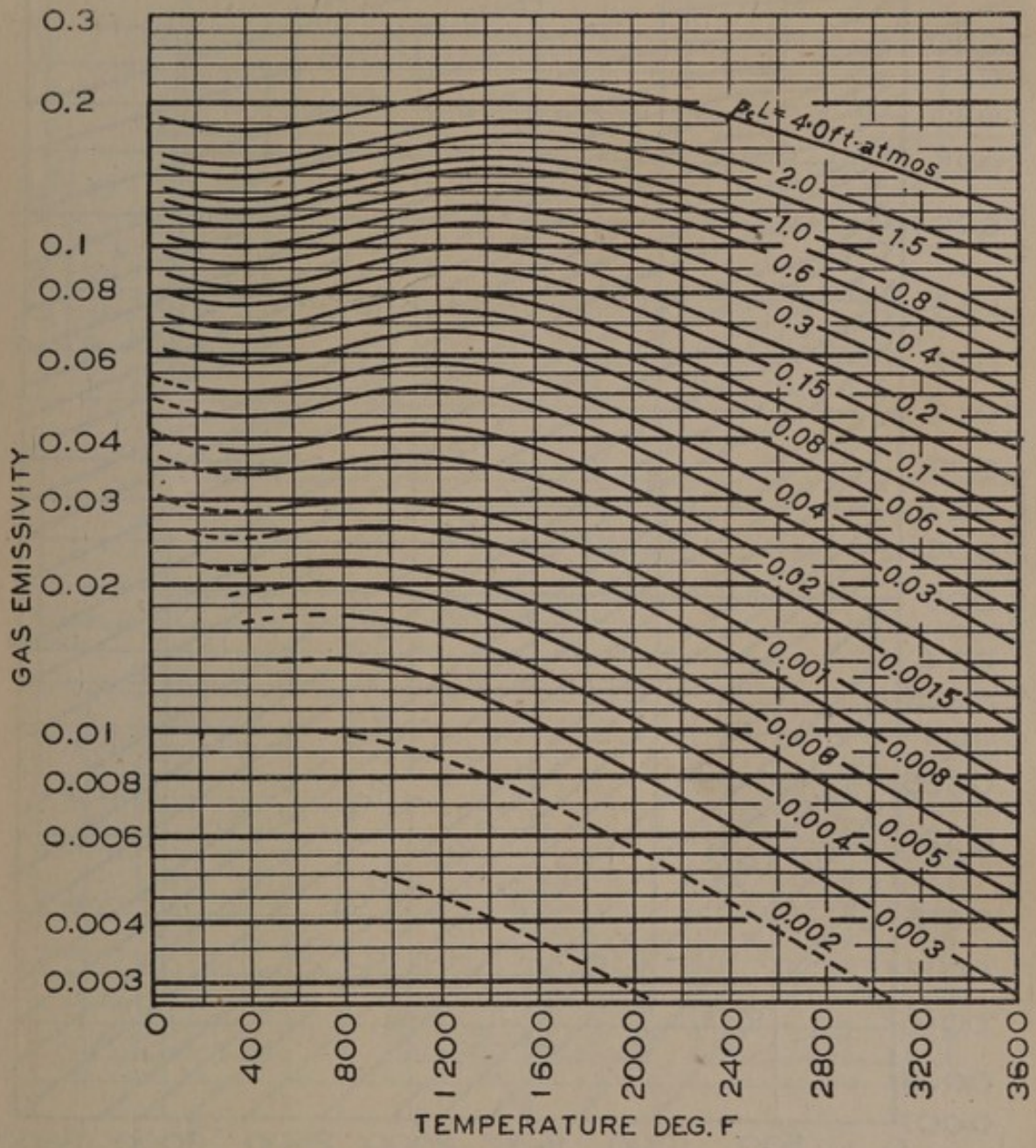

FIG. 43. Emissivity of CO_2 for various values of p_cL.

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The complete calculation is complicated, but although it can be shortened in certain circumstances it will be given in full here step by step.

A relation, originally developed by Schack and amended by Hottel, exists between gas emissivity, temperature, and the product of the partial pressure of the gas and the thickness of the gas layer that is emitting the radiation. This relationship is the basis of Figs. 43 and 44.

From these figures, subject to certain corrections for which Figs. 45 and 46 are used, it is possible to compute the coefficient of emissivity of the gas E_G.

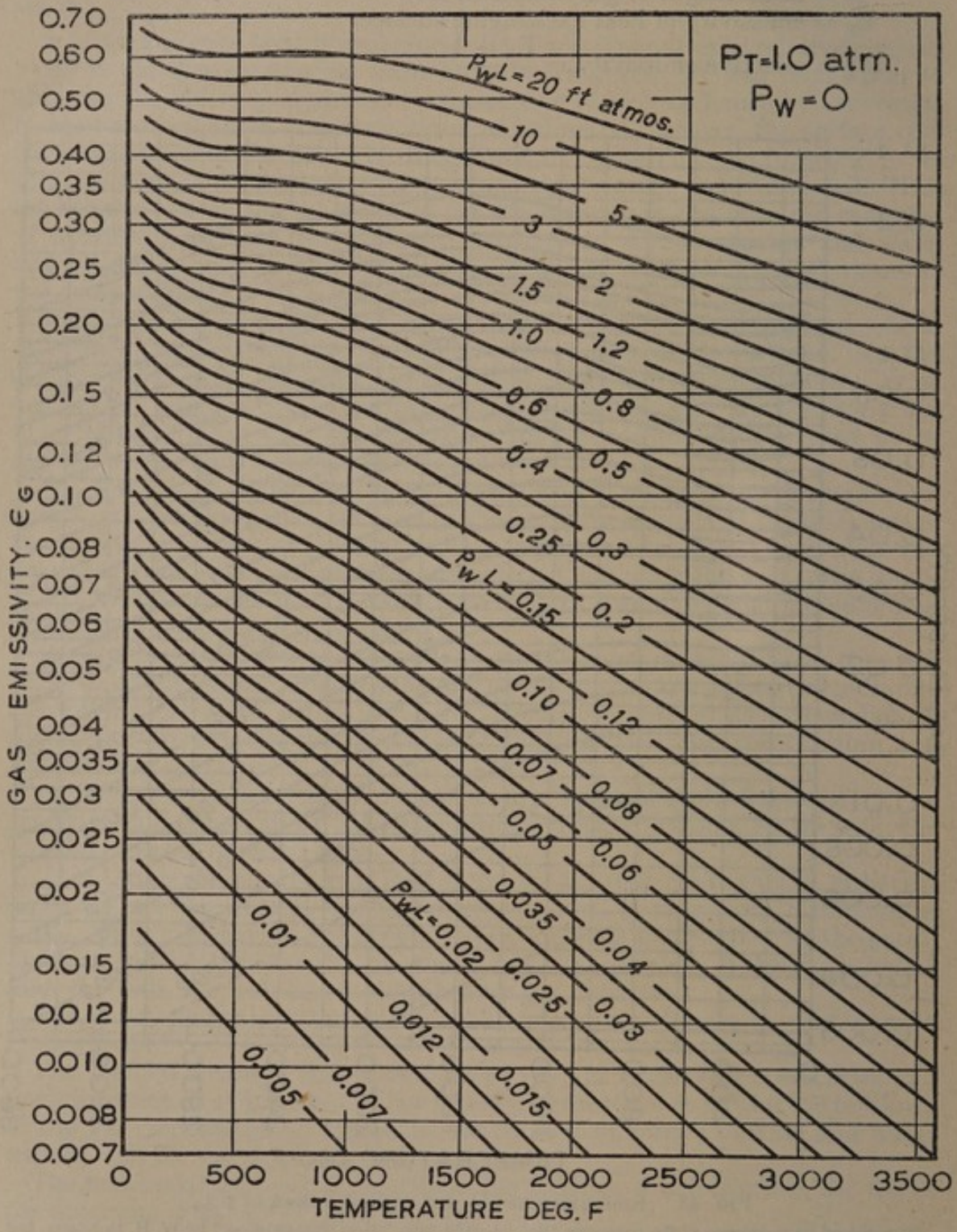

FIG. 44. Emissivity of water vapour for various values of p_wL.

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As will be explained later, the effective emissivity of the surfaces must also be taken into account, this being E'_s.

The gases absorb radiation emitted from the walls, and the value of the coefficient of absorption of the gas for this radiation, α_G, must also be determined.

These coefficients are then used in equations 27 or 27a as follows :—

$$H = 0{\cdot}173 \,.\, E'_s \left[E_G \left(\frac{T_G}{100} \right)^4 - \alpha_G \left(\frac{T_s}{100} \right)^4 \right] \text{B.Th.U./sq.ft./hr.} \quad . \quad . \quad . \quad . \quad (39)$$

The steps by which this calculation is made will now be given in order.

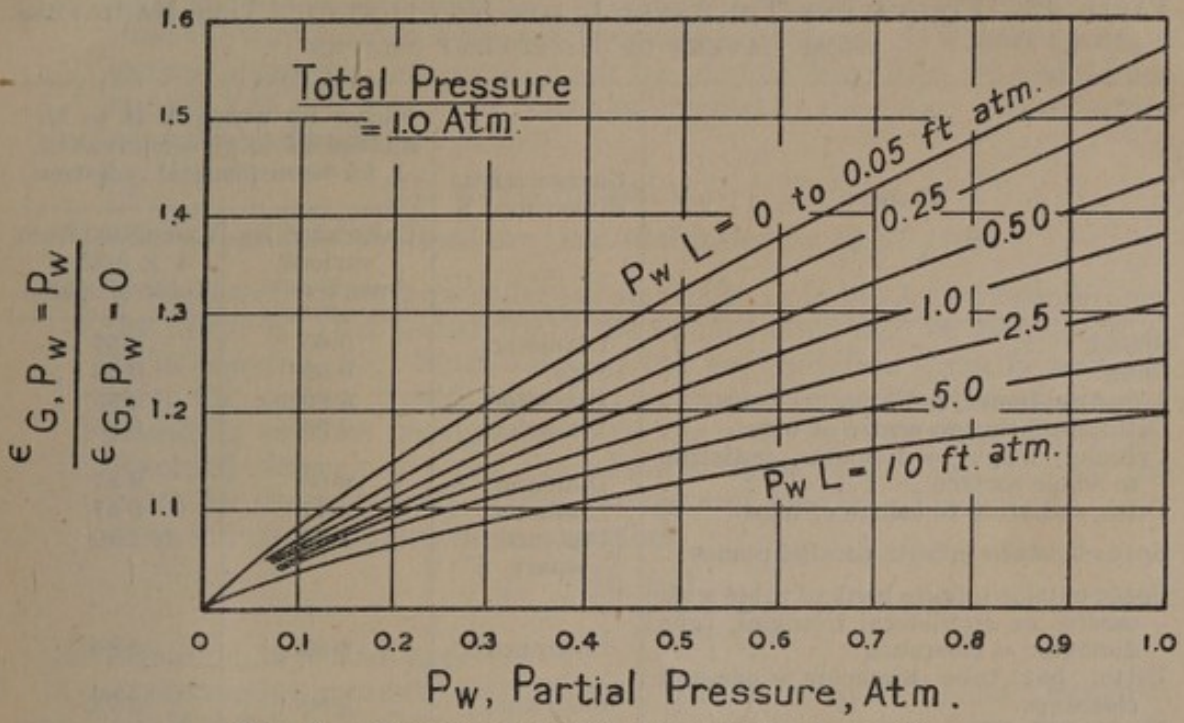

FIG. 45. Chart for evaluating gas radiation factor, C_1.

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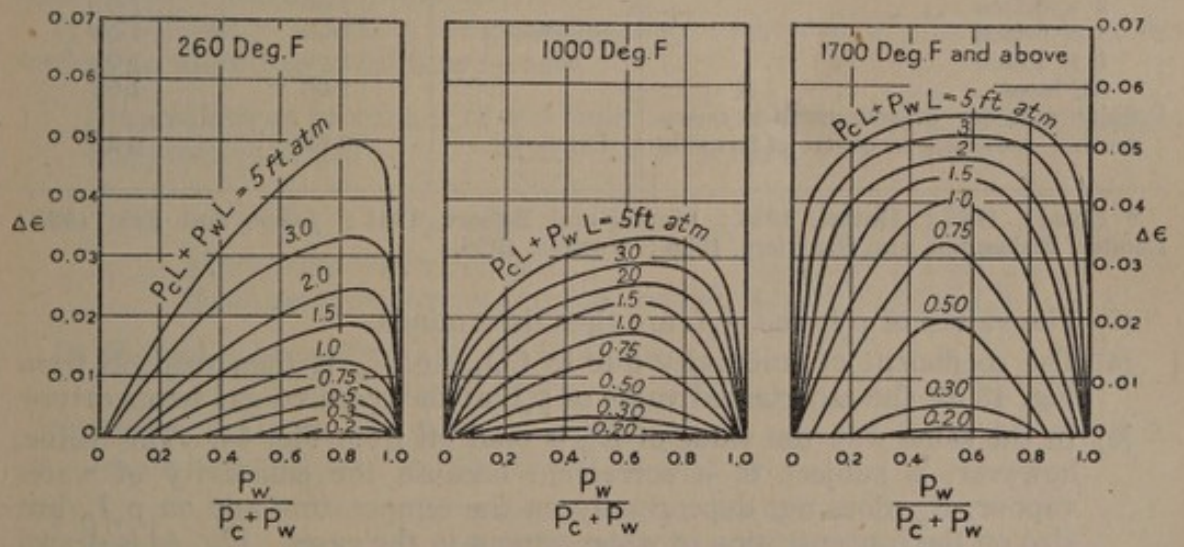

FIG. 46. Chart for evaluating the absorption of gas radiation by the gas itself.

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(1) The partial pressure of the radiating gases CO_2 and H_2O in atmospheres is determined.

(2) The effective thickness of the gaseous layer is determined. In Figs. 43 and 44, the same thickness of gas in all directions is assumed. This is not generally the case and the effective thickness, L, is found from the general expression

$$L = \frac{4 \times \text{vol. of gas space}}{\text{area of bounding walls}} \times 0{\cdot}85.$$

Table 42 enables this value to be determined for several industrially important shapes.

TABLE 42. EQUIVALENT THICKNESS L FOR NON-LUMINOUS GAS RADIATION FROM LAYERS OF DIFFERENT SHAPES

Shape	Characteristic dimension, X	Factor by which X is to be multiplied to give equivalent L for hemispherical radiation	
		Calculated by various workers *	Calculated from $4 \times 0{\cdot}85 \times$ volume ÷ area
Sphere	Diameter	0·60	0·57
Cube	Side	0·60	0·57
Infinite cylinder, radiating to walls	Diameter	0·90	0·85
Ditto, radiating to centre of base	Diameter	0·90	0·85
Cylinder, height = diameter, radiating to whole surface	Diameter	0·60	0·57
Ditto, radiating to centre of base	Diameter	0·77	0·57
Space between infinite parallel planes	Distance apart	1·80	1·70
Space outside infinite bank of tubes with centres on equilateral triangles, tube diameter = clearance	Clearance	2·80	2·89
Ditto, but tube diameter = one-half clearance	Clearance	3·80	3·78
Ditto, with tube centres on squares, and tube diameter = clearance	Clearance	3·50	3·49
Rectangular parallelopiped, 1 × 2 × 6, radiating to :—			
2 × 6 face	Shortest edge	1·06	1·01
1 × 6 face	Shortest edge	1·06	1·05
1 × 2 face	Shortest edge	1·06	1·01
All faces	Shortest edge	1·06	1·02
Infinite cylinder of semicircular cross-section, radiating to centre of flat side	Diameter	0·63	0·52

* Eckert, 1937; Hottel, 1927; Hottel and Egbert, 1941; Jakob and Erk, 1933; Lander, Fishenden, and Saunders, 1936; Nusselt, 1926.

(3) The values of p_cL and p_wL are then determined.

(4) The coefficient of emissivity due to CO_2, i.e. E_c, is then read off from Fig. 43 for the ascertained value of p_cL at the required gas temperature.

(5) In the same way the value of E_w is read off from Fig. 44. This value, however, is subject to a correction because the emissivity of water vapour, E_w, does not depend only on the temperature and on p_wL, but also on the concentration of water vapour in the gases. Fig. 44 is drawn for the ideal condition of $p_w = 0$, and allowance for the real value of p_w must be made by using Fig. 45, the value of E_w read from Fig. 44 being multiplied by a factor C_1 read from Fig. 45.

(6) The combined values of E_c and E_w represent the radiating power of the gas, but since substances that radiate also absorb radiation, a further correction must be made for the radiation absorbed by the gas. This correction, ΔE, is read from Fig. 46.

Then, $$E_G = E_c + E_w - \Delta E.$$

(7) By exactly similar calculations, the value of the absorptivity coefficient,

$$\alpha_G = \alpha_c + \alpha_w - \Delta\alpha,$$

is determined, using the temperature of the surface, not that of the gas, when reading the coefficients from Figs. 43 and 44.
This method of determining α_G is correct if the gas is appreciably hotter than the surface and covers most furnace conditions. If a more accurate method is needed, the values used for reading from Figs. 43 and 44 are not p_cL and p_wL, but these values multiplied by the ratio : (temperature of surface/temperature of gas)

i.e. $$p_cL \times (T_S/T_G) \text{ and } p_wL \times (T_S/T_G)$$

The values thus obtained are then multiplied by $(T_G/T_S)^{0.65}$.

(8) If the bounding surfaces differ substantially from black body conditions, the factor E'_S, the emissivity of the surface must also be introduced as as in equation 29. Since some of the gas radiation initially reflected from the surface would have further opportunity for absorption at a bounding surface, because much of the reflected radiation passes unabsorbed through the gas, the effective value of E'_S lies between the value for the emissivity of the surface and that of a black body. For most industrial applications, therefore,

$$E'_S = \frac{E_S + 1}{2}$$

(9) Equation 39 is then applied.

A comprehensive account of this subject will be found in " Heat Transmission " by McAdams (McGraw-Hill, 1942). Table 42 is taken from the paper on heat transmission by Professor Lander (*loc. cit.*).

The method of calculation will be made clear by an example taken in the same steps as the description above.

It is required to ascertain the gas radiation to the crown of a furnace chamber under the following conditions :—

Dimensions of furnace : 12 feet long × 4 feet wide × 2 feet mean height
Composition of gas : CO_2 13·2 per cent.
H_2O 12·1 ,, ,,
O_2 and N_2 74·7 ,, ,,
Mean temperature of gases : 2,000° F. (1,093° C.)
Mean temperature of furnace crown : 1,500° F. (816° C.)

(1) Partial pressures : CO_2 0·132 atmos.
H_2O 0·121 ,,

(2) From Table 42, the shortest edge is 2 feet, and the factor by which it must be multiplied is 1·06, thus $L = 2·12$.

(3) $$p_cL = 0·132 \times 2·12 = 0·28$$
$$p_wL = 0·121 \times 2·12 = 0·256$$

(4) E_c (from Fig. 43) = 0·091.

(5) E_w (from Fig. 44) = 0·084.
The factor, C_1, from Fig. 45 is 1·08.
$E_w = 0·084 \times 1·08 = 0·091$.

(6) Correction for gaseous absorption (Fig. 46) $\Delta\alpha = 0·022 = \Delta E$.
Then,

$$E_G = 0·091 + 0·091 - 0·022 = 0·160.$$

(7) Determination of α_G.

$\alpha_c = p_c L \times (T_S/T_G) = 0{\cdot}28 \times 1{,}960/2{,}460 = 0{\cdot}223$

Emissivity at 1,500° F. read from Fig. 43 $= 0{\cdot}10$

$\alpha_c = 0{\cdot}10\ (2{,}460/1{,}960)^{0{\cdot}65} = 0{\cdot}116$

$\alpha_w = p_w L \times (T_S/T_G) = 0{\cdot}256 \times 1{,}960/2{,}460 = 0{\cdot}204$

Emissivity at 1,500° F. read from Fig. 44 $= 0{\cdot}090$

$\alpha_w = 0{\cdot}09\ (2{,}460/1{,}960)^{0{\cdot}65}\ C_1 = 0{\cdot}104 \times 1{\cdot}08$ (from 5)

$= 0{\cdot}112$

From Fig. 46, $\Delta\alpha = \Delta E$ at 1,500° F.

$= 0{\cdot}02$

Whence,

$$\alpha_G = 0{\cdot}116 + 0{\cdot}112 - 0{\cdot}020$$
$$= 0{\cdot}208$$

(8) E_S' for a furnace interior may be taken as 0·9

$H = 0{\cdot}173 \times 0{\cdot}9\ [0{\cdot}160\ (2{,}460/100)^4 - 0{\cdot}208\ (1{,}960/100)^4]$

$= 4{,}343$ B.Th.U./sq.ft./hr.

Hence for these conditions, the coefficient of gas radiation is 4,343/500 = 8·686 B.Th.U./sq.ft./hr./°F., exclusive of the effect of radiation transmitted to the roof by refractory surfaces.

RADIATION FROM LUMINOUS FLAMES

Non-luminous flames transmit energy chiefly by the infra-red radiation of carbon dioxide and water vapour. Luminous flames are produced by restricting the air supply so that the hydrocarbons present in the gas in the absence of sufficient oxygen for combustion are cracked to produce solid particles of carbon. The combustion of a luminous flame is consequently more protracted than that of a similar non-luminous flame; that is to say the flame arising from a similar flux of combustible matter is longer. The small particles of carbon absorb heat from the surrounding atmosphere of hot gas and radiate energy to the enclosing surface.

The effect of luminous radiation from flame presents a more difficult subject in which to relate any quantitative analysis of the heat emission to the effect produced in the furnace. Lindmark and Edenholm have approached the subject by an application of the fourth power radiation law to the incandescent particles of carbon in flames produced by oil and by pulverised coal. The inflation of the coal particles during combustion, producing cenospheres having a volume as great as eightfold that of the parent particles, has been associated with a favourable influence on flame radiation from the pulverised fuel.

Lent and Thomas, investigating the influence of benzole added to blast furnace gas under conditions so regulated as not to raise the flame temperature, found that radiation from the flame was four times that emitted from the non-luminous flame, and practically the same as from a black body. The value of luminosity has been appreciated to the extent of injecting hydrocarbons into lean gases. As the flame thickness, or the density of suspended carbon, decreases the emitted radiation decreases, and therefore the apparent radiation temperature falls and differs to a greater extent from the true temperature.

Schack described an instance of a rolling mill furnace heated by coke-oven gas in which, although the rendering of the flame luminous produced a drop of temperature of 180° F., the surface temperature of the crown increased by 145° F. At first in such a furnace the heat transfer to the relatively cold walls is high and the temperature of the waste gas low. Ultimately, however, the temperature of the indirect heating surfaces increases, and the heat transferred to these surfaces decreases as a result of the decreased difference of temperature between the combustion gases and the heat surfaces. Then the temperature

of the waste gas correspondingly increases. These observations gave rise to the suggestion that improvement of practice might arise from periodic carburetting of the flame. This is, of course, what happens in hand fired coal-burning furnaces.

The analysis of total heat transfer in furnaces operating at reheating temperatures (1,150°–1,300° C.) has shown the following distribution between the various modes of heat transmission.

	Percentage of total heat transfer
Direct gas radiation	54
Furnace wall and crown radiation	34
Convection	12

Heat transfer by convection has only a subordinate influence in such furnaces.

HEAT EXCHANGERS

Heat exchangers are usually classified into two groups :—

(1) those in which heat is transferred from one flowing material to another through a solid wall ; and

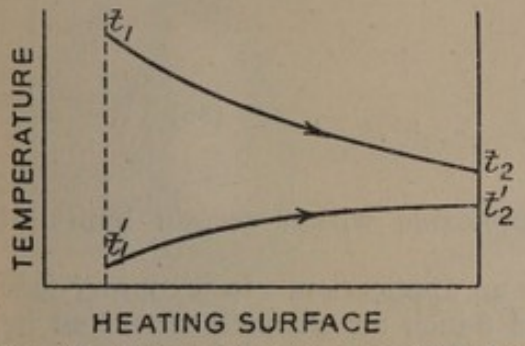

FIG. 47. Temperatures in a parallel-flow heat exchanger.

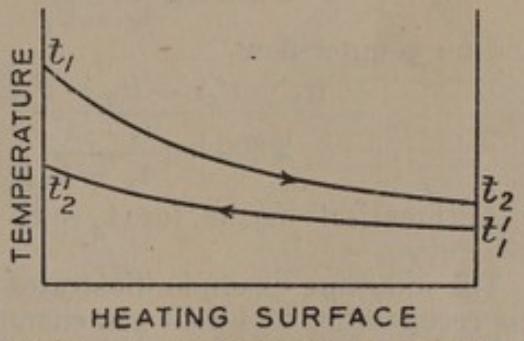

FIG. 48. Temperatures in a counter-flow heat exchanger.

(2) those in which heat is stored for some time and the heating gases and heated gases alternate in direction through the heat storage appliance.

To the first class belong recuperators, condensers, air heaters and economisers ; to the second regenerators, usually applied to the preheating of gas and air in high temperature melting furnaces and in blast furnace stoves.

(1) *Heat Exchangers Without Heat Storage.* Heat exchangers may be operated either in parallel or counter-flow, and the temperature conditions along the heating surface are indicated in Figs. 47 and 48.

One material may flow at right angles to the second and the character of the flow is then referred to as cross-flow.

Under conditions of constant heat transmission, that is with no marked fluctuations of temperature, the simplest case by way of introduction only to a complicated study is that of two gases flowing through a heat exchanger with the temperature conditions as specified graphically in Fig. 47.

If q = heat transfer, B.Th.U. per hour.
H = transmittance, B.Th.U./sq.ft./hr./°F.
A = total heating surface, square feet.
$t_g - t_a$ = mean temperature difference, between the flowing gases, °F.
w = mass flow of the hotter gas, lb. per hour.
c = specific heat of the hotter gas, B.Th.U./lb./°F.
w' = mass flow of the colder gas, lb. per hour.
c' = specific heat of the colder gas, B.Th.U./lb./°F.

Then the heat transfer referred to the gas giving up heat is

$$q = wc\,(t_1 - t_2) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (30)$$

and to the gas receiving heat

$$q = w'c'\,(t'_2 - t'_1) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (31)$$

The heat transfer is also governed by the transmittance H, the heating surface and the mean temperature difference between the flowing gases. Accordingly

$$q = H.A.\,(t_g - t_a) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (32)$$

Since the heat given up must be equal to the heat absorbed the right-hand sides of the equations 30, 31 and 32 must be equal to each other and

$$HA\,(t_g - t_a) = wc\,(t_1 - t_2) = w'c'\,(t'_2 - t'_1) \quad . \quad (33)$$

from which the heating surface A is given by either

$$A = \frac{wc\,(t_1 - t_2)}{H\,(t_g - t_a)} \quad \text{or} \quad \frac{w'c'\,(t'_2 - t'_1)}{H\,(t_g - t_a)} \quad . \quad . \quad . \quad . \quad . \quad (34)$$

The mean temperature difference $t_g - t_a$ is for parallel flow

$$= \frac{(t_1 - t'_1) - (t_2 - t'_2)}{2{\cdot}303 \log \frac{t_1 - t'_1}{t_2 - t'_2}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (35)$$

and for counter-flow

$$= \frac{(t_1 - t'_2) - (t_2 - t'_1)}{2{\cdot}303 \log \frac{t_1 - t'_2}{t_2 - t'_1}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (36)$$

Arithmetical means for $t_g - t_a$ are applicable within certain limits of accuracy.

The following example illustrates briefly an application. In a counter-flow air recuperator of 4-inch gas channels and 3-inch air channels separated by refractory tiles 1¼ inches thick, waste combustion gases containing 13 per cent. CO_2, 6 per cent. H_2O and 5 per cent O_2 are used to preheat air. The volumes of waste gases and air flowing are respectively 35,000 and 32,000 cubic feet per hour measured at 60° F. The waste gas enters the recuperator at 1,470° F. (799° C.) and air entering at 70° F. is required to be preheated to 1,000° F. What is the temperature of the waste gases at the exit from the appliance and what is the area of heating surface required ?

The problem involves a knowledge of the heat capacity of the gases and air in the ranges of temperature involved, and of the overall coefficient of heat transmittance H, which normally has to be calculated from the surface conductances by convection and gas radiation on the waste gas side, the conductance through the refractory tile and the surface conductance on the air side. For the sake of simplicity in explanation, the value of H is taken to be 1 B.Th.U./sq.ft./hr./°F., and the volumetric specific heats respectively 0·0215 and 0·0197 B.Th.U./cu.ft./hr. for waste gas and air.

Accordingly working in terms of volumes of gas instead of weights which were used in equations 32–34

$$q = 0{\cdot}0215 \times 35{,}000\,(1{,}470 - t_2) = 0{\cdot}0197 \times 32{,}000\,(1{,}000 - 70)$$
$$= 586{,}270 \text{ B.Th.U. per hour.}$$

from which $t_2 = 691°$ F.

From the complete knowledge so gained of the temperature conditions $t_g - t_a$ (equation 36) becomes 539° F. and

$$A = \frac{q}{H\,(t_g - t_a)} = \frac{586{,}270}{1 \times 539} = 1{,}087 \text{ square feet.}$$

(2) *Heat Exchangers with Heat Storage Regenerators.* A regenerator usually consists of a mass of chequer brickwork built in a separate chamber through which the waste gases from the furnace descend to heat the chequers. They then pass to a reversing valve on their way to the chimney. A regenerative system in reheating and melting furnaces and certain types of kiln is applied to furnaces having two sets of ports, one at each end of the furnace chamber, through which the direction of the gases is periodically reversed.

When air alone is preheated there are two regenerator chambers, one serving to take up the waste heat from the hot gases, and the other to preheat the air

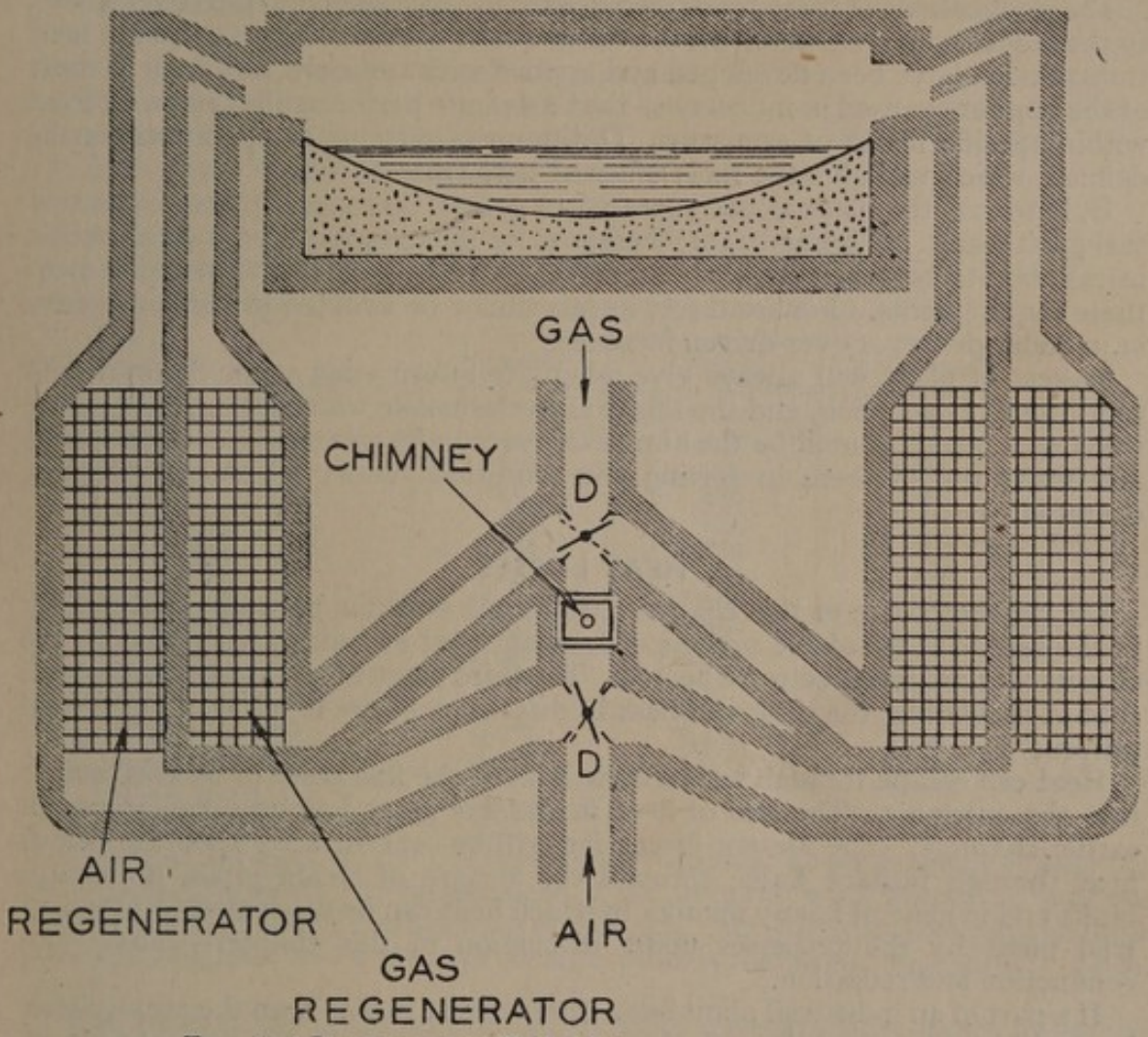

FIG. 49. Diagrammatic illustration of a regenerative furnace.

for combustion. Each regenerator has accordingly a heating cycle on waste gas and a cooling cycle on air. A blast furnace stove operates on the same principle except that gas is burnt in a combustion chamber built into the stove to provide the heat storage in the chequer brickwork during the heating cycle, the combustible gas being shut off during the cooling cycle when the blast is being preheated.

Fig. 49 depicts a regenerative system applied to an open-hearth steel furnace, or a glass furnace.

Here both gas and air are preheated. With the valves set as shown, air and gas flow up the left-hand regenerator where they are preheated by the hot chequer bricks to perhaps 900°–1,000° C. Each issues through its appropriate ports and combustion takes place in the furnace. The products of combustion

pass down through the right-hand regenerators, giving up their heat to the chequers, and then flow to the chimney. After a suitable period, the valves are changed to the dotted positions, when the flow of the gas and air is reversed. The cold gases now pass through the hot right-hand regenerators and the hot products of combustion leave the system through the cooled left-hand regenerators.

The problem of calculating the heat exchange is beyond the scope of this volume, since its solution involves the study of the fluctuating flow of heat. The subject has been adequately dealt with by Heiligenstaedt, Schack and Rummel, and applications to practice have been given by Barnes and Sarjant.

The implications of these brief references to the calculation of heat exchangers to the reader not concerned with the technique of design is that the laws of heat transmission have been developed and applied with sufficient certainty to most of the appliances used in industry so that a definite performance can be forecast within specific limits of accuracy. Optimum conditions of operation can be defined, which will give the most efficient performance.

If, however, the plant is not operated to satisfy these conditions, a waste of fuel must result. It is important, therefore, for the duty required from a specific installation to be clearly understood, and the load on the plant adjusted to meet these requirements. For example, losses cannot be avoided in banked boilers or in under-driven or over-driven furnaces.

Makers of plant will always give adequate information upon the optimum conditions of operation, and the limits of performance which give rise to excessive losses, and it should be the aim of the user to be aware of these conditions and make use of them in setting the conditions under which the plant is operated.

HEAT LOSSES

During the course of this chapter attention has so far been directed mainly to the second stage of the process of heating, that of transmitting heat to the objects that are required to be heated. The third stage must now be considered, that of preventing the escape of heat in directions where it will serve no useful purpose.

Heat can escape through the sensible heat of the flue gases or by conduction into the outer air. The loss of heat in the flue gases has been considered in earlier chapters. The present discussion will be concentrated upon the loss of heat through furnace walls, through the surface of steam pipes, hot water tanks and in general in any manner in which heat can be dissipated from industrial plant by the processes under discussion in this chapter—convection, conduction and radiation.

If a part of an industrial plant has a temperature higher than the atmosphere, heat will be conducted through the casing or outer wall by the normal process, depending upon temperature difference between the hot and cold surfaces of the wall, and the thermal conductivity of the material.

This heat will raise the temperature of the outer sides of the wall or casing above that of the atmosphere and its surroundings. The outer surface will then lose heat by convection to the air and by radiation to the cooler objects around in the manner previously described.

It is an essential characteristic of this form of heat loss that the quantity of heat conducted through the wall or casing must equal exactly the quantity of heat radiated and convected from the outer surface. If the steady state has not been reached and this condition is not fulfilled, the temperature of the outer surface will change until the condition is fulfilled; at this point the steady state will be set up. Some account of the industrial significance of heat losses in the unsteady state will be found in Chapter XVIII.

When the steady state has been reached it is thus possible to calculate the surface temperature of the wall. An example will make the methods of calculation clear and in order to give an example of practical value attention will be directed to the insulation of a furnace.

HEAT LOSSES FROM SURFACES

The total radiation and convection is obtained by a combination of equations 15 and 27.

$$H_c + H_r = C(t_1 - t_2)^{1\cdot25} + 17\cdot3 \times 10^{-10} \times E(T_1^4 - T_2^4), \text{B.Th.U./sq.ft./hr.} \quad (37)$$

As stated above, the temperature of the hot surface is normally governed not only by the character of the heat loss from the external surface, but also by the heat flux through the mass forming the wall.

The calculated values of the combined radiation and convection loss from a furnace wall in a still atmosphere at 70° F. are shown in Table 43 (J. S. F. Gard, *J. Inst. Fuel*, X, 250).

TABLE 43

Surface temp. ° F.	Heat loss B.Th.U./sq. ft./hr.		
	Upwards	Vertical	Downwards
100	54	48	41
150	172	152	128
200	316	280	236
250	494	436	360
300	696	618	526
350	926	825	708
400	1,190	1,070	930
450	1,510	1,360	1,190
500	1,850	1,680	1,480
550	2,250	2,050	1,810
600	2,700	2,470	2,200
650	3,200	2,940	2,640
700	3,770	3,480	3,140
750	4,370	4,050	3,680

Table 43 is based on the coefficient of emissivity, E, for firebrick (Table 40), and would require to be recalculated if the material with which the wall was coated had a different coefficient of emissivity. Actually the governing factor in determining heat losses through furnace walls is the insulating character of the wall.

The effect of the coefficient of emissivity of the material is shown in Fig. 50. By using a bright unscratched metal surface (e.g. nickel plated) the value E can be reduced to about 0·05 (cf. Table 40), and the heat loss by radiation becomes reduced to very small dimensions. The heat loss by convection is unaffected and the total heat loss is the sum of the convection and radiation effects for any given temperature. This device is employed in hospitals and other places where it is possible to maintain the surface in something approaching its initial bright condition.

The convection loss for the same temperature conditions is greatly affected by the air currents prevailing. Table 43 and Fig. 50 refer to still air. A suggested approximate expression to take air movement into account is

$$\text{loss of heat by convection} = (1\cdot2 + 0\cdot19V)(t_s - t_a), \text{B.Th.U./sq.ft./hr.} \quad (38)$$

where V = air velocity in feet per second

t_s, t_a = temperatures of surface and air, °F.

Table 44 (A. C. Pallot, *J. Royal Soc. Arts*, XCI, 123) shows another (and slightly higher) estimate of the amount of heat transfer in still and moving air respectively from a vertical surface (E = 0·9) maintained at various temperatures, the air being at 70° F.

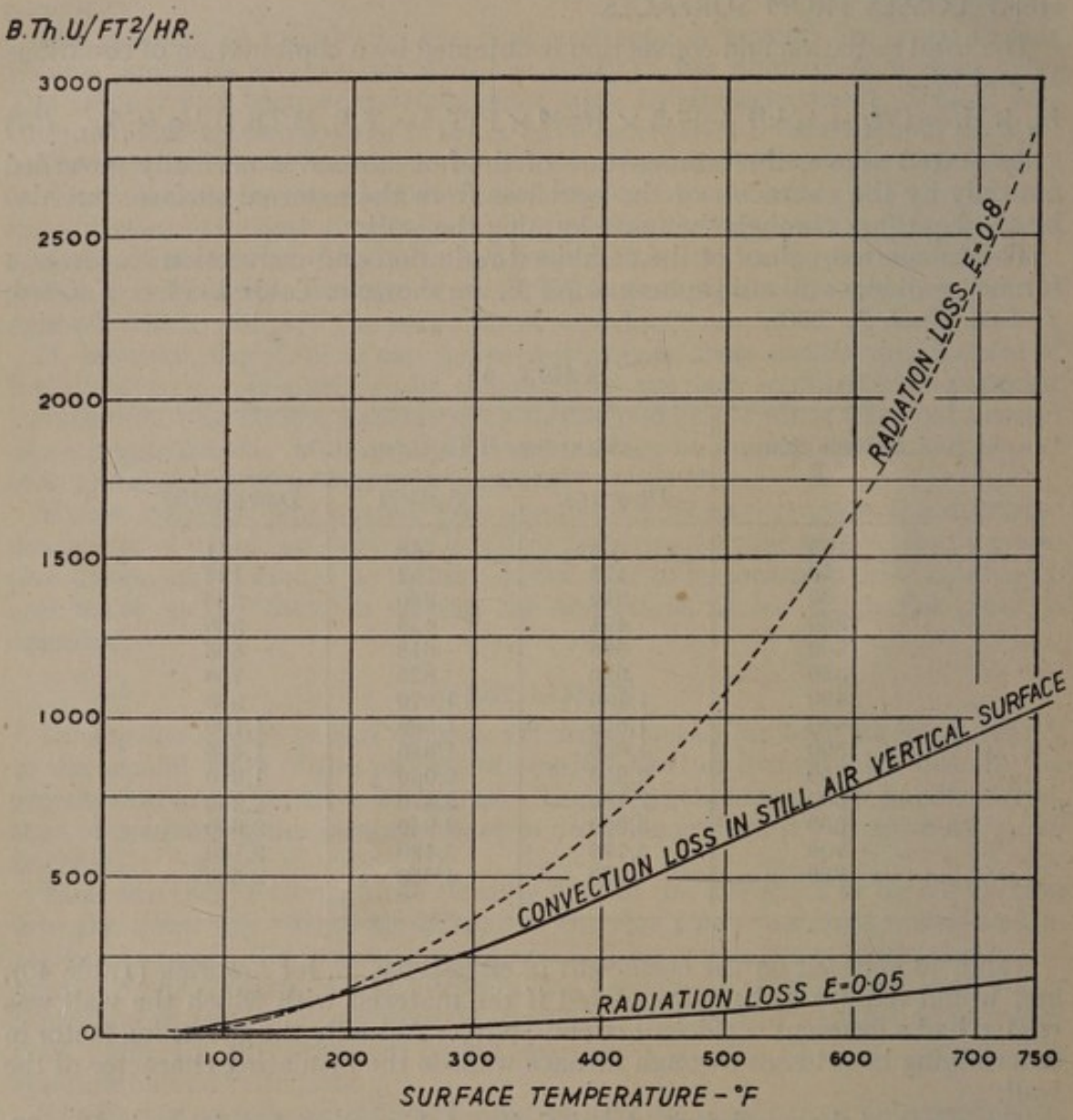

FIG. 50. Curves showing convection and radiation loss from surfaces in still air at 70° F.

TABLE 44. EFFECT OF AIR MOVEMENT ON HEAT LOSS FROM FURNACE.

Temp. of Surface ° F.	B.Th.U./sq.ft./hr.					
	Still air conditions			Air moving at 15 m.p.h.		
	Radiation	Convection	Total	Radiation	Convection	Total
100	30	23	53	30	162	192
200	169	140	309	169	700	869
300	390	287	677	390	1,240	1,630
400	715	450	1,165	715	1,780	2,495

The importance of properly housing furnace structures, hot water tanks, etc., and of ensuring that their outer surfaces are not subjected to draughts will be seen from this example.

HEAT LOSS THROUGH A FURNACE WALL

The temperature of the interior surface of a furnace wall should be known, since it can be measured ; it should also presumably be reasonably constant, since it is a temperature at which the furnace is intended to be operated. To indicate the method of calculating the effect of insulation, a furnace is selected, the internal temperature of which is 2,000° F., and having walls which are composed of 13½ inches of firebrick having a thermal conductivity, k, of 10 B.Th.U./sq.ft./hr./°F./in.

The first step is to make an estimate at the outside furnace temperature ; this estimate may be 600° F. From the application of the conductivity formula,

$$H = \frac{10\ (2{,}000 - t_2)}{13\frac{1}{2}}$$

it is calculated that when t_2 is taken as 600° F., H = 1,035 B.Th.U./sq.ft./hr. From Table 43 (vertical wall), it will be seen that with a surface temperature of 600° F. the surface loss is 2,470 B.Th.U. Evidently the estimate at the value of t_2 was much too high.

If t_2 be now taken as 400° F., H = 1,185, by the same method of calculation as before ; but by interpolating in Table 43 the surface loss is seen to be 1,070 B.Th.U./sq.ft./hr. This estimate is evidently slightly too low.

Proceeding in this way, it can be ascertained by trial and error (or more exactly by graphical methods) that at a temperature of 420° F., H = 1,170, and this virtually coincides with the surface loss from a vertical wall as given in the table.

The surface temperature of the wall is therefore 420° F. and the loss of heat is 1,170 B.Th.U./sq.ft./hr.

If now a 4½-inch covering of insulating brick (k = 1·2) is built outside the 13½-inch firebrick wall, the formula for a composite wall must be used :—

$$h_K = \frac{(t_1 - t_2)}{\frac{L_1}{k_1} + \frac{L_2}{k_2}} = \frac{2{,}000 - t_2}{\frac{13\frac{1}{2}}{10} + \frac{4\frac{1}{2}}{1{\cdot}2}} \text{ B.Th.U./sq.ft./hr.}$$

Proceeding in the same manner as before, it can be calculated that by adding 4½ inches of insulating brick the external surface temperature is reduced to 225° F. and the heat loss to 360 B.Th.U./sq.ft./hr.—another striking commentary on the value of insulation.

Some calculations on these lines are summarised in Table 45.

TABLE 45.

Furnace wall		Furnace wall temperature		
Firebrick inches	Insulating brick inches	Internal ° F.	External ° F.	Heat loss B.Th.U./ sq. ft./hr.
13½	nil	2,000	420	1,170
9	nil	2,000	500	1,680
13½	4½	2,000	225	360
9	4½	2,000	230	380
13½	9	2,000	170	205
9	9	2,000	172	210

These figures indicate (*a*) the relative efficiency of thickening furnace walls to reduce heat loss ; (*b*) the outstanding value of insulating bricks for this purpose ; (*c*) the fact that a moderate thickness of insulating material achieves most of the thermal saving.

In a similar manner the temperature at the interface between the normal and the insulating refractory layer can be calculated, for example, for a composite wall of 9 inches of firebrick and 9 inches of insulating brick, cited in Table 45.

$$\frac{k\,(t_1 - t_2)}{L} = \frac{k_1\,(t_2 - t_3)}{L_1}$$

$$\frac{10\,(2{,}000 - t_2)}{9} = \frac{1{\cdot}2\,(t_2 - 172)}{9}$$

whence $t_2 = 1{,}804$ °F. (984° C.)

A knowledge of the temperature of the interface is required in order to determine whether the maximum safe temperature is reached to which the insulating refractory can be subjected without breakdown. External insulation always increases the average temperature of the inner refractory lining and may in extreme conditions cause this to be raised above its safe working temperature.

Fig. 51 shows a method for the rapid estimation of temperature gradients and heat loss through furnace walls due to General Refractories Ltd. The principle is to express the ratio between wall thickness and conductivity as a quantity x. The hot face temperature is known as being either the temperature at which the furnace is to operate or as being measurable. The surface temperatures, interface temperatures and heat losses can then be read from the chart.

Two examples will make the method clear :—

(1) A furnace wall (hot face temperature 1,600° C.) consists of 18 inches thickness of firebrick, conductivity 10 B.Th.U./sq.ft./hr./°F./in. thickness. It is required to calculate the external surface temperature and the heat loss (cf. " Example 1," Fig. 51).

$$x = \frac{\text{thickness}}{\text{conductivity}} = \frac{T/AB}{k/AB} = \frac{18}{10} = 1{\cdot}8$$

Reading from Fig. 51 at the junction of the horizontal line $x = 1{\cdot}8$ and the curve for 1,600° C.,

cold face temperature = 210° C.
heat loss = 1,250 B.Th.U./sq.ft./hr.

(2) A furnace wall (cf. example 2, Fig. 51) consists of 18 inches firebrick (conductivity 10) 4½ inches insulation (conductivity 1·2) and 4½ inches building brick (conductivity 6) and with a hot face temperature of 1,600°C. It is required to calculate the cold face temperature, the heat loss, and the interfacial temperatures.

$$x = \frac{T/AB}{k/AB} + \frac{T/BC}{k/BC} + \frac{T/CD}{k/CD}$$

$$= \frac{18}{10} + \frac{4{\cdot}5}{1{\cdot}2} + \frac{4{\cdot}5}{6} = 6{\cdot}3$$

Reading from Fig. 51,

cold face temperature = 120° C.
heat loss = 520 B.Th.U./sq.ft./hr.

If A° = temperature at face A
B° = " " B, etc. (Fig. 51, example 2)

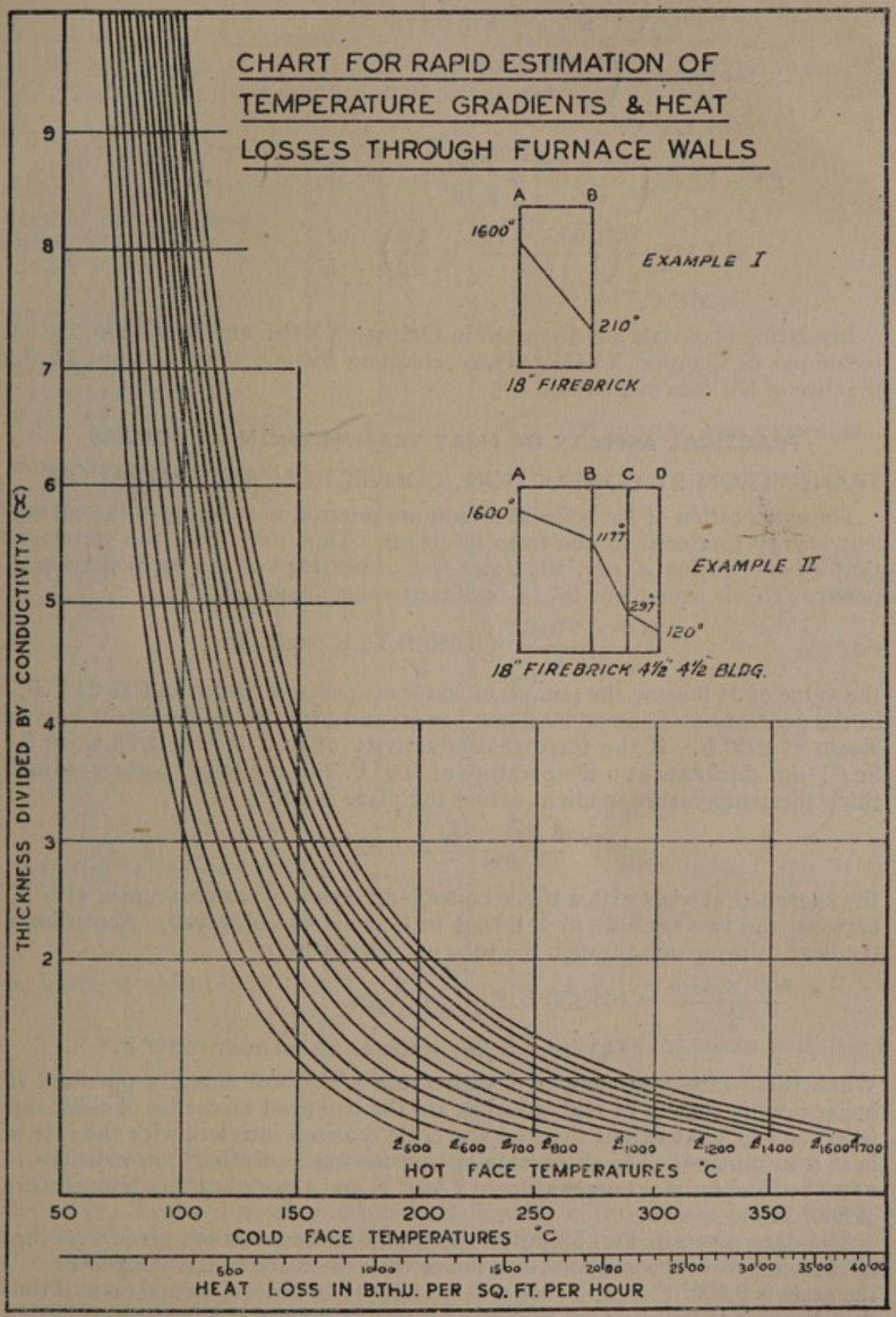

FIG. 51. Chart for rapid estimation of temperature gradients and heat loss through furnace walls.

$$B^\circ = A^\circ - \left(\frac{A^\circ - D^\circ}{x} \times \frac{T/AB}{k/AB}\right)$$

$$= 1{,}600 - \left(\frac{1{,}600 - 120}{6{\cdot}3} \times \frac{18}{10}\right)$$

$$= 1{,}177^\circ \text{ C.}$$

$$C^\circ = B^\circ - \left(\frac{A^\circ - D^\circ}{x} \times \frac{T/BC}{k/BC}\right)$$

$$= 1{,}177 - \left(\frac{1{,}600 - 120}{6{\cdot}3} \times \frac{4{\cdot}5}{1{\cdot}2}\right)$$

$$= 297^\circ \text{ C.}$$

Insulating materials are discussed in Chapter XXII ; and furnace insulation technique in Chapter XVIII. These chapters include some account of the practice of hot face insulation.

PRACTICAL ASPECTS OF HEAT TRANSMISSION IN BOILERS

TRANSMISSION BY CONDUCTION, CONVECTION AND RADIATION

The evaporation of the boiler is sometimes referred to in terms of the equivalent energy produced in the form of steam. Thus may arise the statement that a boiler rating is, say, 10 square feet of heating surface per boiler horse-power. This is equivalent to a rate of heat transmission of

$$\frac{34{\cdot}5 \times 970{\cdot}3}{10} = 3{,}350 \text{ B.Th.U./sq.ft./hr.}$$

the value of 34·5 being the pounds of water evaporated " from and at 212° F." in the generation of one boiler horse-power, and 970·3 being the latent heat of steam at 212° F. If the thermal conductivity of iron is 408 B.Th.U./sq.ft./hr./°F./in. thickness at a temperature of 400° F. and the boiler plate is $\frac{1}{10}$ inch thick the temperature gradient across the plate is only

$$\frac{3{,}350 \times \frac{1}{10}}{408} = 0{\cdot}82^\circ \text{ F.}$$

In experimental trials with a Heine boiler, a difference of temperature of 41·5° F. between the two surfaces of a tube $\frac{1}{8}$ inch thick was observed. Accordingly the heat transmission through the tube must have been

$$\frac{408 \times 41{\cdot}5}{\frac{1}{8}} = 135{,}450 \text{ B.Th.U./sq.ft./hr.}$$

$$= 140 \text{ lb./sq.ft./hr. of steam " from and at 212° F."}$$

When these rates are compared with those which are actually operated in boiler practice it will be realised what are the potential capacities of clean surfaces, and to what extent the soot and scale coatings interfere with the rate of heat transmission. The above rate of transfer was equivalent to conditions of a fuel bed and furnace temperature of 2,640° F. and a sooted surface temperature of 800° F.

The data given in Fig. 52, after the Cochrane Corporation, gives a picture of the temperature conditions in a boiler tube when the initial temperature of the gases is 2,500° F. and the boiler is working at 10 times a normal rate. From this diagram it will be seen that the process of getting the heat by radiation and convection through to the dry surface of the boiler tube is slow compared with that on the wet side. The course of the temperature conditions through the gas passages of a boiler is shown in Fig. 53, based on experimental observa-

tions on water tube boilers. Heat is transmitted from the hot fuel bed and adjoining refractory surfaces to boiler heating surfaces by direct radiation. from the products of combustion by gas radiation, and by convection. The conductance by convection varies between 1 and 12 B.Th.U./sq.ft./hr./°F, difference for the gases flowing according to conditions of flow. The pheno-

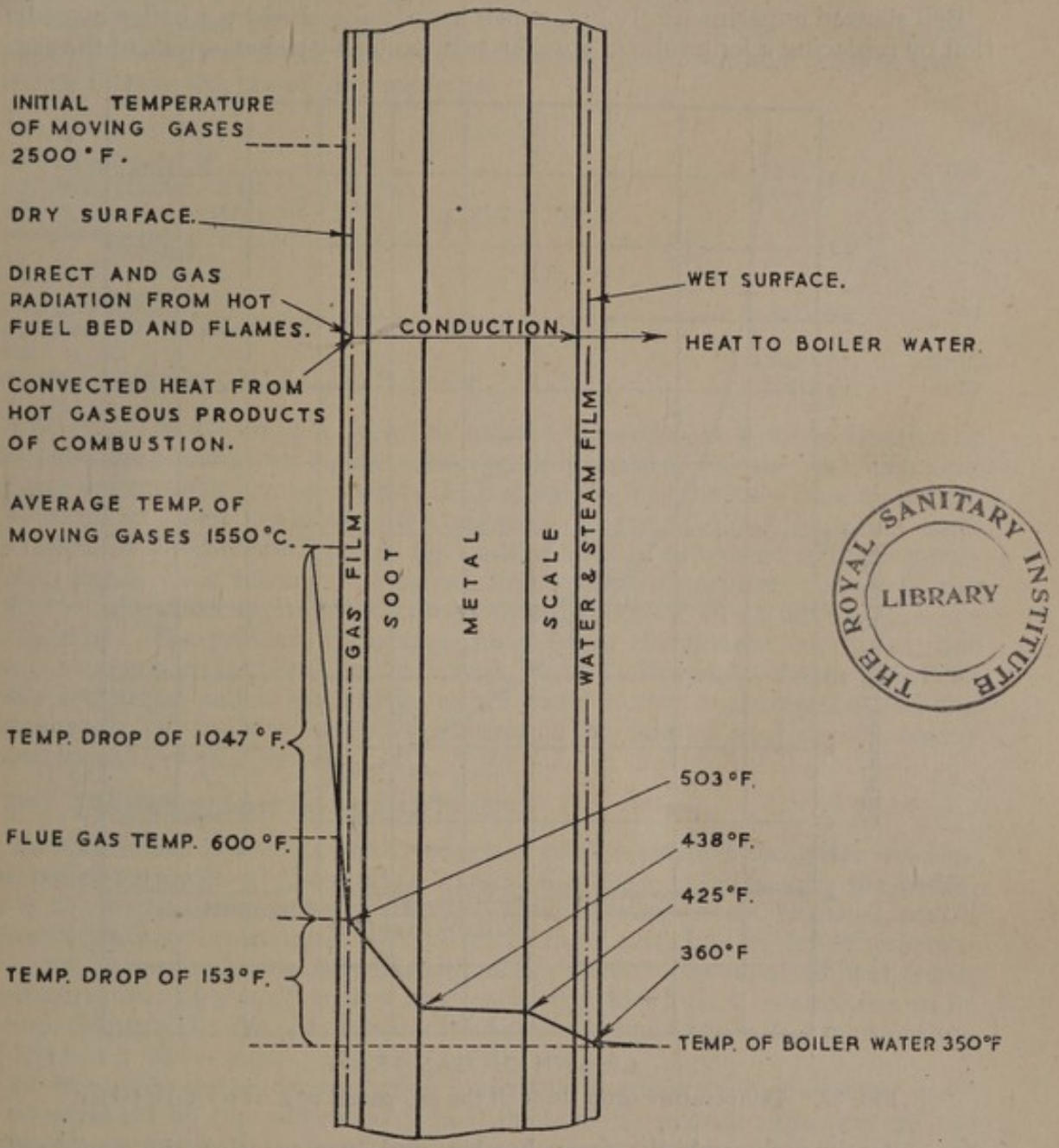

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FIG. 52. Heat transmission from hot flue gases to boiler water.

menon is illustrated most suitably by reference to waste heat boilers fitted with fire-tubes. These are usually equipped with high-power exhaust fans to give the high velocities associated with conditions of efficient heat transfer by convection.

The furnace of the locomotive boiler presents an adequate illustration of heat transfer by radiation. The hot surface is the fuel bed, and the cold surface the combined surface of the tube sheet, the crown sheet, the front sheet and the plates on both sides of the fuel bed. Increasing the furnace temperature from

2,000° F. to 3,000° F. will nearly quadruple the amount of heat imparted by radiation. A drop of temperature of 100° from 2,500° to 2,400° which readily arises from unskilful firing will reduce the amount of heat transferred by radiation by 12 per cent. This indicates the importance of maintaining the boiler furnace fires in a state of the brightest incandescence for efficient utilisation of the available heat of the fuel.

Bell showed experimentally how much heat is absorbed by a boiler by radiation by replacing a long tube in a water-tube boiler by a shorter one of the same

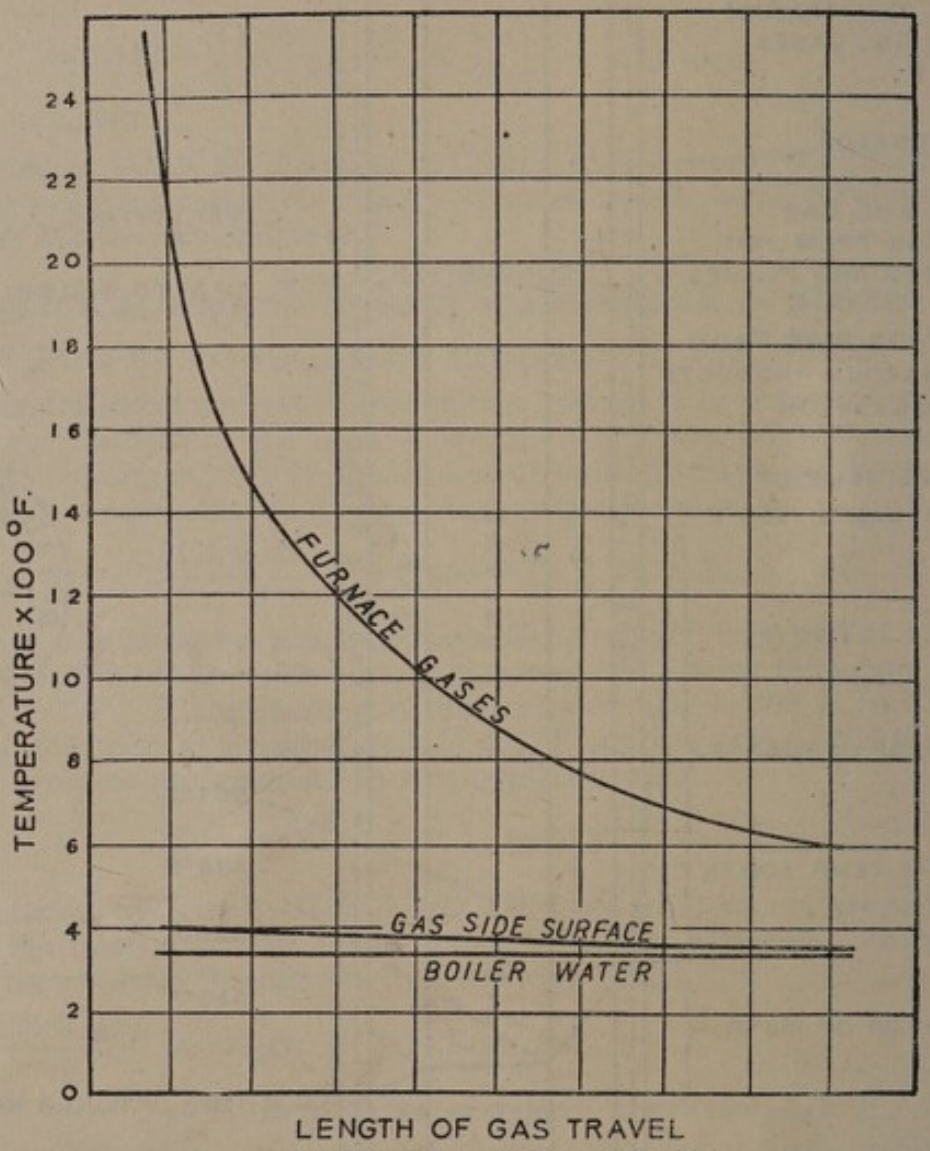

FIG. 53. Temperature drop through the gas passes of a water-tube boiler.

size extending through the front header, and just entering the first baffle. The heat absorption was determined from the temperature and mass of the water which was circulated independently through this separate tube. The heat absorbed was equivalent to a rate of evaporation of 75 lb. of steam per square foot per hour. Only a portion of the surface was heated by radiation, and it was estimated that this was raised to about 100 lb. per square foot per hour from the impingement of the gases (convection). As the gases were at about 2,500° F. the results were of an order which checked well with the fourth power radiation law. The water-cooled walls of large stoker-fired boilers make use of the direct radiation from the flame.

Broido, quoted by the Cochrane Corporation, has given the distribution of

heat transmitted from the gases flowing through a boiler at the rate of 6,000 lb. per hour per square foot of area, at two points where the gas temperatures were 1,400° F. and 800° F., respectively, steam pressure 250 lb. per square inch gauge, water and tube temperature about 406° F. The amount of heat transmitted by convection depends greatly on the velocity of the gas at the point under consideration. Broido's results are given in Table 46.

Where the gases fall in temperature in the gas circuit convection transfer may be predominant if the velocity is sufficiently great, but high rates of transmission require the use of high velocities.

TABLE 46

Gas temperature, ° F.		1,400		800
Total heat transfer, B.Th.U./sq. ft./hr.		7,660		2,620
Total conductance coefficient—B.Th.U./sq. ft./hr./° F.		7·7		6·65
Gas radiation—CO_2—B.Th.U./sq. ft./hr.	1,340		280	
,, ,, H_2O— ,, ,,	477		107	
Total gas radiation		1,817		387
Heat transfer by convection (by difference) B.Th.U./sq. ft./hr.		5,843		2,233
Convection coefficient, B.Th.U./sq. ft./hr./° F.		5·83		5·66

One of the most complex problems of heat transmission is the evaluation of the performance of a combustion chamber of a furnace or boiler, in which heat is being transmitted simultaneously in all of the ways which have been discussed. Allowance must be made for the combined actions of direct radiation from the flame to the medium heated, the boiler plates or tubes and refractory surfaces, thence back through the flame, with partial absorption therein to the colder evaporation surfaces. Convection and external losses have also to be considered. The problem is beyond the scope of the present treatment, but the considerations involved are dealt with by H. C. Hottel.* By the use of the same technique and a knowledge of the rate of fuel consumption and the evaporation of the boiler the distribution of the flow of heat can be traced throughout the boiler system.

HEAT TRANSMISSION IN ECONOMISERS

An economiser is a water heater which is used to preheat the water entering the boiler by means of the sensible heat in the waste gases leaving the boiler. It is in effect a heat exchanger in which evaporation is not required, and a form of heating surface must be substituted more suited to the lower temperature conditions involving only heating of the water. The mode of heat transmission is that of a counter-flow heat exchanger in which a balance has to be struck between the rise in temperature of the water and the fall in temperature of the boiler outlet gases.

In an economiser of conventional design the temperature of the outlet gases is so adjusted by the allowance of heat exchanging surface as to prevent too close an approach to steam formation. In recent years steaming types of economisers have come into use and this limitation then no longer holds.

It has been shown in the discussion of heat transfer by convection, which is the predominant means of heat transfer from gases at low temperatures, that increase of gas velocity is necessary to raise rates of heat transfer. In the older installations where chimney draught was limited velocities had to be kept low in order to limit loss of draught due to the resistance of the tubes, and a large heating surface or a large number of tubes was required for a given duty. The efficiency of the heat transfer through the surface of the tubes was preserved by the fact that they were automatically scraped clean of soot.

* W. A. McAdams, " Heat Transmission." McGraw-Hill (1942), Chapter 3.

The possible effect of the better heat transfer due to increased velocity naturally comes into question. This can be effected by modifying the design as, for example, in the vertical type of apparatus, and by the use of baffles. A heat transfer coefficient of 3 B.Th.U./sq. ft./hr./°F. thus becomes raised to values of the order of 5·5 to 7·7. Higher velocities tend to keep the tubes cleaner, and the limit in the development of velocity of the gases depends upon the available draught and the additional power to be provided where forced draught is employed.

DRAUGHT LOSS IN RELATION TO ECONOMISER EFFICIENCY

The relation has been shown by G. E. Tansley * in terms of temperature rise to be as follows :—

If W = weight of gases, lb. per hour
w = evaporation, lb. per hour
t = temperature of gases at fan, °F.

Then, volume of the gases weighing 0·076 lb./cu. ft. at 60° F.

$$= \frac{1}{0{\cdot}076} \times \frac{t + 460}{520}$$

$$= 0{\cdot}0252\ (t + 460) \text{ cu. ft./lb.}$$

$$= \frac{W}{60}.0{\cdot}0252\ (t + 460) \text{ cu. ft./minute.}$$

Fan horse-power per 1 inch w.g. draught

$$= \frac{W}{60} \cdot \frac{0{\cdot}0252\ (t + 460) \times 5{\cdot}2}{33{,}000}$$

For a motor efficiency of 90 per cent., and a fan efficiency of 50 per cent. the power input to the motor

$$= \frac{W}{60} \cdot \frac{0{\cdot}0252\ (t + 460) \times 0{\cdot}746 \times 5{\cdot}2}{60 \times 33{,}000 \times 0{\cdot}9 \times 0{\cdot}5} \text{ kWh}$$

If the heat in the steam required per kilowatt-hour is taken as 20,000 B.Th.U., the heat equivalent of the motor input

$$= \frac{W}{60} \cdot \frac{0{\cdot}0252\ (t + 460) \times 0{\cdot}746 \times 5{\cdot}2 \times 20{,}000}{60 \times 33{,}000 \times 0{\cdot}9 \times 0{\cdot}5} \text{ B.Th.U./1 in. w.g.}$$

and the rise in the feed water temperature equivalent to 1 inch water gauge

$$R = \frac{W\ (t + 460) \times 0{\cdot}0252 \times 0{\cdot}746 \times 5{\cdot}2 \times 20{,}000}{w \times 60 \times 33{,}000 \times 0{\cdot}9 \times 0{\cdot}5}$$

$$= \frac{W}{w} \cdot \frac{(t + 460)}{455}$$

Actually the heat per kilowatt-hour may vary widely according to the efficiency of power generation. Further, it is not correct to say that a higher draught loss necessarily means a greater rate of heat transfer unless it is clearly understood that the form, type and arrangement of the heating surface remain constant. Further, the question of fan power involves economic considerations outside the heat transfer problem.

A related question may also be examined, viz., does the staggering of the tubes of an economiser confer any appreciable benefit ? Here the generalisation that staggered tubes are better than non-staggered breaks down. The pitching of the tubes in an economiser, say $4\frac{9}{16}$ inches diameter on 8-inch centres is so

* *Trans. Inst. Mining Engineers*, LXXIX, Pt. 2, pp. 190–208.

wide and the gas velocities in general so low that the increase in both heat transfer and draught loss is quite negligible. The point has been subjected to lengthy examination, and the modern method for speeding up rate of gas travel in certain types of economisers is to arrange the tubes in groups with vertical baffles between, so that instead of being across them as in the normal arrangement, the flow of gases is approximately parallel to the length of the tube. Thus in an economiser of say 192 tubes, 8's wide, 9 feet long, the area for cross gas flow is 33·5 square feet. If this economiser is arranged in three groups of 64 tubes each, with vertical baffles between, the gas flow area becomes 23·18 square feet and with the same total gas quantity the gas speed is proportionately greater.

Both heat transfer rate and draught loss increase, but the latter is not a direct function of the former in this example, since part of the increased loss in draught is due to change of direction over and under the baffles. A typical set of performance figures will illustrate this point (Table 47).

TABLE 47. PERFORMANCE FIGURES FOR AN ECONOMISER

Economiser of 192—8's 9 ft.
Heating surface 2,234 sq. ft.

	Normal arrangement	In three groups with vertical baffles
Weight of water, lb./hour	20,000	20,000
Weight of gases, lb./hour	45,000	45,000
Gas flow area—sq. ft.	35·5	23·18
Gas velocity in lb./hour/sq. ft. of flow area	1·345	1·945
Rate of heat transfer in B.Th.U./sq. ft./hr./° F. ..	3·5	3·8
Inlet gas temperature—° F.	650	650
Outlet gas temperature—° F.	407	394
Inlet water temperature—° F.	120	120
Outlet water temperature—° F.	244	251
Total draught loss due to friction and change of direction, in. w.g.	0·07	0·24
Draught loss due to change of direction, in. w.g. ..	—	0·08

For this particular set of conditions, therefore, there was shown a gain in performance represented by 7° F. rise in water temperature, as against an extra 0·17 inch w.g.

How far this gain in performance is justified depends again upon circumstances involving a wide range of considerations, operational and economic. If, for example, induced draught enables an inferior and formerly unusable fuel to be used, some proportion of the fan charges should logically be added to the fuel cost, and not debited entirely to the economiser saving.

The wider aspects of these problems are dealt with in the later chapters on boilers and their auxiliaries.

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CHAPTER IX

THE FLOW OF FLUIDS

The mechanics of fluid flow—Bernouilli's theorem—Streamline and turbulent flow—Viscosity of fluids—Equations of fluid flow—Enlargement and contraction losses—Effect of bends, etc.—Special formulae used for gas transmission—Principles of the measurement of fluid flow—The flow of gases in furnaces—Examples of fluid flow calculation.

FLUID flow is a highly important branch of fuel technology. Liquids, vapours (including steam) and gases have to be transported in pipes, an operation involving the expenditure of power. It is necessary to know the principles of fluid flow in order to design the plant. Gases are caused to flow through furnaces comprising mains and chambers of many sizes and shapes. Knowledge of the pressure variations in these systems is important in order that the furnace shall be so designed that a proper flow of gases is maintained ; on the correct flow of the gases depends the heating of the furnace and the stock contained in it. Draught (see Chapter VI) is a special case of fluid flow. Convection heating is effected by fluid flow. The measurement of fluid flow is essential to the control of industrial operations. In this chapter, methods are given for the calculation of fluid flow with as simple an explanation as possible of the mathematical basis of these methods.

LIST OF SYMBOLS USED IN THIS CHAPTER

		Pronounciation of Greek letter.
Absolute viscosity—c.g.s. units (poises), f.p.s. units (no special name)	η	eta
Kinematic viscosity—c.g.s. units (stokes), f.p.s. units (no special name)	ν	nu
Density of fluid—lb. per cubic foot ..	ρ	rho
Gravitational constant—32·2 ft./sec./sec.	g	
Static pressure—feet head of fluid ..	x, h	
Velocity of flow—feet per second.. ..	v	
Coefficient of friction	f, ζ	zeta
Frictional force per unit area of wetted surface	F	
Mechanical work done—foot-pounds ..	w	
Work done by expansion of a fluid—foot-pounds	X	
Pressure	p (lb./sq. ft.) P (inches w.g.)	
Volume	V, Q	
Diameter of pipe or conduit	D (feet) d (inches)	
Reynolds number	Re	
Length of pipe or conduit—feet	L	
Hydraulic mean depth (= sectional area ÷ wetted perimeter	M	
Sp. gr. (air = 1)	s	
Coefficient of flow	α	alpha

N.T.P., meaning normal temperature and pressure, refers to a gas at 32° F. and 29·92 inches bar., i.e. 0° C. and 760 mm. bar.

S.T.P., meaning standard temperature and pressure, refers to a gas at 60° F. and 30 inches bar.

FLUID FLOW

THE MECHANICS OF FLOW

As in other branches of mechanics, the law of conservation of energy requires that the total energy in the system must remain constant.

The kinetics of fluid flow are best illustrated by reference to the fundamental basis upon which they have been worked out, namely Bernouilli's theorem. This may be illustrated in its simplest form by a liquid flowing through a pipe. The total energy possessed by the liquid at any point depends essentially on the algebraic sum of four factors, namely :—

Potential energy due to position + potential energy due to pressure + kinetic energy due to motion — energy lost as friction.

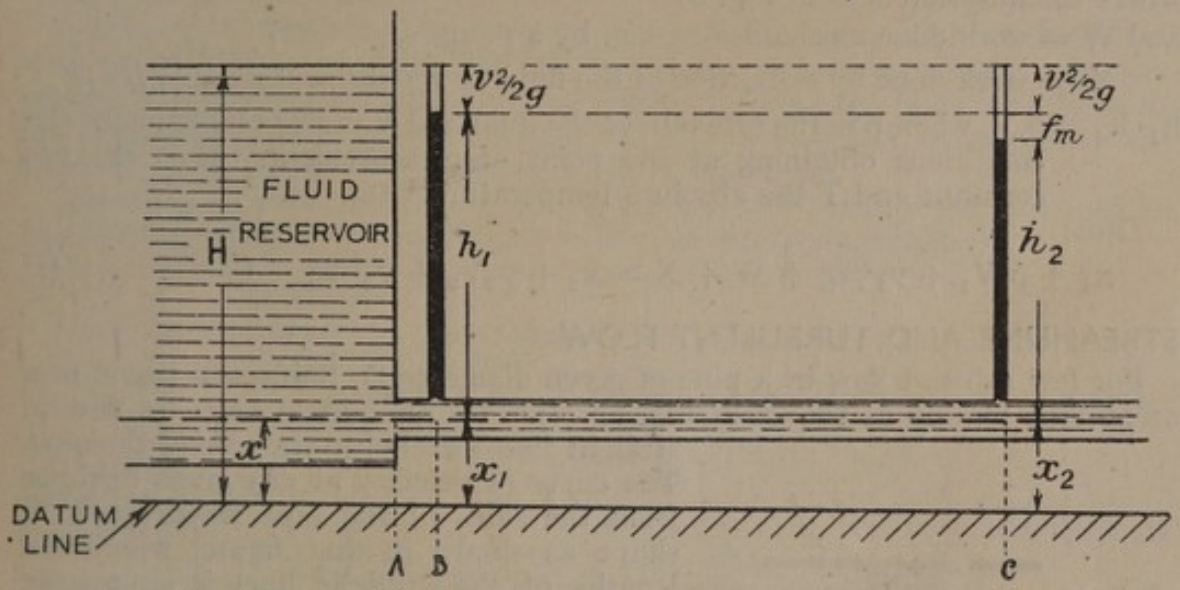

FIG. 54. Diagram illustrating Bernoulli's theorem of fluid flow.

If x is the height in feet above an arbitrary datum line (Fig. 54),
h is the static pressure (measured in terms of a pressure gauge) expressed as feet head of the liquid under consideration.
v is the velocity of flow of the liquid in feet per second.
f_m is a measure of the friction losses.
m is the mass of the liquid in lb./cu. ft.
g is the gravitational constant = 32·2 ft./sec/sec.

Then by elementary mechanics :—

$$\text{mechanical energy} = mgx + mgh + \tfrac{1}{2}mv^2 - mgf_m$$

$$= mg\left(x + h + \frac{v^2}{2g} - f_m\right)$$

and since the energy of the fluid in general terms = mgH
where H is the effective head,

$$H = x + h + \frac{v^2}{2g} - f_m.$$

Fig. 54 indicates how the energy is distributed in practice. Here H is the total head available ; part of this at any given height is due to the vertical distance from the (arbitrary) datum line. If the pipe is horizontal this distance x remains constant. As the liquid flows through the pipe, neglecting friction in the short length AB, part of the remaining energy H—x, is converted into the kinetic energy of motion and part remains as the static pressure h_1. Over the length BC if the pipe is of uniform bore, the velocity remains constant and so,

therefore, does the value of the velocity head $v^2/2g$. Frictional losses, however, reduce the static pressure, so that a pressure gauge indicates a drop in pressure between B and C of $h_1 - h_2$ (i.e. of f_m).

Bernouilli's theorem for gases is a little more complicated since the energy derived from compression or expansion of the gases may have to be taken into account. In a complete system, moreover, any work done on the fluid by a pump, etc., must be considered. The complete Bernouilli theorem for all fluids is given below. It is based, as stated above, on the principle that the total energy of the fluid at any one section of the system must equal its total energy at any other section. Thus :—

$$x_1 + h_1 + v_1^2/2g + W + X = x_2 + h_2 + v_2^2/2g - f_m$$

where the notation is as in Fig. 54

and W = work done mechanically, e.g. by a pump

X = work done by expansion of the fluid $= \int p\,dv = RT \log (V_2/V_1)$

h_1, h_2 = p.V, when p is the pressure of the fluid and V is its volume under the conditions obtaining at the point of measurement. R is the gas constant and T the absolute temperature.

Thus,

$$x_1 + p_1V_1 + v_1^2/2g + W + X = x_2 + p_2V_2 + v_2^2/2g - f_m \quad . \quad . \quad (1)$$

STREAMLINE AND TURBULENT FLOW

For low rates of flow in a pipe of given diameter D, fluids will travel in a streamline motion as shown in Fig. 55 (*a*). The individual particles flow in straight lines parallel to the axis of the pipe. The curve of velocity at any given distance from the sides of the pipe is parabolic in shape as shown in that figure, where the lengths of the straight lines is a measure of the relative velocities.

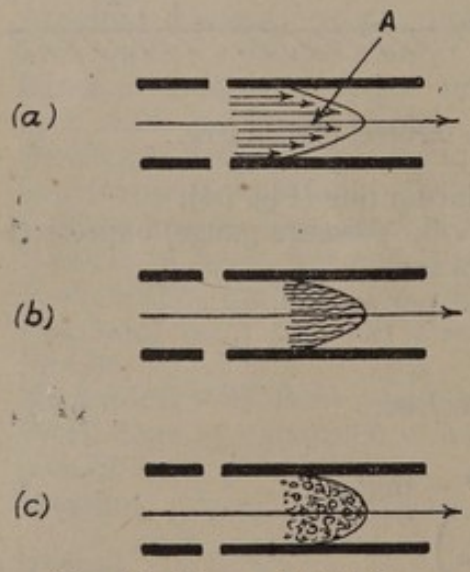

FIG. 55. Types of fluid flow.

In the same pipe when the velocity is further increased the straight-line flow becomes disturbed and a "wavy" or "transitional" type of flow is set up (Fig. 55 (*b*)).

A small additional increase in velocity causes the flow to become turbulent (Fig. 55 (*c*)). Here the straight-line flow is broken up by eddies and vortices. The velocity-distribution curve is different; it rises more sharply against the wall and is flatter in the central portion of the pipe.

The transitional velocity from one type of flow to another was investigated by Reynolds, and was found to depend on the viscosity, the diameter of the pipe and the velocity of flow of the gas. Viscosity is discussed in this chapter.

The relationship was expressed by the "Reynolds number," Re, where

$$\mathrm{Re} = \frac{vD\rho}{\eta} = \frac{vD}{\nu}$$

where v = velocity of gas
D = diameter of pipe
ρ = density of fluid
η = absolute viscosity
ν = kinematic viscosity

} These terms are defined on pp. 170 and 171.

Thus in c.g.s. units,

$$\mathrm{Re} = \frac{\text{v(centimetres per second)D(centimetres)}\rho\text{(grams per cubic centimetre)}}{\eta \text{ (poises)}}$$

and in f.p.s. units,

$$\mathrm{Re} = \frac{\text{v(feet per second)D(feet)}\rho\text{(lb. per cubic foot)}}{\eta \text{ (f.p.s. units)}}$$

Provided a consistent set of units is used, i.e. all in the c.g.s. system or in the f.p.s. system, the same value for Re is obtained by these formulæ. When Re is below about 2,000, the flow is streamline ; between 2,000 and 3,000 it is transitional and above about 3,000, turbulent. In most industrial practice, when handling gases and liquids of low viscosity, flow is turbulent.

Although the foregoing discussion indicates the broad outlines of fluid flow, it is worth while examining the mechanism of flow more closely. At any average velocity of flow in a pipe, even when high enough to be turbulent, at the immediate surface of the pipe the velocity is very low, consequently the value of Re will be low and the flow is here streamline.

Likewise, a short distance farther from the pipe surface the flow will be transitional, whilst in the main body of fluid it will be turbulent.

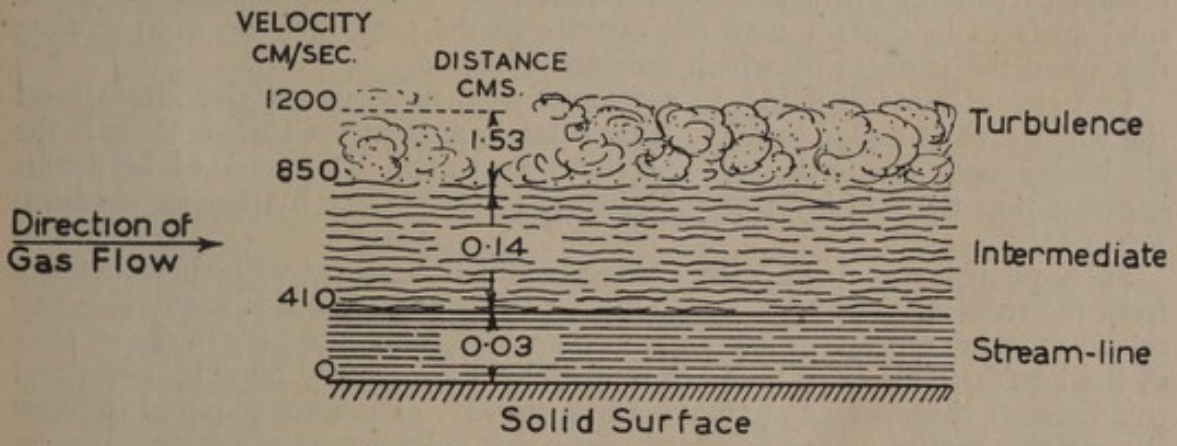

FIG. 56. Fluid flow near surfaces.

Thus experiments (Zijnen, Thesis, Delft, 1924) on air flowing parallel to a horizontal plate of glass gave the results shown in Fig. 56.

EFFECT OF MECHANISM OF FLOW ON CONVECTION HEATING

The mechanism of flow influences the rate of heating by convection. In convection heating, it is necessary for a hot particle to come in contact with a cooler particle in order to share its heat energy with it. The cooler particle may be another gas or liquid particle, or it may be part of the solid heating wall, etc. Thus, if a heating fluid is circulating in a pipe, the objective must be to cause the heating fluid to give up its heat to the pipe through which the heat will be conducted.

A molecule of hot gas at the point A (Fig. 55 (*a*)) has little opportunity of coming in contact with the pipe. It is travelling in a streamline path and can thus only reach the side by a slow process of diffusion. It is likely that the heating medium will have left the apparatus before this happens. Consequently, the only means left for heat transfer from the particle at A is by thermal conduction. The thermal conductivity of gases and most liquids is very low, so that with streamline flow heat transmission must be slow.

A similarly situated particle in a fluid flowing turbulently will quickly be mixed with the remainder of the fluid which will thus tend to be at a uniform

temperature. Against the side of the pipe, however, there still persist the transitional and streamline layers, through which the rate of flow will be slow. These layers are thin, and consequently there is more opportunity for diffusion in them ; they still have low thermal conductivity, but their small thickness enables heat to be transmitted more quickly. Thus, gaseous or fluid heating must always be slowed up by the presence of the streamline layer against the walls, but the thinner this layer, the more rapid the heating.

In the experiments of Zijnen previously mentioned, it was found that by increasing the velocity of the main stream of air from 13·1 to 78·7 feet per second, the thickness of the streamline layer was decreased from 0·087 to 0·018 centimetres. This indicates why higher velocities of flow give a greater rate of convection heating.

VISCOSITY

A liquid which flows with difficulty is generally said to be viscous, and the property which determines the internal resistance to flow is its viscosity.

Resistance to fluid flow is caused by friction. Friction in solids is a force exerted between surfaces in contact. Fluids in motion are in contact with the surfaces of the pipes or ducts in which they move, but the layer of fluid in immediate contact with a surface is stationary so that the force acting between solid surfaces in contact does not operate in the same way between moving fluids and the surfaces by which they are bounded.

Particles of moving fluids do not move with uniform velocity throughout the whole area of a pipe or duct. Immediately against the surface there is the stationary layer, perhaps only one molecule thick. The work of Langmuir suggests that the first layer may be bound to the surface by physico-chemical forces.

Subsequent layers move with a velocity which increases with their distance from the surface. The work which must be expended in order to produce this relative motion represents frictional forces which come into play in the interior of the liquid. This force tends to retard the more rapidly moving layers and to accelerate those which are moving less quickly. The force required to cause the relative motion under standard conditions is known as viscosity.

The numerical value of viscosity at a given temperature is a specific physical property of a fluid, so that different fluids have different viscosities.

The magnitude of the viscosity in any fluid may be defined as (*a*) the force expressed in dynes acting on an area of 1 square centimetre necessary to produce a difference of velocity of 1 centimetre per second between a layer of fluid and a similar layer 1 centimetre distant. Thus in Fig. 57, if the layer of

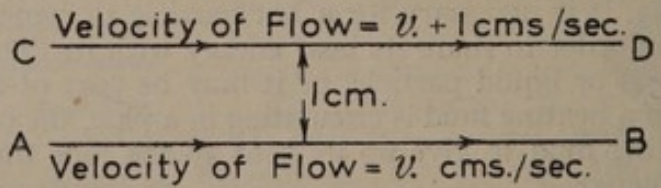

FIG. 57. Illustration of the principle of viscosity.

fluid AB moves with a velocity v centimetres per second, and the layer CD moves with a velocity of v + 1 centimetres per second, the force in dynes per square centimetre acting on the layer CD just sufficient to cause this difference of motion is known as the coefficient of viscosity in c.g.s. units. This unit is known as the " poise."

(*b*) In f.p.s. units, the coefficient of viscosity is the force in poundals, which acting on an area of 1 square foot will produce a differential velocity of 1 foot

per second in two layers of fluid 1 foot apart. There is no name for the f.p.s. unit. Numerically,

Viscosity in poises × 0·0672 = viscosity in f.p.s. units.
Viscosity in f.p.s. units × 14·88 = viscosity in poises.

These units are known as " absolute " viscosity, and usually designated by the Greek letter, η.

In practical work it is often convenient to use another value known as " kinematic " viscosity commonly denominated by the Greek letter, ν. In c.g.s. units this is known as the " stokes."

$$\text{Kinematic viscosity} = \frac{\text{absolute viscosity}}{\text{density}}$$

$$\text{which, in c.g.s. units} = \frac{\text{poises}}{\text{grams per cubic centimetre}} \text{ stokes}$$

$$\text{and, in f.p.s. units} = \frac{\text{f.p.s. units of viscosity}}{\text{lb. per cubic foot}} \text{ ft.-lb.-sec. units}$$

kinematic viscosity in stokes × 0·0010764 = f.p.s. units
kinematic viscosity in f.p.s. units × 929·03 = stokes.

It is commonly appreciated that viscous liquids become thinner when heated. They flow more readily in this condition. This means in practice that the force required to overcome the internal friction becomes less. For liquids, therefore, viscosity becomes less as the temperature rises. In gases, however, the opposite occurs, and the viscosity becomes greater as the temperature rises. This fact is not always appreciated. Its practical significance is that the power required to move gases, e.g. in fans, becomes greater at higher temperatures. Thus in supplying preheated gases to burners not only does the greater frictional resistance produced by the expansion of the heated gases require to be overcome, but also the effect of the increased viscosity.

Viscosity changes rapidly with temperature. Thus kinematic viscosity must always be calculated with reference to the viscosity and density of the fluid at the temperature concerned. It is of great importance in all problems connected with the flow of fluids. At pressures reasonably near atmospheric, even up to, say, 100 lb. per square inch, viscosity may be regarded as independent of the pressure.

No general law can be given regarding the effect of temperature on the viscosity of liquids. Experimentally determined values must be taken. In Table 48 are given some values for water (Bingham and Jackson, *Bulletin, Bureau of Standards*, 1917, **14**, 75) ; intermediate values can be obtained by interpolation. Values for certain other liquids are to be found in Table 49.

TABLE 48. VISCOSITY OF WATER

Temperature		Absolute viscosity, η		Kinematic viscosity, ν	
°C.	°F.	Poises	F.P.S. units	Stokes	F.P.S. units
0	32	0·01792	0·001204	0·01792	·00001930
10	50	0·01308	0878	0·01308	1407
15	59	0·01140	0766	0·01141	1228
20	68	0·01005	0675	0·01007	1084
30	86	0·00801	0538	0·00804	0866
50	123	0·00549	0369	0·00556	0599
70	158	0·00406	0273	0·00415	0447
100	212	0·00284	0191	0·00296	0319

TABLE 49. KINEMATIC VISCOSITIES—STOKES

Temperature		Petrol No. 1	Power alcohol	Motor benzole	Diesel oil	Heavy fuel oil
° C.	° F.					
0	32	0·0080	—	—	0·108	204
10	50	0·0074	0·015	0·0083	0·070	50
20	68	0·0068	0·012	0·0072	0·049	12
30	86	0·0062	0·010	0·0066	0·040	5

The viscosity of fuel gases is given in Tables 50 and 51.
The kinematic viscosity at any given temperature, t° is

$$\frac{\text{absolute viscosity at t°}}{\text{density at t°}} \text{ i.e. } \frac{\eta_t}{\rho_t}$$

Both the viscosity and density must be expressed throughout in the same units system, e.g. in c.g.s. units or in f.p.s. units.

TABLE 50

ABSOLUTE VISCOSITY OF INDUSTRIAL FUEL GASES IN C.G.S. UNITS (POISES)

Calculated from the modified Sutherland formula,

$$\eta_t = \frac{\eta_o}{1 + t/C^1} \times \left(\frac{273 + t}{273}\right)^{\frac{3}{2}}$$

when η_t and η_o are viscosities at t° C. and 0° C. respectively.

Type of gas	Blast furnace gas	Coke oven gas debenzolised	Continuous vertical retort gas (steaming)	Mechanical producer gas		Blue water gas	Steam	Air
				Coal	Coke			
Density at 0° C. and 760 mm.	0·0013	0·00049	0·00062	0·00113	0·00116	0·000712	0·000804	0·001293
Temp.	10^6 × absolute viscosity in C.G.S. units							
0° C.	160	114	132	163	166	153	88	170
20	170	121	140	172	176	162	95	180
40	179	128	148	181	185	171	103	190
80	196	141	162	198	203	187	118	209
120	214	153	177	215	220	203	135	225
160	230	166	191	231	236	217	152	242
200	245	177	204	248	252	231	167	258
250	263	190	219	263	270	248	186	276
300	280	203	233	280	287	264	203	295
400	312	227	261	312	319	293	238	329
500	342	249	285	341	349	321	272	359
C^1	405	423	416	393	394	392	950	397

Densities in gm./c.c. at 0°C. and 760 mm.
To obtain values for density in lb./cu. ft. at 32° F. and 29·92 in. bar., multiply by 62·428.
To obtain value for viscosity in f.p.s. units, multiply by 0·0672.

TABLE 51. KINEMATIC VISCOSITY OF INDUSTRIAL FUEL GASES IN C.G.S. UNITS (STOKES) AT ATMOSPHERIC PRESSURE

Derived from values of absolute viscosity calculated according to the modified Sutherland formula by dividing by the density at the corresponding temperature.

Type of gas / Temp.	Blast furnace gas	Coke oven gas debenzolised	Continuous vertical retort gas (steaming)	Mechanical producer gas		Blue water gas	Steam	Air
				Coal	Coke			
0° C.	0·123	0·233	0·213	0·144	0·143	0·217	0·109	0·131
20	0·141	0·264	0·242	0·164	0·162	0·246	0·128	0·152
40	0·159	0·298	0·274	0·184	0·182	0·276	0·147	0·172
80	0·197	0·371	0·338	0·228	0·226	0·342	0·191	0·212
120	0·238	0·449	0·411	0·275	0·272	0·412	0·240	0·254
160	0·282	0·535	0·489	0·325	0·322	0·488	0·295	0·300
200	0·328	0·622	0·570	0·378	0·374	0·567	0·356	0·346
250	0·390	0·740	0·676	0·448	0·444	0·672	0·436	0·413
300	0·455	0·866	0·789	0·523	0·517	0·783	0·528	0·480
400	0·595	1·136	1·037	0·681	0·676	1·023	0·730	0·628
500	0·749	1·437	1·300	0·857	0·850	1·280	0·963	0·787

To obtain value for viscosity in f.p.s. units multiply by 0·0010764.

COEFFICIENT OF FRICTION

The viscosity of a fluid is a measure of its internal friction. Viscosity is not, however, the only cause of pressure loss. The eddies set up in turbulent flow are also responsible for frictional losses, and it is not surprising to find that there is a relationship between the coefficient of friction of fluid flow and the Reynolds number.

In order to determine the flow of a fluid it is thus necessary to calculate the Reynolds number for the conditions and from that to ascertain the coefficient of friction from experimentally determined data. The principles involved will now be discussed.

EQUATIONS OF FLUID FLOW

The fundamental equation (derived from Bernouilli's theorem equation (1)) expressing the pressure drop along a horizontal pipe in which a fluid is flowing is :—

$$p = 8(F/\rho v^2) \cdot \frac{L}{D} \cdot \rho \frac{v^2}{2g} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

where p = pressure drop in lb./sq. ft.
F = frictional force per unit area of wetted pipe surface
ρ = density of fluid, lb./cu. ft.
v = mean linear velocity of flow, ft./sec.
L = length of pipe in feet
D = diameter of pipe in feet
g = 32·2 ft./sec./sec.

This equation is a more fundamental form of Fanning's equation to be discussed later.

For *streamline flow*, Poiseuille's equation holds :—

$$p = \frac{32\eta\, Lv}{gD^2} \text{ lb./sq. ft.} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

p, L, v, g and D have the same meaning as above, η = viscosity in f.p.s. units.

For *turbulent flow* in pipes or conduits, the assumptions are made that (1) the pressure drop along the conduit equals the frictional force, and—

(2) the frictional force is assumed to be proportional to—

(*a*) the surface of contact between fluid and conduit—
i.e. the area of the conduit per unit cross-sectional area ;
i.e. the length of main × (wetted perimeter per cross-sectional area) ;
i.e. L/M.

(*b*) the square of the average fluid velocity (ft./sec.) = v^2 ;

(*c*) the density of the fluid (lb./cu. ft.) = ρ.

Hence,

$$p = \zeta \cdot \frac{L}{M} \cdot \frac{v^2}{2g} \cdot \rho \text{ lb./sq. ft.}$$

where ζ is a constant of proportionality and is the coefficient of friction, f.

2g is in the denominator, converts velocity into velocity head (see earlier in this chapter and Fig. 54).

Thus,

$$p = f \cdot \frac{L}{M} \cdot \frac{\rho v^2}{2g} \text{ lb./sq. ft.} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

This is known as Fanning's equation, and it has been shown by mathematicians and experimentally that f is determined for any given circumstances by the Reynolds number (Fig. 58).

$$\text{M, the "hydraulic depth"} = \frac{\text{cross-sectional area}}{\text{wetted perimeter}}$$

$$\text{For circular pipes M} = \pi r^2/2\pi r = \frac{D}{4}$$

$$\text{For an annular space, M} = \frac{D_1 - D_2}{4}$$

where D_1 is the inside diameter of the outer shell of a conduit of annular cross-section and D_2 is the outside diameter of the inner shell.

Thus for round pipes, Fanning's equation becomes

$$p = f \times \frac{4L}{D} \times \frac{\rho v^2}{2g} \text{ lb./sq. ft.} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (5)$$

These two equations—Poiseuille's and Fanning's—may be correlated by noting that at the critical conditions when the flow is just changing from streamline to turbulent, both must give the same value for p.

i.e.
$$\frac{32\eta Lv}{gD^2} = f \cdot \frac{4L}{D} \times \frac{\rho v^2}{2g}$$

where
$$f = \frac{16\eta}{vD\rho} = \frac{16}{Re} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (6)$$

hence the Fanning equation can be used for both streamline and turbulent flow if the appropriate value is taken for f.

Some idea of the rate of flow at the critical point for various substances can be gathered from Table 52.

The factor $\rho v^2/2g$ in the Fanning equation is the dynamic energy of the fluid in motion. For liquids, the effect of temperature need not generally be taken into account.

For gases and vapours, the values of ρ and v must be taken at the temperature and pressure ruling. As a convenient method of calculation, if P be the

TABLE 52. CALCULATED CRITICAL VELOCITIES IN STANDARD 2-INCH PIPE

Fluid	Temp., °F.	Absolute pressure, atmospheres	Specific gravity, $s = \rho/62{\cdot}3$	Relative viscosity, centi-poises	Critical velocity, ft./sec.
Hydrogen	70	1	0·0000834	0·0088	6·2
Air	70	1	0·0012	0·0184	0·904
Air	70	10	0·012	0·0184	0·0904
Steam	212	1	0·00060	0·0120	1·18
Steam	357	10	0·00524	0·0144	0·162
Water	68	Any	1·0	1·0	0·059
Refined oil	68	Any	0·90	10·0	0·65
California crude oil ..	68	Any	0·963	3,450	211

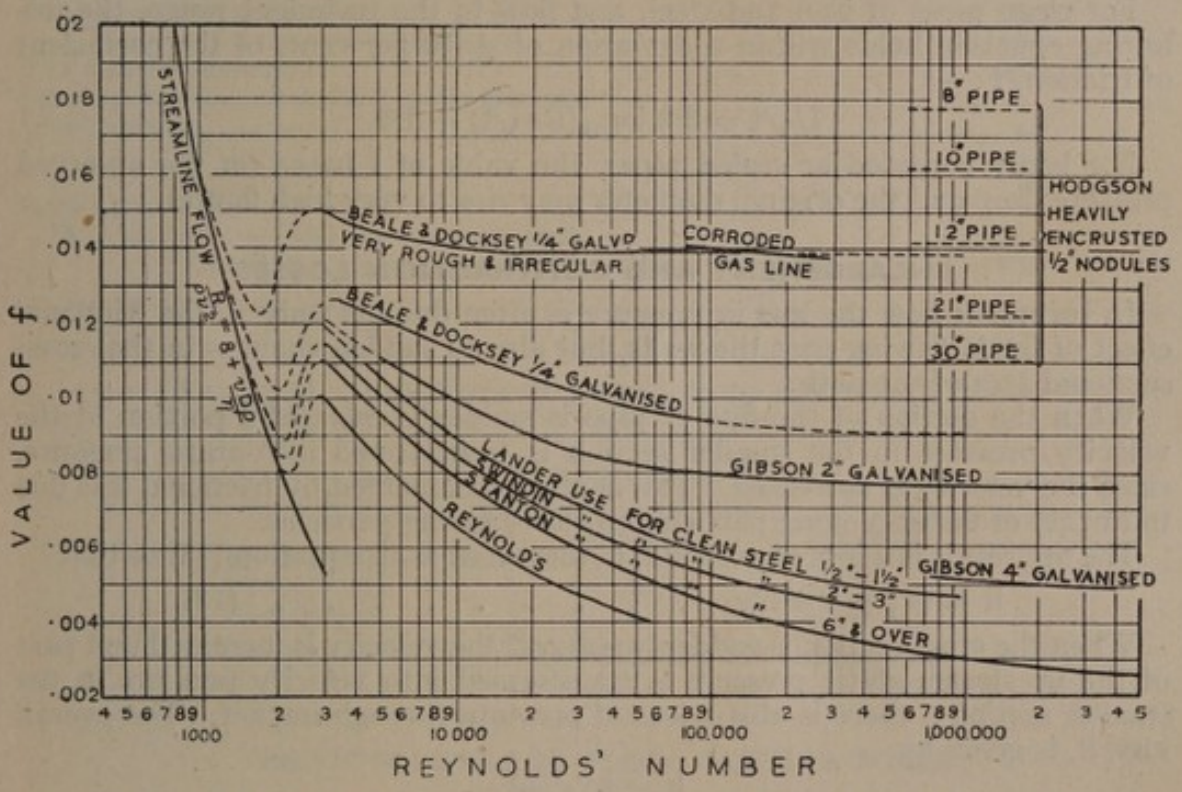

FIG. 58. Curves showing the relation between Re and f.
(Based on Fig. 8, page 64 of "Technical Data on Fuel," 4th edition.)

absolute pressure in lb. per square inch, and ρ_o and v_o are the values at 60° F. then under any temperature t °F. and pressure P lb. per square inch absolute.

$$\frac{\rho v^2}{2g} = 0{\cdot}000439\ \rho_o\ v_o^2\ (460 + t)/P \text{ lb./sq. ft.}$$

Under atmospheric pressure, P = 14·7.
Pressures in lb./sq. ft. ÷ 5·2 = inches water gauge.

The value of the friction factor, f, is taken from Fig. 58 which is a graphical representation of a large number of experimental results. In using this and similar tables it is to be noted that a common form of this method of expressing experimental results is to plot $F/\rho v^2$ (cf. equation 1) against Re. Equation (2) can be resolved into

$$p = 2(F/\rho v^2) \cdot \frac{4L}{D} \cdot \frac{\rho v^2}{2g}$$

which is the same form as equation (5). Thus when $F/\rho v^2$ is taken as the friction coefficient, its value must be doubled to accord with the value of f required in Fanning's equation.

In Fig. 58, the value of $f = 2(F/\rho v^2)$, and this value can be directly used in Fanning's equation.

Formula (5) *can be used for any form of liquid flow in circular pipes, and formula* (4) *when the pipes are not circular*, by selecting the proper value of f from Fig. 58. Due correction must be made for bends, tees, valves, etc., as given in Tables 53 and 54.

It should be noted that the product $v\rho$ in the expression for Reynolds' criterion is the mass velocity. The data given in Fig. 58 shows how the frictional resistance varies with the roughness of the surfaces of the ducts.

Koo showed that for a limited range of Re from 5,000 to 200,000

$$f = 0{\cdot}049/Re^{0{\cdot}2}$$

For clean pipes of iron and steel, and flow in the turbulent range, the following equation holds within a deviation of $\pm$ 10 per cent. of the coefficient of friction, f :—

$$1/\sqrt{f} = 3{\cdot}2 \log_{10}(Re\sqrt{f}) + 1{\cdot}2.$$

For badly corroded or scaled pipes, the value of f based on the observed pressure drop and the original diameter may rise to very high figures.

ENLARGEMENT AND CONTRACTION LOSSES

In turbulent flow the loss in pressure is often due not only to the frictional effect of fluids flowing past the walls, but also to sudden changes in the cross-sectional area of the path.

When the section of the flue or pipe is suddenly *enlarged* a portion of the velocity pressure in the smaller section is transformed into static pressure, since the velocity is decreased. This is also accompanied by frictional loss due to impact of faster-moving particles on slow-moving particles.

The net result is a loss of pressure, h, measured as in equation (13) so that

$$h = (v_1 - v_2)^2/2g \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (13)$$

When the cross-section is suddenly *reduced*, the velocity is increased and part of the up-stream static pressure is transformed into velocity pressure in the smaller section ; there is also a loss of pressure through impact. The overall loss, h, is given by

$$h = kv_2^2/2g$$

where v_2 is the velocity in feet per second in the smaller section and k is a numerical factor which depends on the ratio of the smaller to the larger pipe area.

When this ratio =	1	0·8	0·6	0·4	0·2	0
k =	0	0·15	0·25	0·34	0·43	0·5

EFFECT OF TEES, VALVES, ELBOWS AND JOINTS IN PIPE LINES

The effect of changes in direction due to tees or elbows is best allowed for by regarding the frictional effect as equivalent to a length of straight pipe. This length would be added on to L in the formulæ given in this chapter. Table 53 represents recommended practice.

Joints in pipe lines increase the friction loss, since in effect they accentuate the roughness of the interior surface. The friction coefficients given in Fig. 58 are for straight lengths of pipe without joints. All industrial pipes are jointed, of course, and the values of the coefficient should be increased as in Table 54 in accordance with the Reynolds number.

TABLE 53. FRICTION LOSS DUE TO BENDS, ELBOWS, VALVES AND TEES FOR CIRCULAR PIPE

This table applies to all fluids, but for turbulent motion only.

Type of bend	Diam. of pipe ins.	Equivalent length, expressed as diameters of additional straight pipe, i.e. L/D
90° bend	⅜–2½	30
	3–6	40
	7–10	50
45° bends 90° bend (long radius) }	1–3	15–20
90° elbow (sharp)	—	50
90° curve, same inside diameter as pipe :—		
centre line radius = diam. of pipe ..	—	20
do. do. = 2–8 diameters ..	—	10
Tee, full-size branch	1–4	60–90
(use higher value when fluid enters branch)		
Globe valves :	⅜–2½	45
	3–6	60
	7–10	75
Gate valves—open	—	7
Do. do. —Rises with successive closed positions, till at ¾-closed it is	—	800

TABLE 54. PERCENTAGE INCREASE IN THE VALUE OF f

Re	Rough joints	Smooth joints
	Per cent.	Per cent.
Up to 5,000.. ..	15	5
5,000–20,000 ..	10	3
Over 20,000.. ..	5	2

FRICTION LOSS FOR FLOW OUTSIDE TUBES

Parallel to Axis. Generally the treatment already recommended suffices.

Flow Normal to Axis. The friction for turbulent flow of fluid normal to a bank of tubes, n rows deep, may be considered as due to n contractions and enlargements, thus :—

$$f = 4f'n\frac{v^2_{max.}}{2wg}$$

where $v_{max.}$ = maximum velocity, feet per second
w = mass of fluid, lb.
g = gravitational constant

f′ has been shown by Andreas and Grimison to depend upon the arrangement. For tubes in line

$$f' = \left(0{\cdot}044 + \frac{0{\cdot}08x_1}{(x_t - 1)^n}\right)\left(\frac{vD\rho}{\eta}\right)^{-0.15}$$

For staggered tubes

$$f' = \left(0{\cdot}23 + \frac{0{\cdot}11}{(x_t - 1)^{1.08}}\right)\left(\frac{vD\rho}{\eta}\right)^{-0.15}$$

Where x_1 = ratio of the longitudinal pitch to tube diameter
x_t = ratio of the transverse pitch to tube centre to centre distance.
D = outside diameter in feet.
n = No. of rows
η = absolute viscosity expressed in f.p.s. units.

Values of $\frac{vD\rho}{\eta}$ applicable are between 2,000 and 40,000.

THE TRANSMISSION OF GASES

Gaseous flow requires special treatment if the pressure changes materially along the pipe.

LOW PRESSURE GAS TRANSMISSION

First, let it be supposed that the pressure does not change materially, so that the Fanning equation is directly applicable. It is convenient for practical reasons to use notation somewhat different from that of equations (4) and (5). Thus v will be replaced by Q, the quantity of gas flowing in cubic feet per hour at S.T.P. ; D the diameter of the mains in feet by d, the diameter in inches ; ρ, the density of the gas by s, its sp. gravity (compared with air as 1) ; and p (lb. per square foot) by P (inches w.g.).

Clearly, $d = 12D$

and, since air weighs 0·076 lb./cu. ft. at S.T.P.

$$\rho = 0{\cdot}076\ s$$
$$P = p/5{\cdot}2.$$

Consequently, the expression

$$p = f \cdot \frac{4L}{D} \cdot \frac{\rho v^2}{2g}$$

becomes

$$P = \frac{4 \times 12 \times 0{\cdot}076}{2g \times 5{\cdot}2} \cdot fLs\ v^2/d$$
$$= 0{\cdot}01089\ fLs\ v^2/d \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (7)$$

But Q (cu. ft./hr.) $= \frac{3{,}600\ v\pi D^2}{4}$

$$= \frac{3{,}600\ v\pi d^2}{4 \times 144}$$

whence $v = Q/19{\cdot}63\ d^2$

Inserting this value in (7) and solving for Q

$$Q = 188\sqrt{\frac{d^5P}{fsL}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (8)$$

If f be taken as 0·0065—a constant value for all conditions—and L be expressed in yards instead of feet, this formula becomes

$$Q = 1{,}350\sqrt{\frac{d^5P}{sL}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (9)$$

which is Pole's well-known formula for the flow of gas in pipes that has for so long been used by the gas industry.

Pole's formula (9) is reasonably correct for low pressures up to about 6–8 inches w.g. It is less correct than (8), now known as Lacey's formula, even at these low pressures.

Lacey's formula (8) can be used up to about 3 lb. per square inch gauge pressure, the value of f being read from Fig. 58, as before.

HIGH PRESSURE GAS TRANSMISSION

For high pressure gas transmission (see Unwin, "Treatise on Hydraulics," Chapter XI, and Lacey, *Trans. Inst. Gas E.*, 1927) the formula adopted in gas units is

$$Q = 2{,}675 \sqrt{\frac{d^5}{fsL} \cdot \frac{(p_1^2 - p_2^2)}{p_a^2}} \text{ cubic feet per hour at S.T.P.} \quad . \quad (9a)$$

where p_1 and p_2 = absolute pressure in lb./sq. in. (not per sq. ft.) at beginning and end of pipe

p_a = absolute pressure of atmosphere in lb./sq. in. (generally 14·7 lb./sq. in.)

L = length of main in feet.

Other formulæ have been proposed, but Unwin's is probably the soundest fundamentally.

THE MEASUREMENT OF FLUID FLOW

The practical side of the measurement of gases, steam and liquids is discussed in Chapter X. An account is here given of the principles on which it is based.

From equation (1) it can be seen that if a pipe or duct in which a fluid is flowing is horizontal, $x_1 = x_2$.

Over a very short length, f may be neglected. If no work is done on the fluid, $W = 0$. For liquids, $X = 0$, and for gases, when the pressure drop is small, X can be neglected. Thus equation (1) becomes

$$p_1V_1 + v_1^2/2g = p_2V_2 + v_2^2/2g$$

whence

$$\begin{aligned} v_2^2 - v_1^2 &= (p_1V_1 - p_2V_2) \cdot 2g \\ &= h \cdot 2g, \end{aligned}$$

where h is the difference in head (expressed as feet head of the liquid or gas under consideration) between two points taken.

From this

$$\sqrt{v_2^2 - v_1^2} = \sqrt{2gh} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (10)$$

THE STANDARD ORIFICE

If an orifice be inserted in a pipe line as in Fig. 59, a pressure drop is caused

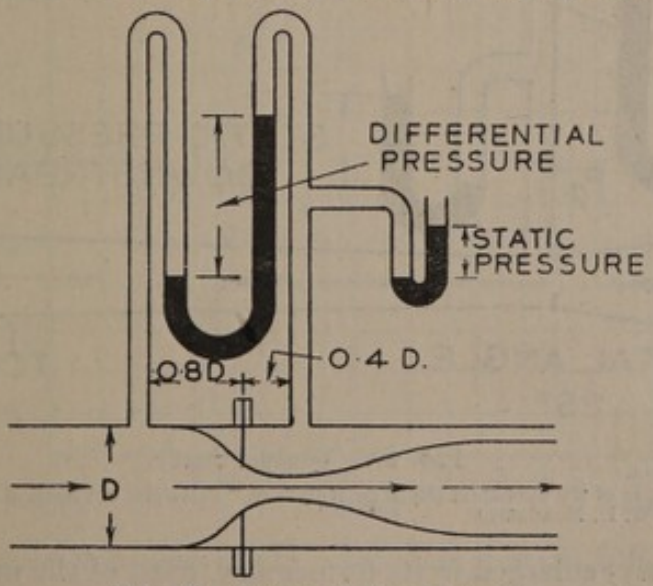

FIG. 59. Orifice plate and gauge.

(Reproduced by permission of the publishers (McGraw-Hill) from "Principles of Chemical Engineering," by W. H. Walker, W. K. Lewis and W. H. McAdams.)

and in this way a portion of the static head h_1 (Fig. 54) is converted into velocity head in the restricted section. The observed decrease in static pressure (obtained by a differential gauge) is caused by the increase in fluid velocity and theoretically the flow can be calculated as in equation (10).

In practice the flow is less than that given in equation (10), because this has been based on the assumptions that there is no friction loss and that the stream does not contract. As neither of these assumptions is correct, the discrepancy is allowed for by introducing a " coefficient of discharge," α, as in equation (11).

$$\sqrt{v_2^2 - v_1^2} = \alpha\sqrt{2gh} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (11)$$

where v_1 is the average up-stream velocity in feet per second

v_2 is the average velocity through the orifice in feet per second.

If a sharp-edged orifice be used and its diameter is large compared with the thickness of the plate, $\alpha = 0{\cdot}61$ (for exact values, see Chapter X).

If the diameter of the pipe be five times that of the orifice, the contraction of the stream, and therefore its velocity, is greatest at a distance down-stream 4/10ths of the diameter of the pipe. By situating one measuring pipe at this point, and another not less than 8/10ths diameter up-stream from the orifice, formula (11) simplifies to

$$v_2 = \alpha\sqrt{2gh} \quad . \quad . \quad . \quad . \quad . \quad . \quad (12)$$

since v_1^2 is negligible compared with v_2^2.

The sole disadvantage of this method lies in the fact that the suddenness of the velocity change causes a permanent loss of much of the observed drop in static pressure, which is converted into heat due to impact and internal friction. If the ratio of orifice diameter to the diameter of the pipe be 1 : 5, the loss is 93 per cent. of the static pressure drop. With a ratio 1 : 3, the loss is 85 per cent. and at 1 : 1·5, 55 per cent.

THE VENTURI METER

If the orifice is streamlined and the change in section is gradual α can become very nearly 1, say 0·97 or 0·99. Moreover, if the pipe section be gradually

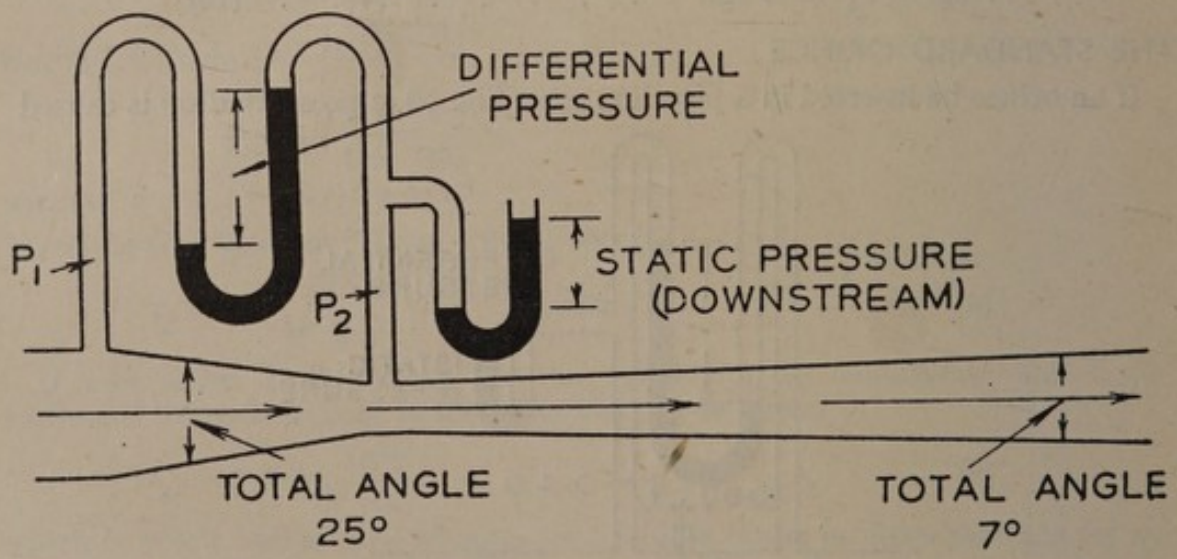

FIG. 60. Venturi meter.

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(instead of suddenly) enlarged to its former size, most of the increased velocity head is not lost, but is reconverted into static pressure. Under these conditions the orifice becomes a throat.

This arrangement and the general conditions required to make it effective are shown in Fig. 60. This is known as the Venturi meter. The equation of flow is equation (11). The permanent loss of static pressure with the arrangement shown in Fig. 60 is only 10–12 per cent. of the Venturi reading.

This meter is unsatisfactory with viscous liquids.

THE PITOT TUBE

If, as in Fig. 61, one limb (*a*) of a differential pressure gauge is connected to an opening flush with the side of the main or pipe and the other (*b*) is connected

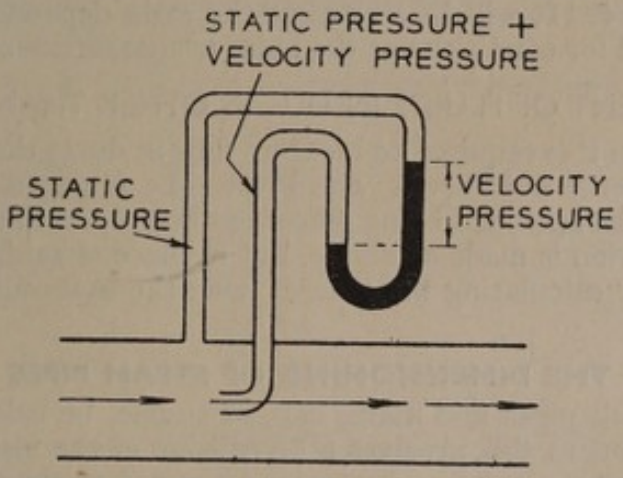

FIG. 61. Pitot tube.

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to a tube having its open end pointing against the gas stream, pipe *a* is subjected to the static pressure only and pipe *b* to the sum of the velocity and static pressures, so that the differential reading will indicate the velocity head.

The velocity head, as has been indicated in Fig. 54, is $v^2/2g$. Therefore if

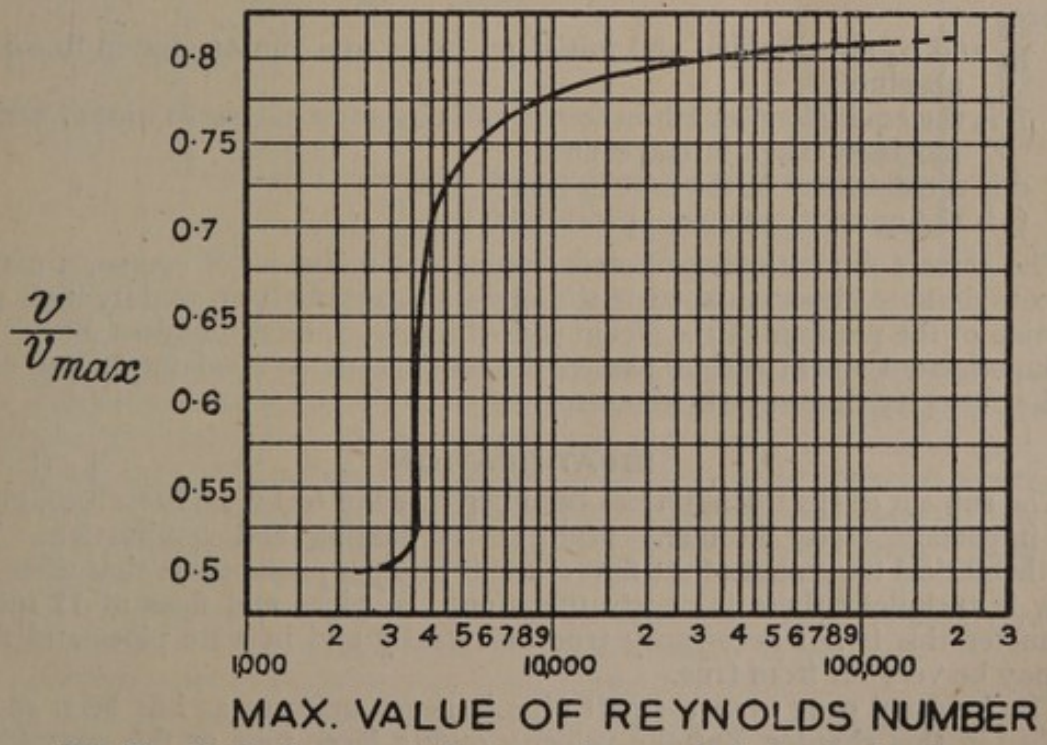

FIG. 62. Ratio of mean velocity to maximum velocity in circular pipes.
(From *Nat. Phys. Lab.*, 11, 1914.)

the reading of the differential gauge, expressed in feet of the fluid under consideration, is h,

$$h = v^2/2g$$

$$v = \sqrt{2gh} \quad . \quad . \quad . \quad . \quad (12a)$$

This measured velocity, v, is the velocity at the point where the tube facing the gas stream is situated. If this tube is placed along the centre line of the pipe, and there are no disturbing effects from bends, etc., the ratio of the average velocity throughout the pipe to the maximum velocity (at the centre line) is given by Fig. 62, which represents results obtained at the National Physical Laboratory in 1914. It will be seen that the ratio depends on the Reynolds number calculated for conditions at the point of measurement.

THE MEASUREMENT OF FLUIDS IN DUCTS OTHER THAN CIRCULAR

In general, when it is required to measure flow in ducts that are not circular, the problem concerns hot gases. The Pitot tube is generally the most convenient method, the open end being placed, as before, on the central line of the flue. The calculation is made as above, but in place of taking the diameter of a circular tube for calculating the area of flow, the hydraulic mean depth, M, is used.

THE DIMENSIONING OF STEAM PIPES

The size of steam pipes and mains can, of course, be calculated from Fanning's equation, but as this involves a knowledge of the viscosity of steam at possibly high pressures and temperatures, saturated or superheated, the usual practice is to adopt experimentally determined formulæ, necessarily of an empirical nature.

A formula stated to be accurate is due to Dr. Wierz of Charlottenburg. This formula as modified for English units is given below, the various numbers contained in it being derived from his experimental results :—

$$\frac{p_2^{1\cdot9375} - p_1^{1\cdot9375}}{L} = 0\cdot00012\,\frac{Q^{1\cdot853}}{d^{4\cdot987}}$$

where,

p_1 and p_2 are the final and initial pressures in a run of pipe in lb./sq. in. absolute.

L is the equivalent length in feet of the pipe after the resistance of fittings has been taken into account,

d is the diameter of the pipe in inches, and

Q is the quantity of steam passing in lb./hr.

The correct dimensioning of steam pipes and mains is, of course, a matter of considerable importance, as if the pipes are too small an unduly high proportion of the pressure of the steam is used in overcoming the resistance.

Superheated steam can be passed through the pipes at about 50 per cent. higher velocity than saturated steam.

STRATIFICATION

The subject of stratification has been neglected in technical literature, but is an important factor in many problems of heating and combustion. The mathematical treatment of the flow of fluids in pipes presupposes that when the flow is turbulent there is no stratification. In pipes and flues of 12 inches diameter this is not necessarily true universally, and in wide pipes and flues it may be very far from true.

The velocity over even a small pipe is never uniform as has been shown earlier in this chapter, but the velocity over a large pipe or flue completely filled with the flowing gas (and especially if the flow is turbulent) may be

virtually uniform except at the edges ; on the other hand there may be wide differences in velocity, temperature and composition over the area of the flue.

As a result of stratification, stock in furnaces may be immersed in gases of a temperature different from the average that was anticipated ; measurements of temperature in flues may be very misleading unless the whole cross-section of the flue is explored ; and the CO_2 content of the flue gases may vary in different parts of the flue.

In large flues two effects are always operative, the natural buoyancy of the gas, which tends to give separation of the heavier and lighter constituents in so far as such diffusion is not interfered with by turbulence, and the setting

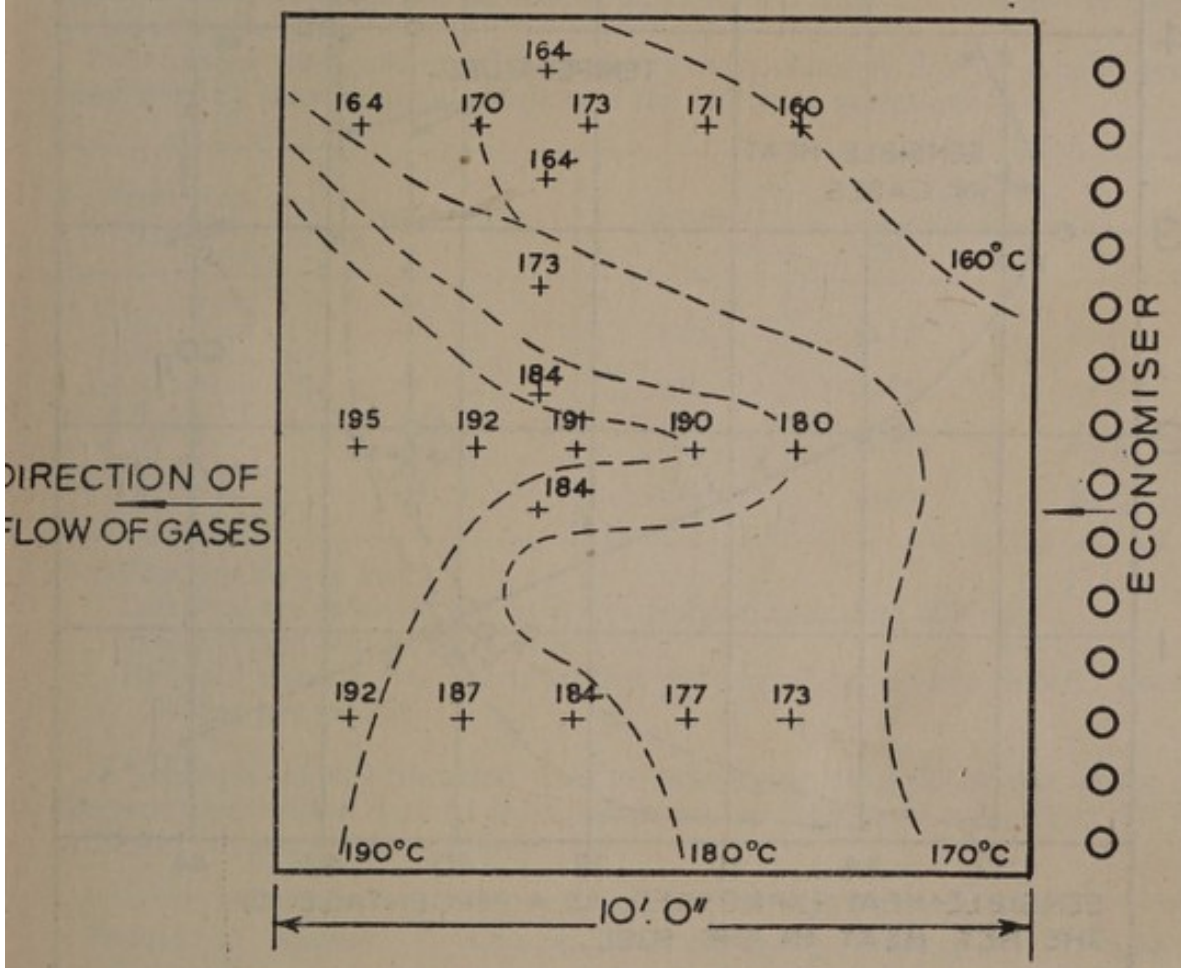

FIG. 63A. Stratification of gases at an economiser outlet.

up of secondary effects due either to heating or cooling of the gases by means of the flue walls. Attempts have been made to correlate the depth of a stream of hot gas flowing into a large duct from a smaller one on the basis of applying the analogy of water flowing over a weir, conditions being inverted when hot gas flows into the colder. For a discussion of the subject the reader is referred to Groume-Grjimailo's work in the "Flow of Gases in Furnaces" (English translation by A. D. Williams).

In practice the conditions of flow in large flues are complicated and the general formulæ worked out for smaller flues do not necessarily apply.

Stratification may also arise from other causes :—

(1) Local cooling, especially if the bottom of a flue be cooler than the top.
(2) Inleakage of air causing in effect two converging streams of different temperature and composition.

(3) Converging gas streams at different temperatures and of different compositions meeting in a common flue.
(4) Jet effects introduced by high velocity streams entering those at lower velocity.

These causes are likely to set up differences in temperature and composition.

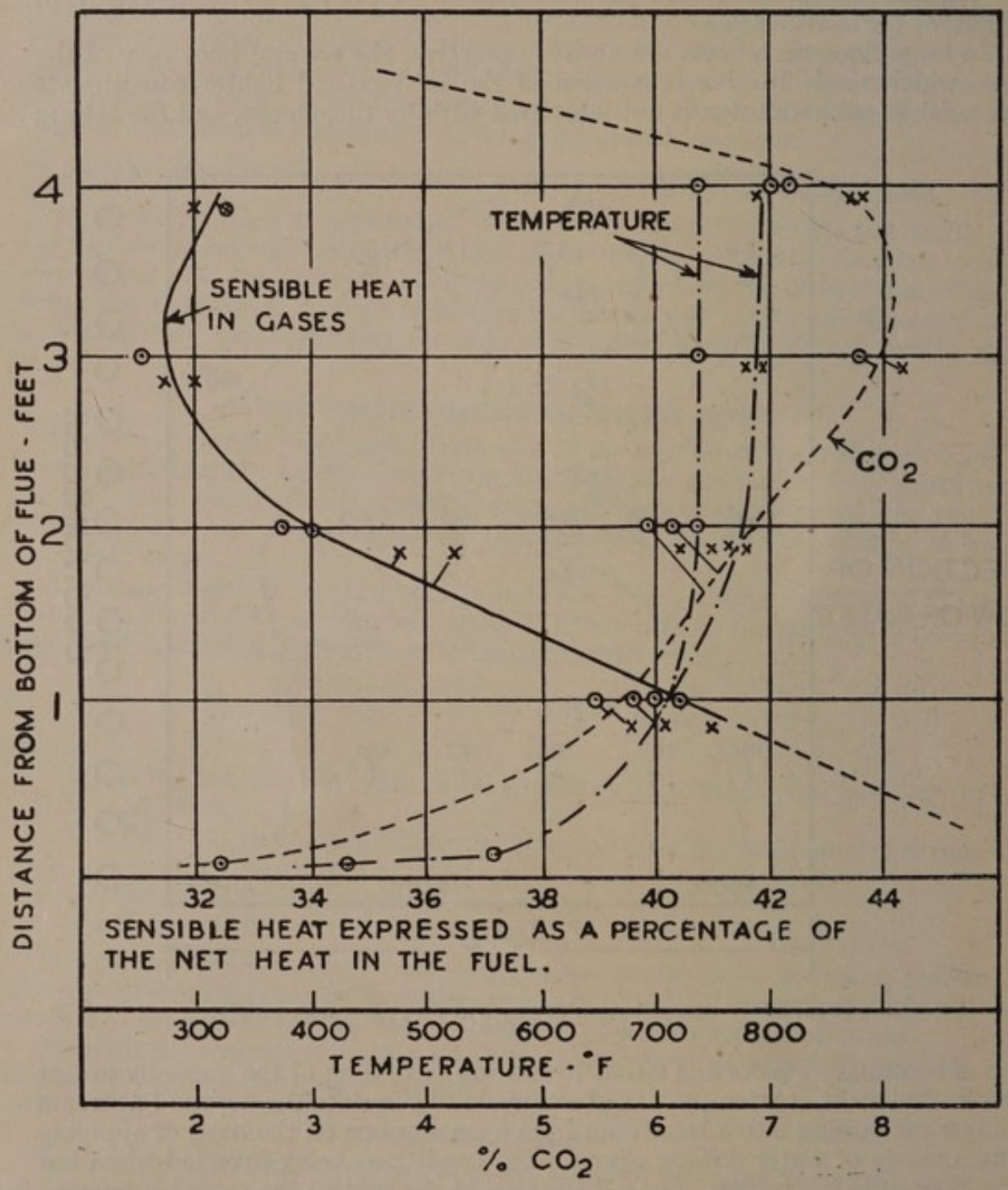

FIG. 63B. Stratification of gases in a boiler flue.

If the gas stream completely fills the flue and the flow is highly turbulent (i.e. Reynolds number high) the differences soon disappear, but if the Reynolds number is low and particularly if the gas stream does not fill the flue, the stratification may persist indefinitely.

Fig. 63a shows the temperature distribution in a boiler flue at the outlet from the economiser with the deflectors open and indicates that a single tem-

perature taken at this position would be quite unreliable as an indication of the true temperature of the gases. Fig. 63b shows an even more striking example of differences in temperature and CO_2 content occurring in a boiler side flue 5 feet deep by 1 foot 6 inches wide, the temperatures and CO_2 contents being taken from top to bottom.

The maximum content of sensible heat in the gases flowing in the flue is shown to occur towards the top of the flue, and it is apparent that in this zone there is a greater flow of the products of combustion than in the lower parts of the flue. The temperatures were measured with bare thermocouples of 18G, and would therefore indicate a temperature more nearly approaching the true temperature of the gas than would be found with the usual type of rod pyrometers, whose readings are influenced by radiation from the adjacent hot surfaces of the flue.

Two further examples may be cited. A main chimney flue for a coke oven plant fired by blast furnace gas showed the following variations :—

Depth from top of flue	Temperature	Depth	Analyses	
			CO_2	O_2
6 ins.	317 °C.			
1 ft. 6 ins.	421 °C.	1 ft.	19·8%	4·2%
2 ft. 6 ins.	317 °C.			
3 ft. 6 ins.	301 °C.	3 ft.	19·0%	4·6%
4 ft. 6 ins.	290 °C.			
5 ft. 6 ins.	286° C.	5 ft.	19·6%	4·0%
6 ft. 6 ins.	283° C.			
7 ft. 6 ins.	281° C.	7 ft.	19·0%	4·6%

Flue depth = 8 feet.
Temperature recorder was at 3 feet position recording 309° C.
True average temperature = 300° C.
The cold velocity was about 10 feet per second or the hot velocity about 20 feet per second.

An example of stratification due to converging streams of flue gases of different composition derived from coke oven gas (combined velocity at 60° F. = 10 feet per second) is as follows :—

Position	Bottom	4 ins. up	8 ins. up.	12 ins. up	16 ins. up (top)
CO_2% ..	6·5%	6·8%	8·8%	7·9%	6·0%
Temperature	400° C.	445° C.	498° C	490° C.	420° C.

THE FLOW OF GASES IN FURNACES

A natural draught furnace system consists of a series of communicating chambers open at two points to the atmosphere, through which gas and air are drawn as a result of the liberation of heat by combustion at some point within the system.

The flow of gases in existing furnaces can be studied by taking the pressure and temperature of the gases at different points. When designing new furnaces the dimensions and shape of the chambers and intercommunicating passages or flues must be fixed partly by the requirements of the process and partly by known data on the flow of gases.

The first step in the aerodynamic design of a furnace is to estimate as fully as possible the force available to bring about the required gas flow. This will be the algebraic sum of a series of chimney effects and the initial pressure :—

Chimney effect where hot gas rises — chimney effect where hot gas descends — resistances in the system + initial pressure.

Chimney effects have been dealt with in some detail in Chapter VI to which readers are referred for methods of calculation. It is necessary here to distinguish between the chimney effect of a rising flue or chamber and the effect on the system of the main chimney. Boilers and those furnaces which have no working doors or openings for the charging of material to be heated operate by the draught provided. This creates the motive power necessary for moving the air through the furnace and subsequent flues.

In furnaces which have working doors to their heating chambers, e.g. reverberatory furnaces, melting furnaces, puddling furnaces, the main chimney is nothing more than a means of removing the products of combustion. The draught effect of the main chimney makes itself felt at some arbitrarily fixed point in the system. Thus a furnace may operate with a slight pressure in the furnace chamber and following this chamber there will be a section of zero pressure from which the chimney operates to remove the products of combustion.

BUOYANCY

If a closed vertical tower contains a gas lighter than air, it can be shown that the buoyancy effect of the gases sets up a difference in pressure between the top and bottom of the tower equal to the difference in weights of 1 cubic foot of air and of the lighter gas multiplied by the height of the column,

i.e. $$p_s = p_1 - p_2 = (w_a - w_g)H \text{ lb./sq. ft.} \quad . \quad . \quad . \quad (14)$$

where p_1 and p_2 are the gauge pressures at the top and bottom of the tower respectively in lb./sq. ft.

w_a and w_g are the densities of the outside air and the gas in lb./cu. ft. under the conditions prevailing.

H is the height of the tower in feet.

The value obtained by (14) divided by 5·2 gives the pressure difference in inches w.g.

If the top of the tower is open to the atmosphere the system is in effect a chimney with the damper closed. Clearly p_1, the pressure difference between the tower and the atmosphere, = 0. Equation (14) becomes

$$p_2 = -(w_a - w_g)H \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (15)$$

p_2 being the measured static suction at the chimney base. Similarly if the bottom were open and the top closed, $p_2 = 0$ and

$$p_1 = (w_a - w_g)H \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (16)$$

If both bottom and top were open and the relative densities of air and gas were maintained—this being the condition met in practice—there would be a flow of gas up the chimney arising from exactly the same cause, but part of the static pressure difference, $p_1 - p_2$, would be converted into velocity head and part would be lost in friction, the net effort being given by Fanning's equation (equation (4)).

EFFECT OF BENDS AND OBSTRUCTIONS

In furnace draughting the conditions of flow differ to some extent from the flow in tubes. It is found in the large flues and ducts in furnaces carrying hot gases which are radiating to a colder surface that a more complex system of

flow occurs. The variation in conditions of flow arises mainly from the separation of hot and cold layers of gas in the furnaces, the effect of burners and the location of flue offtakes.

PRESSURE DROP IN FURNACES

It is of considerable interest to be able to calculate the pressure drop through a furnace or similar structure, and although the problem is essentially one of design, ability to know when resistances are abnormal is often of considerable value to the operator. The best method of describing how this can be done is to give an example. The following calculations are due to Dr. A. H. Leckie of the Iron and Steel Industrial Research Council.

In designing any furnace the resistance to gas flow through the system must be calculated to ensure that the necessary quantities of gas and air will enter the furnace with the pressure available and to calculate the draught necessary to exhaust the waste gases and infiltrated air.

In many furnaces, particularly those of the reversing regenerative type such as glass tanks or open hearth steel melting furnaces, most of the pressure drop is caused by the numerous bends and changes of section, and is due in only a minor degree to wall friction. In addition, substantial pressure changes are caused by the variations in vertical level through the furnace system, and the gas temperature is constantly changing. Therefore, although the pressure loss through a furnace may be evaluated by the Fanning equation (equation (4)), the numerous determinations of equivalent length, hydraulic mean depth, and Reynolds number involved make the calculation complex and it is preferable to use the alternative method outlined below.*

The total static pressure change through a furnace system (P) is divisible into two components, that due to eddying caused by bends, changes of section, and wall friction, which is proportional to the square of the N.T.P. velocity of the gas and is denoted here by p_v, and that due to changes in vertical level (buoyancy changes) denoted here by p_s, which is independent of the gas velocity.

In Fanning's equation (equation (4)), the factor $\rho v^2/2g$ is the dynamic or kinetic energy of 1 actual cubic foot of gas under the conditions in the flue ; it is commonly termed the " velocity head." If v_o and ρ_o are the velocity (feet per second) and the density (lb. per cubic foot) at N.T.P., then at t° C.

$$\rho_o v_o^2 \text{ becomes } \frac{\rho_o(273)}{(273+t)} \times \frac{v_o^2(273+t)^2}{(273)^2}$$

$$\text{or } \frac{\rho_o v_o^2(273+t)}{273}$$

g is unaffected by the temperature and thus at any temperature t° C., $\rho v^2/2g$ may be written in terms of ρ_o and v_o ; thus for air the expression becomes

$$\frac{0{\cdot}0805}{5{\cdot}2 \times 64{\cdot}4} \times \frac{v_o^2(273+t)}{273}$$

or $8{\cdot}80 \times 10^{-7}\, v_o^2(273+t)$ inches w.g.

and for any gas of specific gravity s (air = 1)

$$8{\cdot}80 \times 10^{-7}\, s.v_o^2\,(273+t) \text{ inches w.g.} \quad . \quad . \quad . \quad (17)$$

At every bend and change of section a certain fraction (f_1) of this energy is lost due to eddying, and at the walls a fraction f_2 is lost due to wall friction. Hence the total pressure loss, p_v, due to eddying and wall friction is given by

$$p_v = (f_1 + f_2)p_o \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (18)$$

* For a fuller description of this method of calculating furnaces see H. Etherington's " Modern Furnace Technology," pp. 124–190 (C. Griffin & Co. Ltd.).

and the total pressure (or draught) loss given by

$$P = (f_1 + f_2)p_o \pm (p_1 - p_2) \quad . \quad . \quad . \quad . \quad . \quad . \quad (19)$$

The choice of sign depends on whether buoyancy assists or opposes gas flow in the direction required.

The value of $p_1 - p_2$ may be calculated from equation (14) since

$$p_1 - p_2 = (w_a - w_g)H = p_s$$

where H is the height of the column of hot gas in feet. For air, for which w_a at atmospheric temperature may be taken as 0·0765 lb. per cubic foot

$$p_1 - p_2 = \frac{(0{\cdot}0765 - 0{\cdot}0765s)H}{5{\cdot}2}$$

$$= 0{\cdot}0147H(1 - s) \text{ inches w.g.}$$

And at t° C., when t_a = atmospheric temperature in ° C.,

$$p_1 - p_2 = 0{\cdot}0147H\left[1 - \left(\frac{273 + t_a}{273 + t}\right)s\right] \text{ inches w.g.} \quad . \quad (20)$$

The crux of the pressure calculation is the selection of the correct values of f_1 and (to a lesser extent) f_2. (This is, of course, a similar problem to the determination of equivalent length in Table 53.)

DETERMINATION OF f_2

The value of f_2 may be disposed of briefly (it is the value of $f\frac{L}{M}$ in the Fanning equation). Normally it is necessary to calculate the Reynolds number and the hydraulic mean depth, and so obtain the appropriate value after applying a correction for the nature of the surface of the duct. However, most furnace passages are constructed of brick, the surface of which becomes very rough after short use, and accurate determination of f_2 is not possible. Since friction plays a relatively unimportant part in the total pressure drop, it is sufficient to consider only average flow conditions and to determine f_2 from the dimensions of the flue only. For a fuller justification of this see Etherington (loc. cit. p. 155), but for ordinary works' calculation f_2 may be determined as

$$f_2 = 0{\cdot}0125\frac{LC}{A} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (21)$$

where L is the length of the flue in feet.
C is the cross-sectional perimeter of the flue in feet.
A is the cross-sectional area of the flue in square feet.

DETERMINATION OF f_1

Values of f_1 for bends and changes of section are listed by various authorities. Unfortunately the recommended values vary greatly, but those given below are what may be considered the best average as far as is known at present. (See Fig. 64.) Where there is a change in the N.T.P. velocity (i.e. velocity of the gas assumed measured at N.T.P.) the f value should be applied to the higher N.T.P. velocity.

(*a*) Right angle bend :—$f_1 = 1{\cdot}5$.
(*b*) Right angle bend made from T-piece by blanking off one short limb :—$f_1 = 2{\cdot}0$.
(*c*) 45° bend :—$f_1 = 0{\cdot}5$.
(*d*) Rounded bends :—If $R > 5D$, $f_1 = 0$; if $R < 5D$, f_1 increases up to the full 1·5 value for a 90° bend, according to R/D. (R = mean radius of bend, D = diameter of duct.)

(*e*) Ultra-sharp bends such as are found between uptakes and ports :— f_1 may be 2·0 or more if the inside of the bend is not rounded off, but most bends of this type have the inside corner cut off to some extent either by design or through erosion after use when the f_1 value is lower.

(*f*) Entry to passage from large chamber or from atmosphere :—$f_1 = 2{\cdot}0$.

(*g*) Exit from passage to large chamber or to atmosphere :—$f_1 = 1{\cdot}0$.

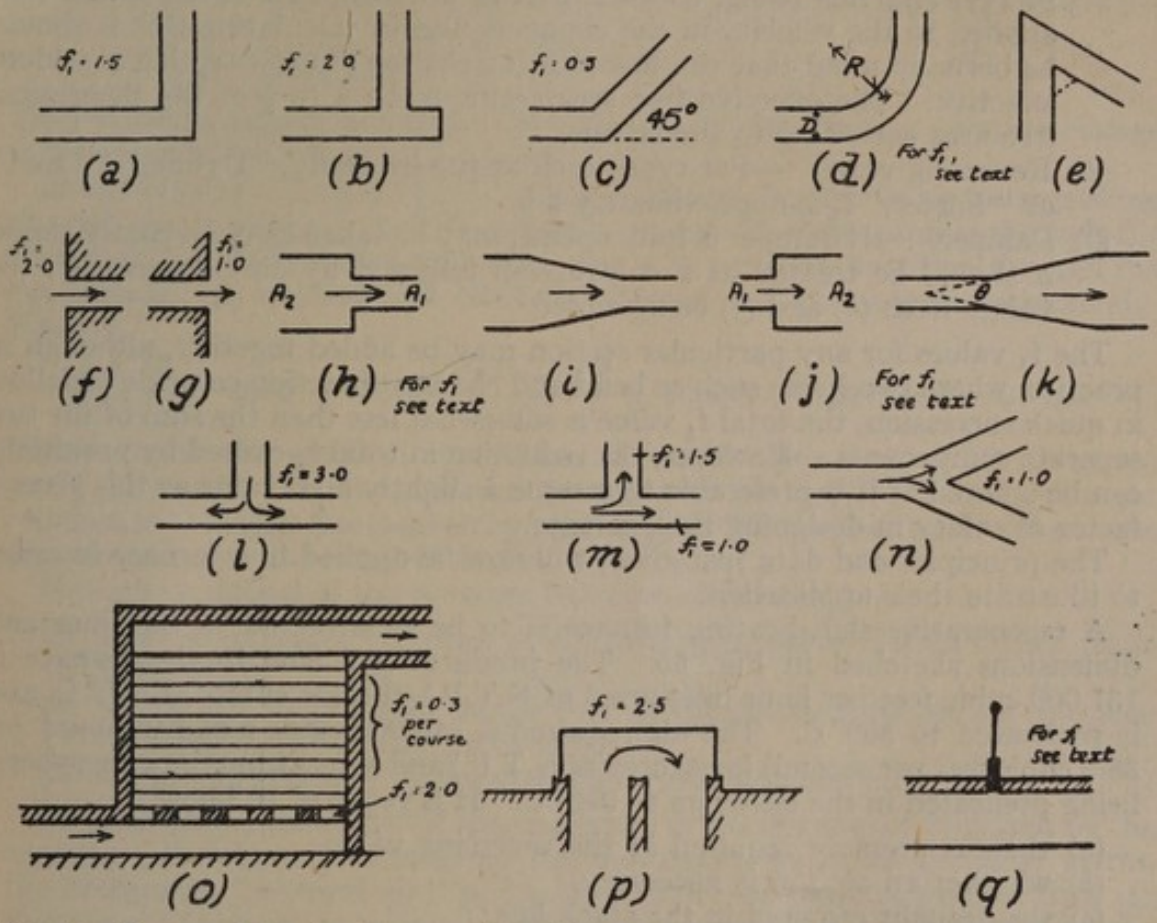

FIG. 64. Types of bends, changes of section, etc., commonly encountered in furnace systems.

(*h*) Sharp-edged contraction :—f_1 depends on the ratio of the areas. Standard curves showing f_1 plotted against this ratio are given in text-books (e.g. Etherington, p. 148), but in practice it is found that the f_1 values so obtained are too low and the value used should be about three times that given by the curve. The recommended values of f_1 are as follows :—

Area ratio (A_1/A_2) =	0·1	0·2	0·4	0·6	0·8	1·0
f_1 =	1·08	1·02	0·81	0·5	0·16	0

(*i*) Gradual narrowing :—f_1 very small.

(*j*) Sharp-edged enlargement :—$f_1 = \left(\frac{A_2 - A_1}{A_2}\right)^2$ where A_1 is upstream and A_2 is downstream area.

(*k*) Gradual enlargement :—$f_1 = k \cdot \left(\frac{A_2 - A_1}{A_2}\right)^2$. According to Spiers' "Technical Data on Fuel," 4th edition, p. 74, k has the following values for varying values for a gradual enlargement having a total angle of $\theta°$:—

θ =	8	14	20	30	45	60	90–180 degrees
k =	·15	·25	·45	·7	·95	1·1	1·0

(*l*) Subdivision of stream :—$f_1 = 3{\cdot}0$.
(*m*) Subdivision of stream :—$f_1 = 1{\cdot}0$ or $1{\cdot}5$ as shown.
(*n*) Subdivision of stream :—$f_1 = 1{\cdot}0$.
(*o*) Chequers :—For entry to chequers $f_1 = 2{\cdot}0$. For the chequer courses themselves the f_1 * value of 0·3 per course of bricks for unstaggered patterns is suggested by Etherington (" Modern Furnace Technology," p. 144) and this seems to work well in practice. However, it must be applied to the velocity in the chequers and in calculating this it should be borne in mind that the theoretical free area of cross-section is seldom effective. The effective free area seems to be $\frac{1}{2}$ to $\frac{2}{3}$ of the theoretical free area according to the layout.
(*p*) Reversing valves :—For types such as the butterfly, " Dyblie," " Lake," or " Forter," f_1 is approximately 2·5.
(*q*) Dampers :—If damper is fully open f_1 may be taken as 0 ; if partly closed it should be treated as a contraction followed by an enlargement, the values from (*h*) and (*j*) being added.

The f_1 values for any particular section may be added together, although in practice, where two items such as bend and change of section coincide or follow in quick succession, the total f_1 value is somewhat less than the sum of the two separate components. However, the reduction in total f_1 caused by proximity can be ignored as it is preferable to assume a slightly high value as this gives a factor of safety in designing the passages.

The principles and data just given will now be applied to a furnace in order to illustrate their application.

A regenerative slab heating furnace is to be constructed to the lines and dimensions sketched in Fig. 65. The producer gas flow to the furnace is 137,000 cubic feet per hour (measured at N.T.P.), the gas enters at 500° C. and is preheated to 800° C. The air required is 140,000 cubic feet per hour† (or 38·9 cubic feet per second) (measured at N.T.P.) and enters from the atmosphere, being preheated in the chequers to 900° C. It is required to know

(*a*) the gas pressure required at the reversing valve.
(*b*) whether an air fan is necessary,
(*c*) the draught required in the stack flue.

The pressure balances are most conveniently calculated in tabular form. Table 55 is that for ingoing gas and Table 56 that for ingoing air. The furnace is most conveniently divided into the sections listed in Column 1. Column 2 shows the average temperature likely to be encountered in each section. Columns 3, 4 and 5 show the length, cross-sectional perimeter and cross-sectional area of the passages concerned. Column 6 gives the calculated N.T.P. velocity in that particular section. Column 7 shows the velocity head p_v calculated from equation (17). Columns 8–10 give the f values ; reasons for the selection of these values are given in the Notes to Tables. Finally, the resulting value of p_v is tabulated in Column 11. p_s is separately tabulated below the p_v table.

It will be noted that in several parts of the furnace values of L, C and f_2 are not tabulated. This is because L is so small that f_2 is negligible and knowledge of these quantities is not required.

Table 57 is the draught balance for the waste gases. Since the dimensions L, C and A are already listed in Tables 55 and 56 these columns are not repeated. In calculating the resistance for the outgoing gases a slight complication is introduced by division of the waste gases between the gas and air chequers.

* The f_2 value for chequers is included in this.
† This quantity of air is less than the theoretical for complete combustion, but is the required inflow so that after allowing for infiltration, a slightly reducing atmosphere is maintained in the working chamber.

This is surmounted by assuming an arbitrary partition, say 30 per cent. through the gas chequer and 70 per cent. through the air chequer. The resistances will then usually work out to a different value for each path. It is known that the total pressure change through each path must be the same, and that p_v depends on the square of the gas velocity while the buoyancy effect, p_s, does not change with velocity. Hence, if x is the true partition, and p_v the eddy losses calculated on a basis of 30 per cent. through the gas chequers, we have :—

$$\left(\frac{x}{0{\cdot}3}\right)^2 p_v + p_s = \left(\frac{1-x}{0{\cdot}7}\right)^2 p_v + p_s \quad . \quad . \quad . \quad . \quad . \quad (22)$$

Thus x is determined and either term above will give the true pressure change to be expected over the divided path.

In the present example, Table 57 shows that p_v for the waste gases passing through the gas system is ·9046 inches w.g., and for those passing through the air system is 1·5952 inches w.g., assuming a 30 per cent. partition through the gas system. p_s is ·2242 and ·2547 for the gas and air systems respectively. Hence :

$$\left(\frac{x}{0{\cdot}3}\right)^2 \times {\cdot}9046 + {\cdot}2242 = \left(\frac{1-x}{0{\cdot}7}\right)^2 \times 1{\cdot}5952 + {\cdot}2547, \text{ and } x = {\cdot}365.$$

Hence, under the specified conditions, about 36·5 per cent. of the outgoing waste gases will pass through the gas regenerative system and the draught required in the stack flue is given by putting x = ·365 in either term of the above equation, the result being 1·57 inches w.g.

Hence calculation of the pressure balances shows that :—

(1) The gas pressure required at the reversing valve is 1·22 inches (from Table 55).
(2) An air fan to give 0·17 inches w.g. at the entrance to the air reversing valve will be necessary (from Table 56).
(3) A stack to give 1·57 inches draught in the stack flue is required.

Table 56 shows that most of the resistance to the ingoing air is caused by the uptakes. If it is desired to avoid the installation of an air fan, and to run the furnace on " natural air " p_s for the ingoing air must be greater than p_v, therefore the size of the uptakes should be increased until p_v is reduced by the required amount.

NOTES APPLICABLE TO f VALUES IN TABLES 55–57*

1 Perimeter of each passage.
2 Total area of passages.
3 Two-thirds of the theoretical free area. In a less favourable layout, such as an open hearth furnace provided with slag-pockets, half the theoretical free area should be used.
4 Velocity at entry. This gradually decreases along the passage as the gas enters the chequers.
5 Entry from gas box, $f_1 = 2{\cdot}0$; 90° bend at bottom, $f_1 = 1{\cdot}5$; enlargement to flue, $f_1 = 0{\cdot}5$. Total 4·0.
6 Two slightly rounded 45° bends at 0·3 each (instead of 0·5), $f_1 = 0{\cdot}6$.
7 For contraction from 10·75 to 7·5 square feet standard curve shows $f_1 = 0{\cdot}1$, 3 times 0·1 = 0·3. Division into 3 passages, $f_1 = 2{\cdot}0$. Total 2·3.
8 Entry to chequer, $f_1 = 2{\cdot}0$; 25 courses at 0·3, 7·5 ; emergence from chequer to collecting space at top, 1·0. Total 10·5. The f_1 value for chequers is included in the assumed 0·3 per course of brick.

* The reference numbers in brackets against certain values in Tables 55 to 57 refer to items 1 to 24 of these notes.

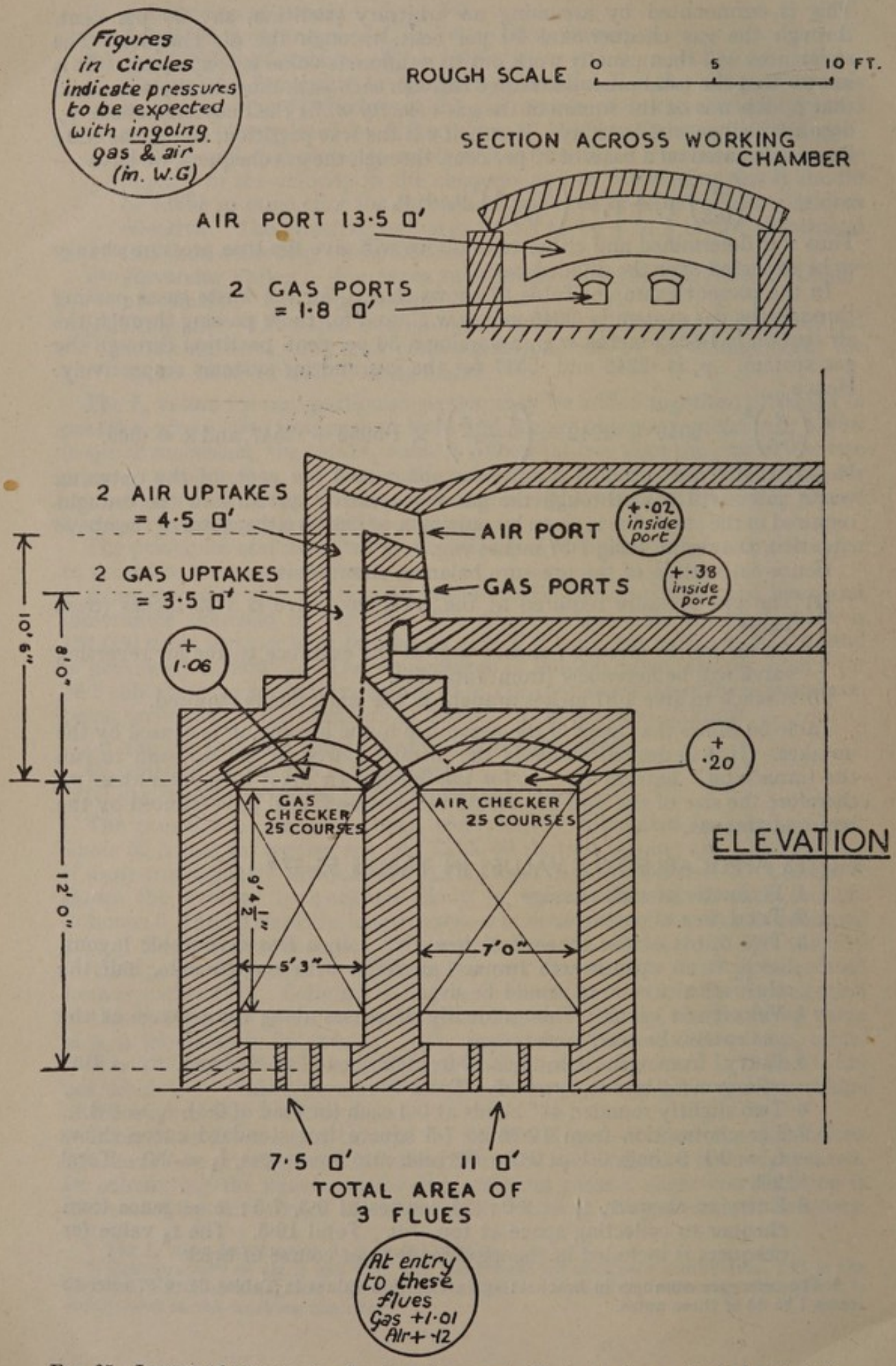

FIG. 65. Layout of regenerative heating furnace, to illustrate the method of calculating the pressure balance.

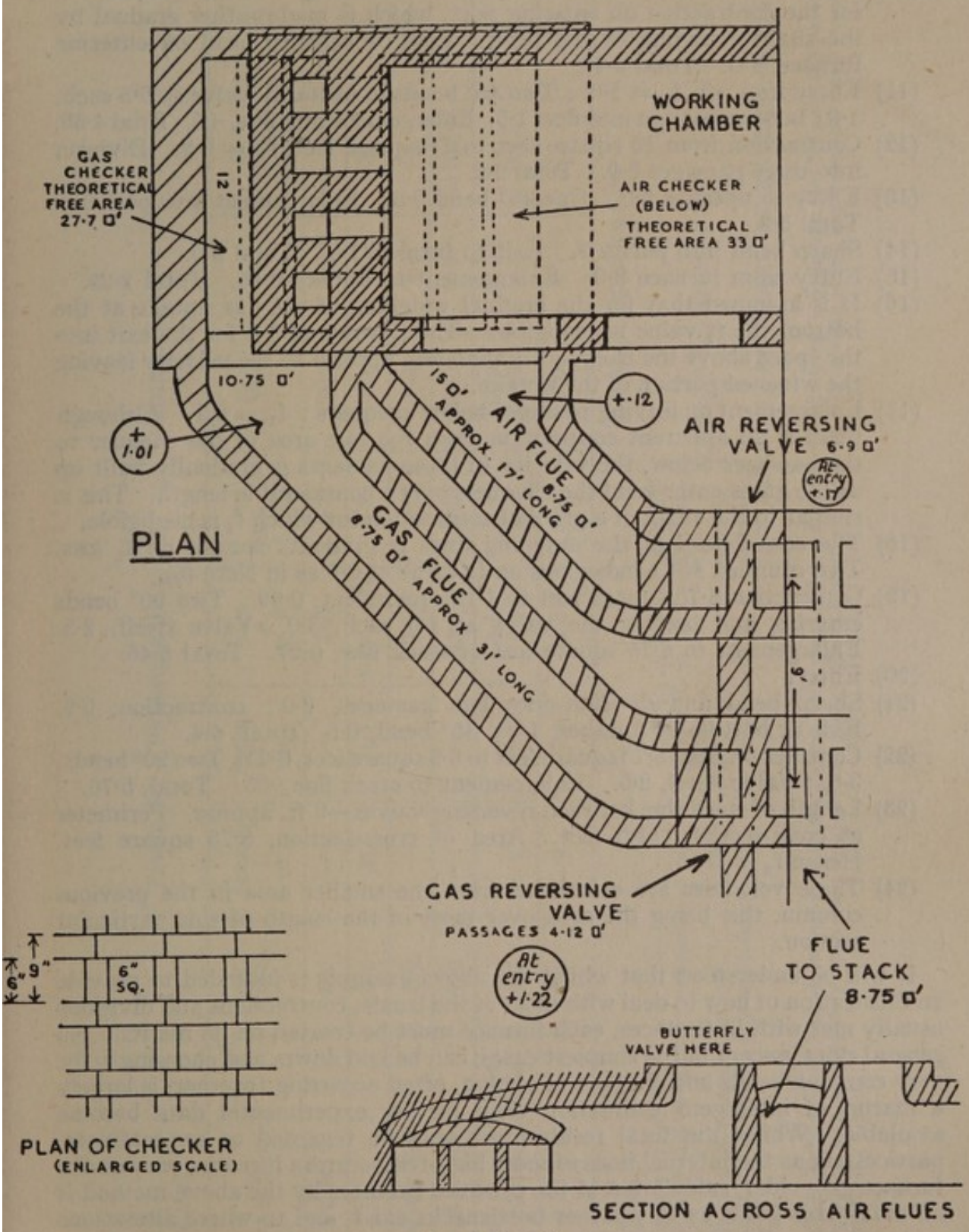
WORKING CHAMBER
GAS CHECKER THEORETICAL FREE AREA 27·7 □'
12'
AIR CHECKER (BELOW) THEORETICAL FREE AREA 33 □'
10·75 □'
15 □' AIR FLUE APPROX. 17' LONG 8·75 □'
+·12
+ 1·01
AIR REVERSING VALVE 6·9 □'
At entry +·17
PLAN
GAS FLUE 8·75 □' APPROX 31' LONG
6'
GAS REVERSING VALVE
PASSAGES 4·12 □'
At entry +1·22
FLUE TO STACK 8·75 □'
BUTTERFLY VALVE HERE
9"
6"
6" SQ.
PLAN OF CHECKER (ENLARGED SCALE)
SECTION ACROSS AIR FLUES

FIG. 65 (cont.).

(9) The f_1 value of 2·0 for the entry to the uptakes is applied to the velocity in the enlarged entry section. The gradual narrowing afterwards does not occasion any further f_1 value, but the f_2 value is applied to the velocity in the 3·5 square feet section which covers most of the length.

(10) Somewhat rounded bend into port—assumed $f_1 = 1·5$. Assume 0·2 for the contraction on entering port, which is made rather gradual by the slight rounding of the bend. Loss of velocity head on entering furnace, 1·0. Total 2·7.

(11) Entry from air, $f_1 = 2·0$. Two 45° bends in butterfly valve at 0·5 each, 1·0; bend at bottom into flue, 1·5. Enlargement into flue, ·05. Total 4·55.

(12) Contraction from 15 square feet to 11 square feet, $f_1 = 0·2$. Division into three passages 2·0. Total 2·2.

(13) Entry to uptakes 2·0. One 45° bend 0·5. Enlargement into port 0·8. Total 3·3.

(14) Sharp bend into port 2·0. Exit to furnace 1·0. Total 3·0.

(15) Entry from furnace 2·0. Enlargement to uptake 0·25. Total 2·25.

(16) It is assumed that for the gradual widening of the gas uptake at the bottom the f_1 value is negligible. The f_1 value of 1·0 for the exit into the space above the chequers is therefore applied to the velocity leaving the widened portion of the uptake.

(17) Enlargement on leaving passages below chequers: $f_1 = 0·1$. Although there is an apparent contraction from the free area of the chequer to the passages below, the velocity in these passages is gradually built up as the gases enter from the chequers over a considerable length. This is similar to the effect of a gradual contraction for which f_1 is negligible.

(18) The contraction in the outgoing flues is gradual, causing no f_1 loss. Two rounded 45° bends gives an f_1 value of 0·6 as in Note (6).

(19) Contraction 8·75 square feet to 4·12 square feet, 0·69. Two 90° bends entering and leaving the valve at 1·5 each, 3·0. Valve itself, 2·5. Enlargement to 8·75 square feet of stack flue, 0·27. Total 6·46.

(20) Entry.

(21) Sharp bend and division into two passages, 2·0; contraction, 0·9. Exit at bottom of uptakes, 1·0; 45° bend, 0·5 Total. 4·4.

(22) Contraction from 8·75 square feet to 6·9 square feet, 0·21. Two 90° bends, 3·0. Valve itself, 2·5. Enlargement to stack flue, ·05. Total, 5·76.

(23) Length of stack flue between reversing valves—9 ft. approx. Perimeter of cross-section, 11·5 feet. Area of cross-section, 8·75 square feet. Hence $f_2 = 0·15$.

(24) These velocities are calculated from the smaller area in the previous column, this being the area over most of the length of this particular section.

It will be understood that whilst the above example is intended to provide an illustration of how to deal with most of the bends, contractions and divisions usually met with in furnaces, each furnace must be treated on its merits. No general rules, except in the simplest cases, can be laid down, and choosing f_1 for very complex bends and changes of section, often occurring together, is largely a matter of intelligent estimation until further experimental data become available. Whilst the total result must only be regarded as approximate, particularly as the internal lines of most high-temperature furnaces vary as the furnace gets older, calculation of the pressure balances by the above method is an invaluable guide as to whether bottlenecks exist, and to where alterations are required. Where quantities such as f_1 are uncertain it is preferable to err on the high side as it is much easier to control flow in a generously designed furnace by dampers or chokes than to enlarge passages after they have been built too small.

TABLE 55. PRESSURE BALANCE FOR INGOING GAS

Calculation of p_v

Part of furnace	Av. Temp. (° C.)	Length L (ft.)	Perim. C (ft.)	Area A sq. ft.	v_o ft./sec. at N.T.P.	p_o in. W.G.	f_1	f_2 from equn. (21)	$f_1 + f_2$	p_v in. W.G.
Reversing valve	500	—	—	4·12	9·22	·0504	4·0 (5)	—	4·0	·2016
Flue	500	31	11·5	8·75–10·75	4·34 (24)	·0111	0·6 (6)	·51	1·11	·0123
Under chequers (3 flues) ..	500	—	—	7·5 (2)	5·07 (4)	·0152	2·3 (7)	—	2·3	·0350
Chequers (25 courses) ..	650	—	—	18·5 (3)	2·05	·0030	10·5 (8)	See Note 8	10·5	·0315
Uptakes (2) Lower mouths	800	8	—	5·6	6·79	·0379	2·0 (9)	—	2·0 (9)	·0758
Main portion ..			5·33	3·5 (2)	10·8	·0886	—	·30 (9)	·30	·0266
Ports (2)	800	3	3·6 (1)	1·8 (2)	21·6	·3650	2·7 (10)	·16	2·86	1·0439

TOTAL p_v (opposing flow) .. .. .. 1·4267

Calculation of p_s

Buoyancy in chequers	12 ft. @ 650° =	·1286 in.
Net buoyancy in uptakes and ports	8 ft. @ 800° =	·0900 in.
Total buoyancy assisting flow		·2186 in.
Less furnace pressure (opposing flow)		·0100 in.

TOTAL p_s (assisting flow) .. .. .. ·2086

NET PRESSURE REQUIRED AT REVERSING VALVE 1·2181 in W.G.

NOTE.—In Tables 55–57 and elsewhere in this calculation, four places of decimals have been used. Although the 3rd and 4th places of decimals have no practical significance they have been retained as making the calculations easier to follow than if the numbers had been rounded off to the nearest significant figure.

TABLE 56. PRESSURE BALANCE FOR INGOING AIR

Calculation of p_v

Part of furnace	Av. Temp. (° C.)	Length L (ft.)	Perim. C (ft.)	Area A (sq. ft.)	v_o ft./sec. at N.T.P.	p_o in. W.G.	f_1	f_2 from equn. (21)	$f_1 + f_2$	p_v in. W.G.
Reversing valve	50	—	—	6·9	5·64	·0090	4·55 (11)	—	4·55	·0410
Flue	100	17	11·5	8·75–15	4·45 (24)	·0065	0·6 (	·22	0·82	·0053
Under chequer	100	—	—	11	3·54	·0041	2·2 (12)	—	2·2	·0090
Chequers (25 courses) ..	500	—	—	22 (3)	1·77	·0021	10·5 (8)	See Note 8	10·5	·0221
Uptakes (2)	900	13	6 (1)	4·5 (2)	8·64	·0771	3·3 (13)	·35	3·65	·2814
Port	900	3	21	13·5	2·88	·0086	3·0 (14)	·04	3·04	·0261

TOTAL p_v (opposing flow) .. .. .. ·3849

Calculation of p_s

Buoyancy in chequers .. .. 12 ft. @ 500° = ·1106 in.
Net buoyancy in uptakes and port .. .. 10 ft. 6 in. @ 900° = ·1164 in.

Total buoyancy assisting flow .. .. ·2270 in.
Less furnace pressure (opposing flow) .. .. ·0100 in.

TOTAL p_s (assisting flow) .. .. .. ·2170

NET PRESSURE REQUIRED AT REVERSING VALVE ·1679 in. W.G.

(See footnote to Table 55.)

TABLE 57. DRAUGHT BALANCE FOR OUTGOING WASTE GAS

Volume of waste gas (including infiltrated air) is assumed to be 293,000 cubic feet per hour (measured at N.T.P.), 30 per cent. of which passes through the gas regenerative system, and 70 per cent. of which passes through the air system. Density of waste gases may be taken as the same as air.

Calculation of p_v

Part of furnace	Temp. °C.	Gas System						Air System					
		v_o ft./sec. at N.T.P.	p_o in. W.G.	f_1	f_2	$f_1 + f_2$	p_v in. W.G.	v_o ft./sec. at N.T.P.	p_o in. W.G.	f_1	f_2	$f_1 + f_2$	p_v in. W.G.
Ports	1,400	13·6	·2710	2·25 (15)	·16	2·41	·6531	4·22	·0263	2·0 (20)	·04	2·04	·0537
Uptakes .. .. (Lower mouths of gas uptake treated separately).	1,400	6·97 4·37	·0716 ·0281	— 1·0 (16)	0·3 (16) —	0·3 1·0	·0215 ·0281	12·7	·2373	4·4 (21)	·35	4·75	1·1272
Chequers	1,000	1·32	·0019	10·5 (8)	See Note 8	10·5	·0200	2·59	·0075	10·5 (8)	See Note 8	10·5	·0788
Under chequers ..	600	3·25	·0081	0·1 (17)	—	0·1	·0008	5·18	·0206	0·1 (17)	—	0·1	·0021
Flues	600	2·79	·0060	0·6 (18)	·51	1·11	·0067	6·52	·0326	0·6 (18)	·22	0·82	·0267
Reversing valves ..	600	5·92	·0270	6·46 (19)	—	6·46	·1744	8·25	·0524	5·76 (22)	—	5·76	·3018
Stack flue between air and gas rev. valves.	600	—	—	—	—	—	—	6·52	·0326	—	·15 (23)	·15	·0049

TOTAL p_v (opposing flow) for gas system .. .. ·9046 .. .. for air system .. .. 1·5952

Calculation of p_s

Gas system		Air system	
Buoyancy in uptakes, etc. 8 ft. @ 1,400° =	·0974 in.	10 ft. 6 in. @ 1,400° =	·1279 in.
Buoyancy in chequers 12 ft. @ 1,000° =	·1368 in.	12 ft. @ 1,000° =	·1368 in.
Total buoyancy opposing flow	·2342 in.	Total buoyancy opposing flow	·2647 in.
Less furnace pressure (assisting flow)	·0100 in.	Less furnace pressure	·0100 in.
TOTAL p_s	·2242 in.	TOTAL p_s	·2547 in.

ILLUSTRATIVE EXAMPLES (other than for furnaces)

In order to illustrate the use of the principles and equations here discussed, some numerical examples are appended :—

(1) Clean producer gas (density at 60° F. and 30 inches bar is 0·07 lb. per cubic foot) is flowing through a 20-inch diameter main, in which is an orifice plate having $\alpha = 0{\cdot}61$ and 4 inches diameter. The pressure before the orifice plate 20 inches upstream is 8 inches w.g. and the differential pressure between this point and one 8 inches past the orifice is 2 inches w.g. If the barometer is at 29 inches and the gas temperature is 150° F., what is the quantity of gas passing per hour at 60° F. and 30 inches bar.

Solution :—

The conditions are similar to those of Fig. 59, and equation (12) can be used viz.,

$$v = \alpha\sqrt{2gh}.$$

The density of the producer gas at 150° F. and 29·6 inches bar. (i.e. atmosphere + 8 inches w.g.)

$$= 0{\cdot}07 \times \frac{29{\cdot}6}{30} \times \frac{460 + 60}{460 + 150} = 0{\cdot}059 \text{ lb./cu. ft.}$$

Since 1 inch w.g. = 5·2 lb./sq. ft., the height of a column of producer gas equivalent to 1 inch w.g. is

$$\frac{5{\cdot}2}{0{\cdot}059} = 88 \text{ feet.}$$

Thus, h, the differential head of 2 inches w.g. expressed as feet of gas

$$= 2 \times 88 = 176 \text{ feet}$$

$$v = 0{\cdot}61\sqrt{2 \times 32{\cdot}2 \times 176} = 65 \text{ ft./sec. average velocity through the 4-inch orifice.}$$

This is now to be converted into cubic feet per hour,

$$\text{and} = 65 \times \pi \times \left(\frac{4}{24}\right)^2 \times 3{,}600 = 20{,}400 \text{ cu. ft./hr. at 29·6 inches and 150° F.,}$$

and at 30 inches and 60° F. this becomes

$$20{,}400 \times \frac{29{\cdot}6}{30} \times \frac{460 + 60}{460 + 150} = 17{,}170 \text{ cu. ft./hr.}$$

(2) (*a*) A mineral oil of specific gravity 0·9 is to be pumped from a storage tank at ground level to discharge freely into an overhead tank 200 feet away and 30 feet above ground level. If the pipe is 3 inches i/d, and it has to take a maximum of 2,000 gallons of oil per hour, what pressure is required at the outlet of the pump at the storage tank (neglecting bends) ?

(*b*) A measuring device is installed in the main consisting of a 2-inch orifice plate in a 10-inch diameter chamber. A vertical U-tube containing mercury (sp. gr. 13·6) and oil shows 0·7 inch difference in levels. How much oil was passing at the time ?

(*a*) The flow of the oil through the pipes can be calculated by formula (5). Reynolds number must first be calculated.

$$\text{Re} = \rho\frac{Dv}{\eta} = \frac{Dv}{\nu}$$

$D = 0{\cdot}25$ foot ; $\nu = 0{\cdot}05 \times 0{\cdot}0010764$ (Table 49) ; and since 1 cubic foot = 6·229 gallons—

$$v = \frac{2{,}000}{6{\cdot}229} \times \frac{1}{3{,}600} \times \frac{1}{0{\cdot}125^2\pi} \text{ ft./sec.}$$

$$= 1{\cdot}82 \text{ ft./sec.}$$

$$\rho = 56 \text{ lb./cu. ft.}$$

$$\text{Re} = \frac{0{\cdot}25 \times 1{\cdot}82}{0{\cdot}05 \times 0{\cdot}0010764} = 3{,}453$$

whence, from Fig. 58, f = 0·011.

From equation (5)

$$\rho = 0{\cdot}011 \times \frac{4 \times 230}{0{\cdot}25} \times \frac{56 \times 1{\cdot}82^2}{64{\cdot}4}$$

$$= 116{\cdot}6 \text{ lb./sq. ft.}$$

To this must be added the head of oil in the 30 feet of vertical pipe, namely, 56 × 30 lb./sq. ft. = 1,680 lb./sq. ft.

The total initial pressure is thus—

$$1{,}680 + 116{\cdot}6 = 1{,}796{\cdot}6 \text{ lb./sq. ft.} = 12{\cdot}45 \text{ lb./sq. in.}$$

(*b*) The conditions are similar to Fig. 59 and equation (12) can be applied.

1 cubic foot of mercury weighs 62·3 × 13·6 lb. = 847·3 lb.
1 cubic foot of oil weighs 56·0 lb.

As the differential pressure is measured by a U-tube containing oil over mercury, the measured difference of 0·7 inch mercury in the level of the U-tube must be converted into feet of oil as follows :—

$$h = 0{\cdot}7 \times \frac{(847{\cdot}3 - 56{\cdot}0)}{12 \times 56{\cdot}0} = 0{\cdot}824 \text{ feet of oil.}$$

This correction means that the vertical height of 0·7 inch between the mercury columns is filled with oil and the weight of this oil (unlike the weight of a similar height of air) is appreciable and must be subtracted from the mercury gauge reading.

$$v = 0{\cdot}61\sqrt{2 \times 32{\cdot}2 \times 0{\cdot}824}$$

$$v = 4{\cdot}45 \text{ ft./sec. through the orifice}$$

Area of 2-inch orifice = $\pi/144$ square feet.

$$\text{Volume of oil passing} = \frac{4{\cdot}45 \times \pi \times 3{,}600}{144} \text{ cu. ft./hr.}$$

$$= 349 \text{ cu. ft./hr.,}$$

or, (since 1 cubic foot = 6·229 gallons) 2,200 gallons per hour.

(3) A Pitot tube is inserted in a 12-inch diameter gas main carrying blue water gas (sp. gr. 0·55, dry gas/dry air), along the centre line of the main. The differential pressure registered by a U-tube attached to the two connections and inclined to the horizontal at 10 : 1 is 0·5 inch w.g. The pressure in the gas main is 24 inches w.g., and the barometer pressure is 29 inches and the gas temperature 80° F. How much gas is the pipe carrying at 60° F. and 30 inches bar. ?

$$\text{Total pressure in the main} = 29 + \frac{24}{13{\cdot}6}$$

$$= 30{\cdot}76 \text{ inches mercury gauge.}$$

The true pressure on a 10 : 1 inclination is 0·05 inch w.g.

The density of the saturated gas under the existing conditions of temperature and pressure can be calculated as follows :—

(*a*) Saturation pressure at 80° F. = 1.035 inch Hg.

(*b*) Volume of saturated gas at 80° F. and 30·76 inches Hg resulting from 1 cubic foot of dry gas at 32° F. and 29·92 inches Hg (N.T.P.)

$$= \frac{540}{492} \times \frac{29{\cdot}92}{30{\cdot}76 - 1{\cdot}035} = 1{\cdot}105 \text{ cu. ft.}$$

(*c*) Under the above conditions, the weight of water resulting from 1 cubic foot of dry gas at N.T.P. will be :—

$$0{\cdot}0502 \times \frac{1{\cdot}035}{30{\cdot}76 - 1{\cdot}035} = 0{\cdot}00175 \text{ lb.}$$

(0·0502 is the weight of water vapour per cubic foot at N.T.P.).

(*d*) The density of the saturated gas under actual flow conditions will be as follows :—

Density of dry gas at N.T.P. $= 0{\cdot}55 \times 0{\cdot}0807 = 0{\cdot}044385$ lb./cu. ft.
Weight of water vapour $= 0{\cdot}001750$,,

Weight of 1·105 cubic foot of saturated gas $= 0{\cdot}046135$,,
,, ,, 1·0 ,, ,, ,, ,, $= 0{\cdot}0418$,,

Head of gas corresponding to 0·05 inch w.g. $= \frac{5{\cdot}2}{0{\cdot}0418} \times 0{\cdot}05$ foot
$= 6{\cdot}22$ feet.

Equation (12a) may now be applied.

$$v = \sqrt{2 \times 32{\cdot}2 \times 6{\cdot}22}$$
$$= 19{\cdot}6 \text{ ft./sec.}$$

This is the velocity at the centre of the main ; from Fig. 62,

since $\frac{\rho D v_{max.}}{\eta} = \frac{0{\cdot}0418 \times 1{\cdot}0 \times 19{\cdot}6}{0{\cdot}000162 \times 0{\cdot}0672} = 75{,}260$

the ratio $v_{average} : v_{max.} = 0{\cdot}81$,
therefore average velocity $= 0{\cdot}81 \times 19{\cdot}6$ ft./sec.
in terms of volume $= 0{\cdot}81 \times 19{\cdot}6 \times \pi \times 0{\cdot}5^2 \times 3{,}600$ cu. ft./hr.
$= 44{,}870$ cu. ft./hr. at 80° F. and 30·76 inches, sat.
$= 43{,}520$ cu. ft./hr. at 60° F. and 30 inches bar, sat.

(4) What size of gas pipe is required to transmit 600 cubic feet of town gas (sp. gr. = 0·45) per hour to a point 100 feet from the gas meter where the pressure at the meter is 4 inches w.g. and the pressure required at the point of use is $3\frac{1}{2}$ inches w.g. ?

If Lacey's formula (equation (8)) is used, a reasonably probable value must be selected for f since Re cannot be calculated. When the size of pipe has been thus calculated, the value of Re can be calculated and the value of f ascertained more accurately ; it may then be necessary to repeat the calculation using this value of f.

It is more convenient for these low pressures to use Pole's formula (equation (9)).

$$Q = 1{,}350 \sqrt{\frac{d^5 p}{sL}}$$

$$600 = 1{,}350 \sqrt{\frac{d^5(4-3\frac{1}{2})}{0{\cdot}45 \times 33{\cdot}3}}$$

when d = 1·43 inches diameter,
hence, a $1\frac{1}{2}$ inch diameter pipe would be used.

(5) Coke oven gas (sp. gr. 0·4) supplied by a booster giving a maximum pressure of 40 lb. per square inch is to be supplied over a distance of two miles through a main 10 inches diameter, the terminal pressure being 1 lb. per square inch gauge. What quantity of gas can be transmitted ?

It is necessary for the preliminary calculation to select an arbitrary value for f, say, 0·010.

$$Q = 2{,}675 \sqrt{\frac{10^5}{0{\cdot}010 \times 0{\cdot}4 \times 10{,}560} \times \frac{54{\cdot}7^2 - 15{\cdot}7^2}{14{\cdot}7^2}}$$

whence Q = 463,900 cu. ft./hr. measured at 60° F. and 30 inches bar.

Under these circumstances, taking a section of the main where p = 3 × 14·7 lb./sq. in., the value of Re is calculated.

$\rho = 0{\cdot}0304$ at S.T.P. $= 0{\cdot}091$ lb./cu.ft. in the gas main.

$D = 10/12 = 0{\cdot}833$ foot

$\eta = 1{\cdot}37 \times 10^{-4} \times 0{\cdot}0672 = 9{\cdot}2 \times 10^{-6}$ f.p.s. units

$$v = \frac{463{,}900}{3{,}600} \times \frac{14{\cdot}7}{3 \times 14{\cdot}7} \times \frac{12^2}{\pi 5^2} = 78{\cdot}5 \text{ ft./sec.}$$

whence $Re = \dfrac{0{\cdot}091 \times 0{\cdot}833 \times 78{\cdot}5}{9{\cdot}2 \times 10^{-6}} = 646{,}800$

whence (from Fig. 58), f = 0·0031.

Since this is markedly different from the assumed value of 0·010, it is necessary to recalculate the value of Q as above, using f = 0·003.

When this is done, Q = 847,000 cu. ft./hr. at 60° F. and 30 inches bar.

Calculation of Re and reference to Fig. 58 shows that Re = 1,179,000 and f = 0·003, whence the calculation is now found to be in order, since the correct value has been taken for f.

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CHAPTER X

THE MEASUREMENT OF SOLID FUEL, STEAM, WATER, GAS AND AIR

Solid fuel ; methods involving measurement and direct weighing—Measurement of fluid flow by orifice plate—Design data—Orifice location—Pressure losses—Formulae—Pulsating flow—Correction factors—Measurement of water by V-notch—Examples of calculation.

THE principles of the measurement of fluids have been discussed in Chapter IX, with special reference to the orifice plate, the Venturi meter and the Pitot tube. Equations (11), (12) and (12a) have been deduced as applying to these forms of measuring device. The practical application of the principles there set out is dealt with in this chapter. There are, of course, many commercial forms of measuring instrument, but these are not generally mentioned here, it being rather the object to describe basic methods.

Solid fuels come under a different category and accurate work necessitates some form of weighing machine.

SOLID FUEL

Accurate determination of the quantity of fuel used is essential if the performance of a plant is to be known. The best method is to weigh the fuel on a weighing machine.

INDIRECT MEASUREMENT

Where the coal allocated to individual plants cannot be weighed, a reasonably accurate figure can be obtained by volume measurement. If this is based on the volume of a coal hopper or bunker, it must be remembered that the volume weight may depend on the height from which it is dropped. Thus when a certain slack, 0–$\frac{3}{8}$ inch in size and containing 4$\frac{1}{2}$ per cent. of moisture, was dropped from a height its volume weight was found to be as follows :—

Distance dropped	Wt. per cu. ft.—lb.
1$\frac{1}{2}$ feet	47·57
2 ,, ...	47·61
4 ,, ...	49·00
6 ,, ...	50·48
8 ,, ...	51·48
10 ,, ...	52·35

The weight per cubic foot also depends on the size of the coal. Thus J. B. Deakin and W. T. K. Braunholtz ("Year Book," Coke Oven Managers' Association, 1930, 266), using a South Wales coking coal with 26 per cent. V.M. found :—

Size of coal	Wt. per cu. ft. lb.—lb.
2 –1 inch	41·87
1 –$\frac{1}{2}$,, ...	42·78
$\frac{1}{2}$ –$\frac{1}{4}$,, ...	40·46
$\frac{1}{4}$ –$\frac{1}{8}$,, ...	39·52

The dependence of volume weight on moisture content may also be considerable as is evident from Fig. 66, which refers to a slack coal. The absolute specific gravity of coals (i.e. specific gravity of the solid lump) may vary considerably and the volume weight will also depend upon the class of coal used. It will also depend on the way in which the coal " packs." A coal of mean sp. gr. 1·4 weighs some 87 lb. per cubic foot, whereas the nuts from such a coal from the figures given above weigh some 42 lb. per cubic foot. The air

spaces in the nuts thus occupy 52 per cent. If the coal consists of various sizes so that the fines can pack in the interstices between the larger pieces, the volume weight may increase considerably.

The volume weight of coke will depend not only upon its size, but upon the method of manufacture. The following figures may be quoted purely as an indication of the extent of these differences :—

	Bulk density—lb./cu. ft.	
	Size $\frac{3}{4}$ in.–$1\frac{1}{2}$ ins.	Rough
Metallurgical (oven) coke ..	26–30	29–32
Horizontal retort coke	23–26	25–28
Vertical retort coke	21–22	23

These facts are given to indicate that when the quantity of coal or coke used as fuel is ascertained by volume measurements instead of by direct weighing,

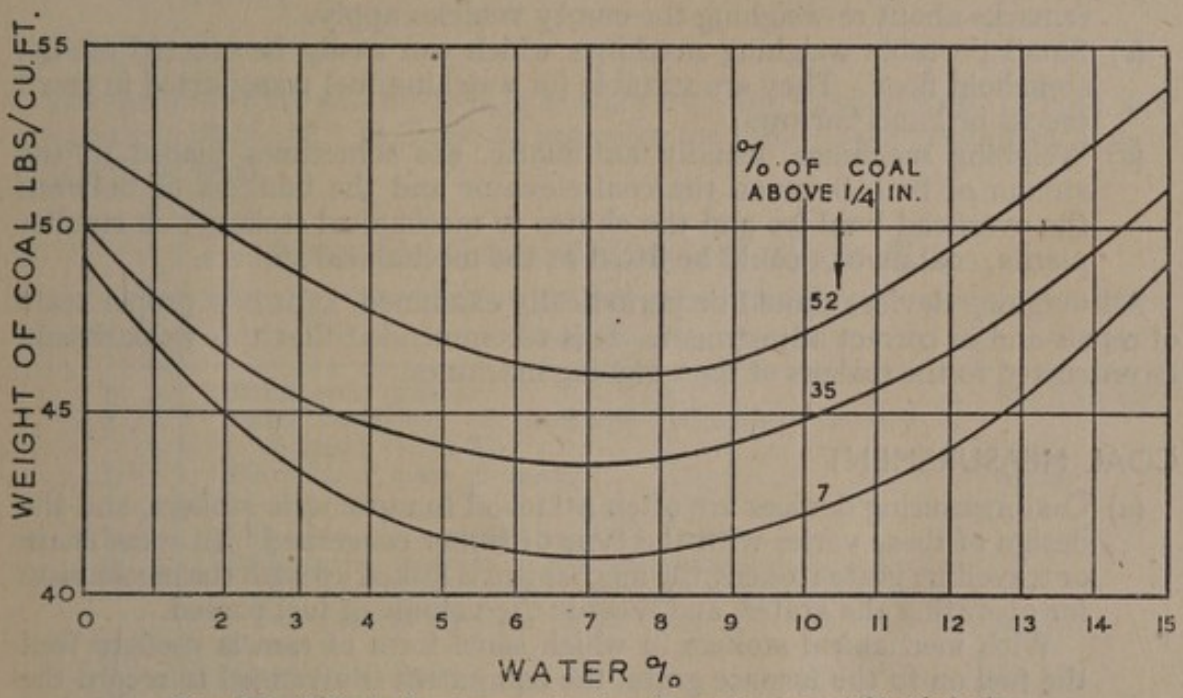

FIG. 66. Effect of water content on the volume weight of slack.

care must be taken to determine accurately the weight of fuel per cubic foot using the particular fuel supplied and the right conditions of packing.

One method of measuring fuel for small plants not provided with a weigh-bridge is to provide a gauge frame. This would not have a bottom, the floor serving instead. The frame is filled with fuel and levelled off by running a rod along the top edge, any surplus fuel being returned to the store. The frame should be of stout construction and provided with handles. If the amount of the fuel in use required to fill the frame is accurately weighed (preferably by more than one filling), a good indication of the total fuel used can be secured.

In small boiler houses, where the fuel is manually handled, a small weighing machine capable of weighing up to 4 cwt., could with advantage be installed. Most steel barrows used at these plants, either for moving coal or removing ashes, hold about 2 cwt. The weighing machine should be placed on a level part of the floor, in a convenient position. A small ramp made of heavy planks should be fitted at each side of the weighing machine platform to facilitate moving the loaded barrow over it. The arm of the weighing machine should be adjusted to balance exactly the weight of the barrow plus the weight of the fuel in it. Each time the full barrow passes over the platform the weight of fuel should be adjusted so that the weighing machine is balanced.

Whatever method of weighing or measuring is employed, it is desirable that the records of the weight of fuel used should be checked against the suppliers' delivery weights, preferably weekly. Variations in the quantity of fuel in stock should be taken into consideration when the comparisons are made.

DIRECT WEIGHING

Direct weighing equipment takes many forms, and those most usually found are :—

(*a*) A weighbridge placed in the railway tracks for weighing full coal waggons : where it is possible to weigh the same waggons, when empty, on the same weighbridge, the exact weight of fuel used can be ascertained.

(*b*) Flush weighbridge for weighing fuel delivered in motor or horse-drawn lorries or carts. These machines give accurate weights and the previous remarks about re-weighing the empty vehicles apply.

(*c*) Small portable weighing machines which can easily be erected on the stokehold floor. They are suitable for weighing fuel transported in small trucks or hand barrows.

(*d*) Weighing machines, usually automatic, are sometimes placed in the stream of fuel, between the coal elevator and the bunkers or between the overhead bunkers and the chutes to mechanical stokers ; in smaller plants, coal meters could be fitted at the mechanical stokers.

All weighing devices should be periodically examined, kept in a proper state of repair and in correct adjustment. It is recommended that this work should be entrusted to the makers of the weighing machines.

COAL MEASUREMENT

(*a*) Coal measuring devices are often attached to automatic stokers, and the design of these varies with the type of stoker concerned. In some chain or travelling grate stokers, the mechanism is linked up with the mechanism for operating the grates, and records the volume of fuel passed.

With mechanical stokers in which some form of ram is used to feed the fuel on to the furnace grate, the equipment is arranged to record the quantity of fuel passed, from the number of strokes of the ram. Devices of this kind should be frequently calibrated, especially in plants where the size and type of fuel used often changes.

(*b*) An appliance for measuring the flow of coal down a chute is sometimes used. This consists of an endless chain, one strand of which runs in the coal in the chute. The movement of the chain rotates a spindle which operates a counter from the readings of which the quantity of coal passed can be computed.

(*c*) A measuring drum attached to the outlets from overhead bunkers which measures the quantity of coal flowing to a chute.

The drum is mounted on trunnions, and when full rotates, thus cutting off the flow of the coal to the bunker and discharging its contents into the chute. The number of times the drum is rotated is recorded by suitable mechanism, and if the weight of fuel the drums holds when full is known, the total weight of fuel passed can easily be checked.

THE MEASUREMENT OF FLUID FLOW IN PIPES BY MEANS OF DIFFERENTIAL PRESSURE DEVICES

The metering of fluid flow in a pipe line by measuring the pressure drop caused by the insertion of a throttling device (Venturi tube, nozzle or orifice

plate) has proved to be one of the most reliable methods of determining the weight or volume of any fluid passing through a specified pipe line.

As indicated in Chapter IX, the method of measurement depends on the change of pressure energy caused by a fluid flowing through a reduced cross-sectional area in a pipe line.

The differential pressure devices in this chapter will be confined entirely to orifice plates. Information on Venturi tubes and nozzles can be obtained from the British Standard Specification No. 1042. which covers the theory as well as the practical application of all flow measuring devices by the differential pressure method.

The reduced area caused by the insertion of an orifice plate creates an increased velocity at this point with consequent loss of static pressure; this pressure loss when measured under specified conditions enables the ratio of flow to be determined.

SYMBOLS

The symbols that will be used in dealing with the measurement of fluid flow are given in Table 58. In Table 59 are collected some of the fundamental data that will be used.

TABLE 58. SYMBOLS USED IN CHAPTER X

Symbol	Description	Units
B	Barometric pressure	ins. mercury
C	Basic coefficient of discharge (excluding velocity of approach factor E)	ratio
D	Diameter of main (internal)	inches
E	Velocity of approach factor $\frac{1}{\sqrt{(1-m^2)}}$	—
H	Differential pressures, as read, inches mercury under water. (For ring balance meters multiply 1,273 in equation (3) and 1,018 in equation (7) by 1·039	in. Hg.
Hg	Mercury	—
P	Absolute pressure	in. Hg.
S	Static pressure	in. Hg.
T	Absolute temperature (460 + t° F.)	° F. abs.
V	Rate of flow, at metering (working) conditions of temperature and pressure	cu. ft./hr.
W	Rate of flow	lb./hr.
d	Diameter of orifice	inches.
h	Effective differential pressure, inches water gauge (12·6 H, when H = inches mercury under water = 13·6 H with ring balance meters)	in. W.G.
g	Acceleration due to gravity	ft./sec.^{2}
m	$\frac{\text{Orifice area}}{\text{Pipe area}} = \left(\frac{d}{D}\right)^2$	ratio
p_1	Absolute pressure, up-stream tapping	lb./sq. in.
p_2	Absolute pressure, down-stream tapping	lb./sq. in.
s	Specific gravity (dry gas relative to dry air)	ratio
t	Temperature (as measured)	° F.
v	Mean velocity of fluid	ft./sec.
α (alpha)	Actual coefficient of flow, including E	ratio
ϵ (epsilon)	Expansion correction factor	ratio
η (eta)	Absolute viscosity at working temperature	—
ρ (rho)	Density of fluid	lb./cu. ft
N.T.P.	Volume at normal temperature and pressure, 32° F. and 29·92 in. Hg, dry	—
S.T.P.	Volume at standard conditions of temperature and pressure, 30 in. Hg and 60° F. saturated	—

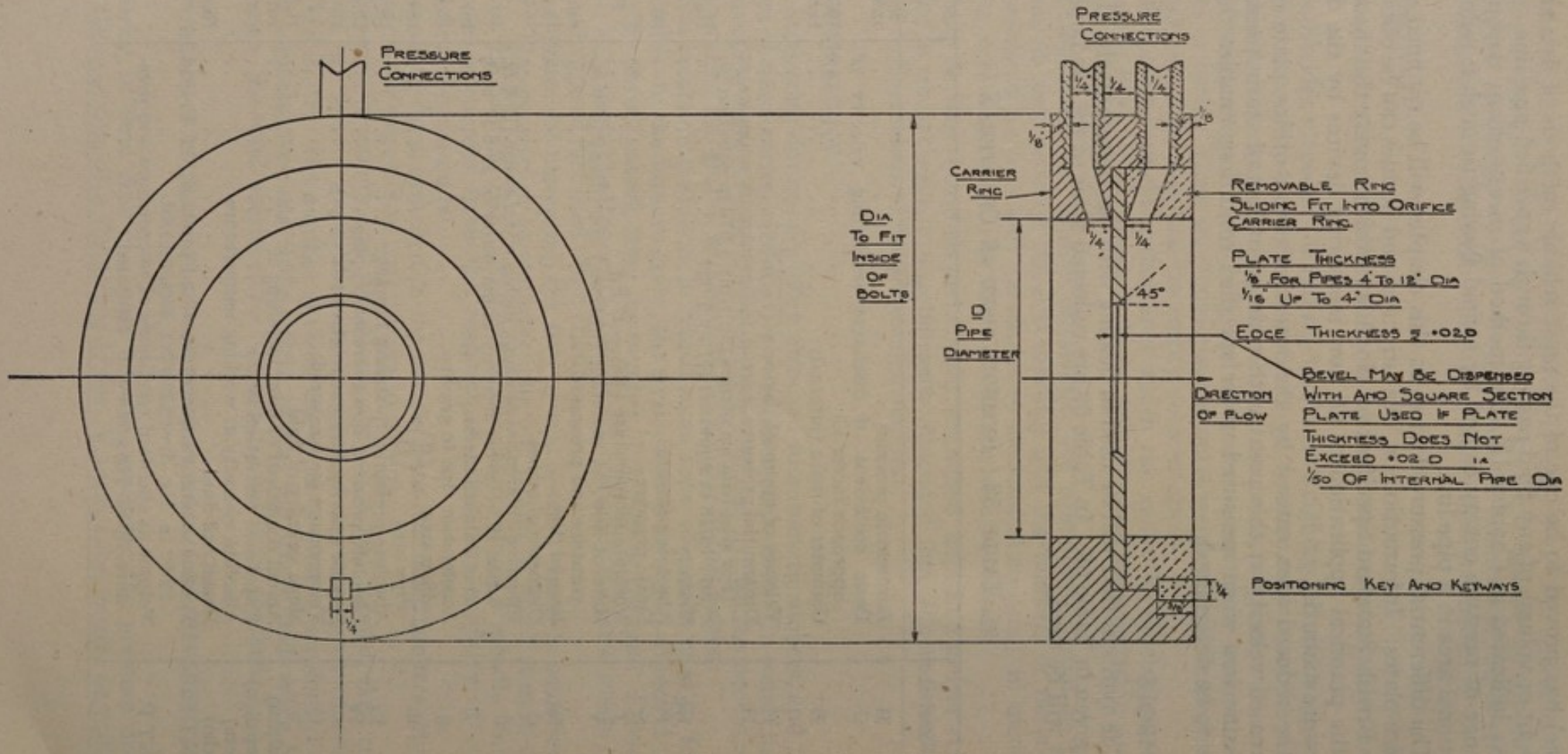

FIG. 67A. Construction of an orifice plate.

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TABLE 59. PHYSICAL CONSTANTS USED IN CHAPTER X

Density of water at 60° F.	= 62·35 lb./cu. ft.
Specific gravity of water vapour (dry air = 1)	= 0·624
g	= 32·2 ft./sec./sec.
1 cu. ft.	= 6·234 gallons.
Density of dry air at 30 in. of mercury and 60° F.	= 0·0764 lb./cu. ft.
Saturated water vapour pressure (w) at 60° F.	= 0·2563 lb./sq. in.
Absolute zero of temperature	= − 460° F.
1 in. Hg. pressure at 60° F. under air	= 0·490 lb./sq. in.
30 in. Hg. at 60° F. under air	= 14·70 lb./sq. in.
1 in. water gauge pressure at 60° F. under air	= 5·2 lb./sq. ft.
	= 0·03604 lb./sq. in.

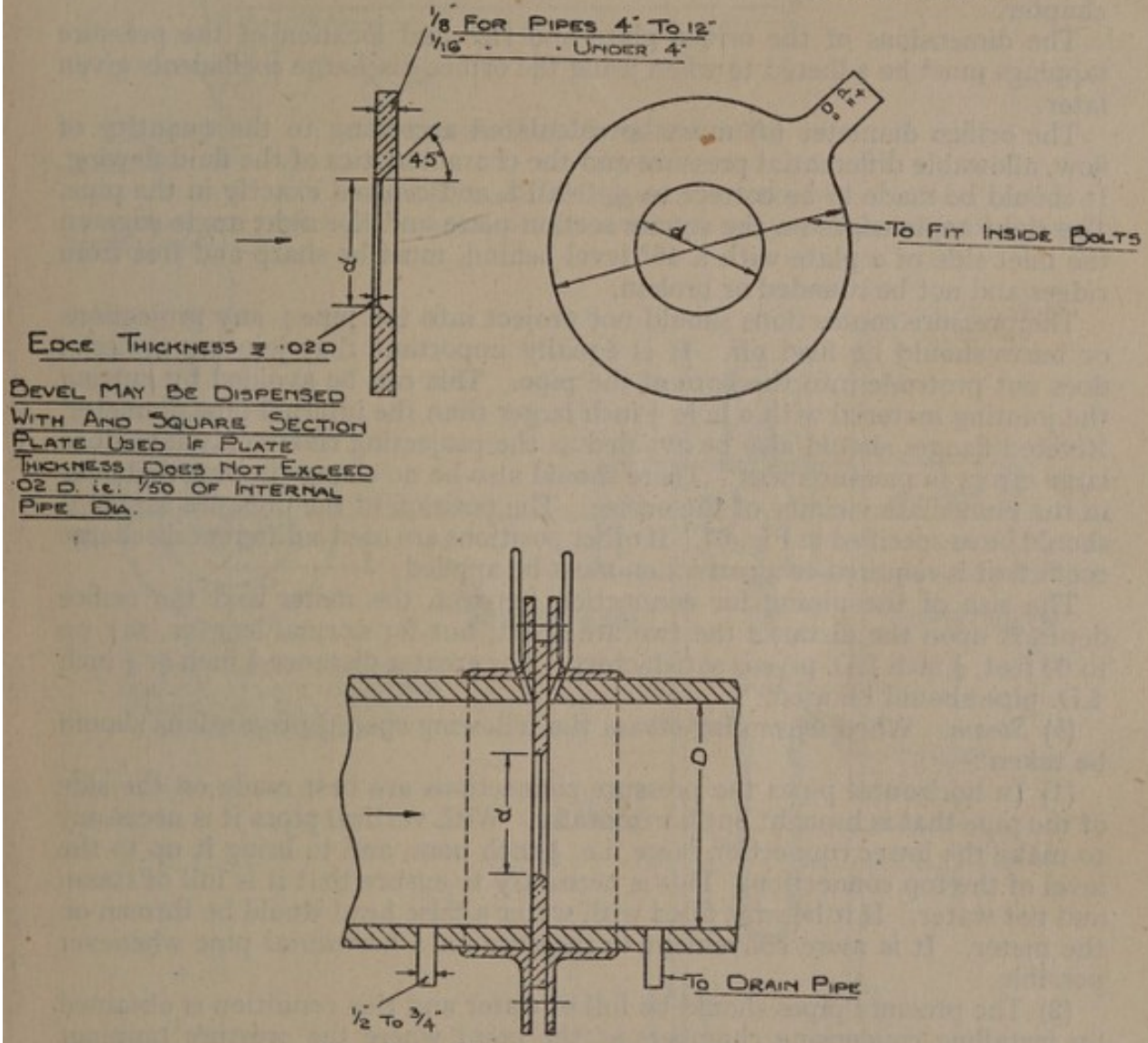

FIG. 67B. Construction of an orifice plate.

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DESIGN DATA

(*a*) *General.* The basic design data covering the measurement of flow by orifice plates applies in general to all types of fluids.

The orifice plate is usually fitted between pipe flanges, and pressure connections are made on either side of the plate. An orifice may be used in any pipe not less than 2 inches internal diameter provided that "m," the ratio of orifice area to pipe area, does not exceed 0·7 and d, the orifice diameter, is not less than 0·25 inch. For pipe diameters less than 2 inches, it is preferable to order a

short length of pipe fitted with the pressure differential device as a complete unit from a meter manufacturing firm.

Typical orifice arrangements are given in Fig. 67 (*a*) and (*b*).

In Fig. 67 (*a*) the orifice plate is fitted to a carrier ring which also includes the pressure connections so that the whole may be inserted between pipe flanges at one operation. As an alternative for conditions when it is difficult to spring the flanges a sufficient distance to insert the carrier ring, method (*b*) may be adopted. In this arrangement the pressure holes should be drilled as close to the orifice plate as possible. This system of " corner tappings " as the position from which the pressure readings are taken is adhered to throughout this chapter.

The dimensions of the orifice plate and size and location of the pressure tappings must be adhered to when using the orifice discharge coefficients given later.

The orifice diameter (*d*) must be calculated according to the quantity of flow, allowable differential pressure and the characteristics of the fluid flowing. It should be made to be correct to $\pm$ 0·001d. and centred exactly in the pipe. The right angle edges on the square section plate and the right angle edge on the inlet side of a plate with a 45° level behind, must be sharp and free from ridges and not be rounded or broken.

The pressure connections should not project into the pipe ; any projections or burrs should be filed off. It is equally important that jointing material does not protrude into the bore of the pipe. This can be avoided by cutting the jointing material with a hole $\frac{1}{2}$ inch larger than the internal pipe diameter. Riveted flanges should also be avoided as the projecting rivets may introduce large errors in measurement. There should also be no change in pipe diameter in the immediate vicinity of the orifice. The position of the pressure tappings should be as specified in Fig. 67. If other positions are used a different discharge coefficient is required or a correction must be applied.

The size of the piping for connection between the meter and the orifice depends upon the distance the two are apart, but for normal lengths, say up to 60 feet, $\frac{1}{4}$ inch I.D. pipe is satisfactory. For greater distance $\frac{3}{8}$ inch or $\frac{1}{2}$ inch I.D. pipe should be used.

(*b*) *Steam.* When measuring steam the following special precautions should be taken :—

(1) In horizontal pipes the pressure connections are best made on the side of the pipe that is brought out horizontally. With vertical pipes it is necessary to make the lower connection large, i.e. $\frac{1}{2}$-inch bore, and to bring it up to the level of the top connection. This is necessary to ensure that it is full of steam and not water. If it became filled with water a false head would be thrown on the meter. It is more convenient to measure on a horizontal pipe whenever possible.

(2) The pressure pipes should be full of water and this condition is obtained by installing condensing chambers at the point where the pressure tappings are made. The pipes transmitting the pressure from the orifice to the manometer or recorder must be free from leaks as a small leak (even though it might appear insignificant) may cause serious errors.

(3) There must be no air locks in the pressure pipes. Air is removed from the pressure pipes, when first connecting up to the manometer or recorder, by slacking off the unions at the instrument, blowing steam through to expel the air from the pipes and finally tightening up the unions.

(4) The condensing chambers can easily be made from standard pipe fittings ; a simple type is illustrated in Fig. 68. The chambers should have sufficient area and condensing capacity to ensure that pressure pipes remain full of water and an equal head of water is thrown at all times on both pipes. The

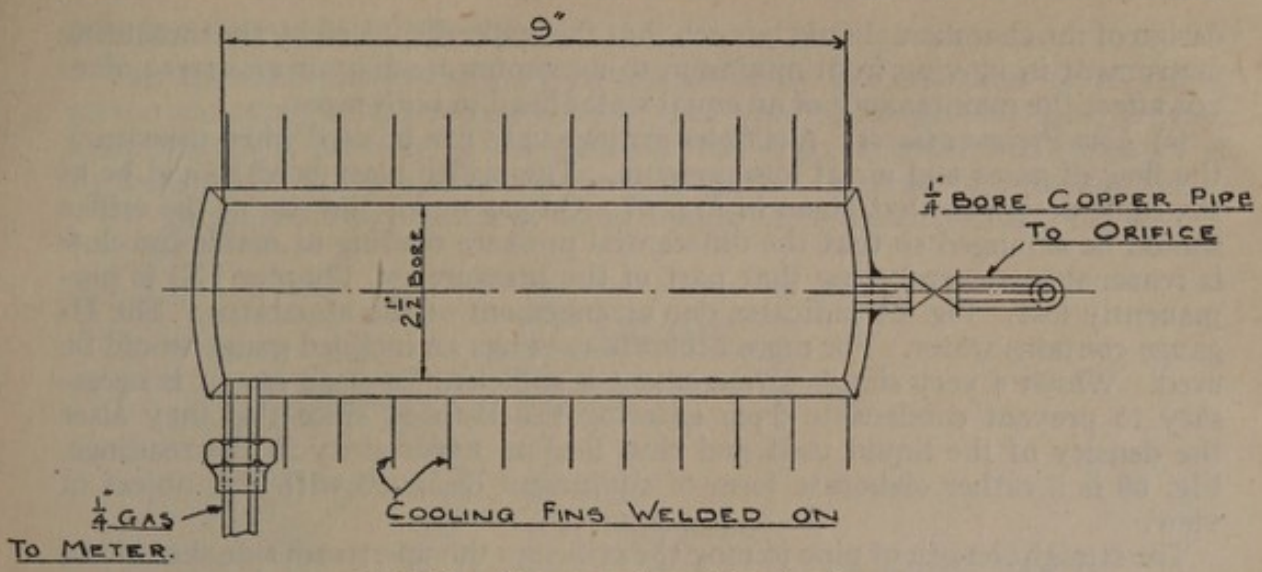

FIG. 68. Steam condensation chamber.

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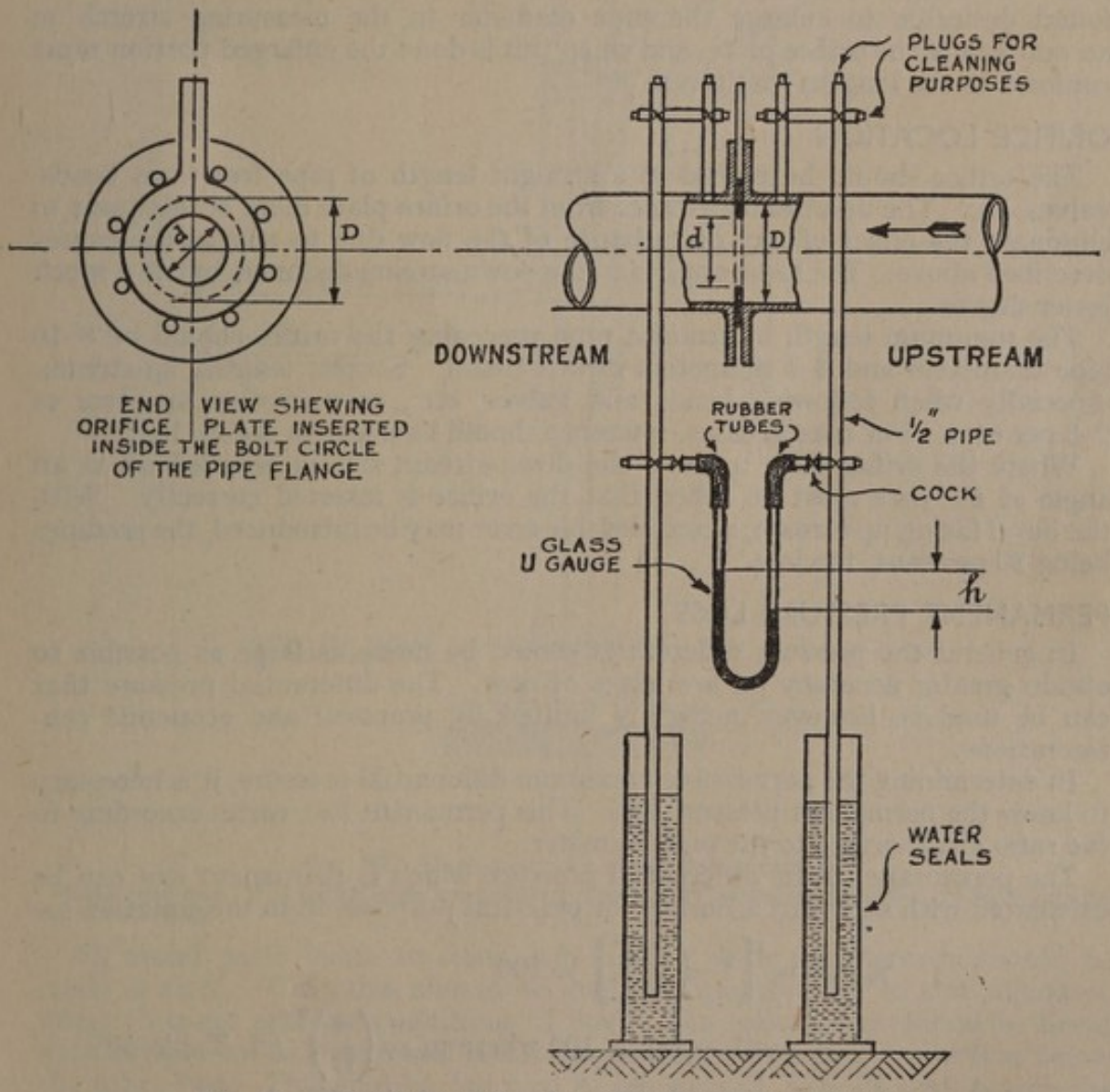

FIG. 69. Arrangement of orifice plate for gas measurement.

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design of the chambers should be such that the water displaced by the measuring instrument in moving from minimum to maximum reading or *vice versa*, does not affect the maintenance of an equal water head in both pipes.

(*c*) *Low Pressure Gases.* A simpler arrangement can be used when measuring the flow of gases and air at low pressure. The orifice plate used should be of the same design as that given in Fig. 67. On gas mains the size of the orifice should be arranged so that the differential pressure reading at maximum flow is reasonable, remembering that part of the pressure (cf. Chapter IX) is permanently lost. Fig. 69 indicates one arrangement of the apparatus. The U-gauge contains water. For more accurate readings an inclined gauge would be used. Whilst a very simple arrangement is sufficient for clean gas, it is necessary to prevent condensate from entering the U-tubes since this may alter the density of the liquid used and thus lead to uncertainty in the readings. Fig. 69 is a rather elaborate form of equipment designed with this object in view.

The straight length of pipe joining the orifice to the up-stream side should not be less than 10 pipe diameters long; on the down-stream side it may be less without the danger of introducing errors into the measurements. The off-takes must be arranged as close as possible to the orifice diaphragm. It is sometimes found desirable to enlarge the pipe diameter in the measuring stretch to accommodate the orifice plate, and when this is done the enlarged portion must conform to the lengths just given.

ORIFICE LOCATION

The orifice should be placed in a straight length of pipe free from bends, valves, etc. The up-stream distance from the orifice plate must be adequate to eliminate the effect of any disturbance of the flow due to any of the causes described above. The same applies to the down-stream distance, but to a much lesser degree.

The minimum length of straight pipe preceding the orifice should be 8–10 pipe diameters and 3–5 diameters down-stream. Shorter lengths up-stream, especially when following bends and valves, etc., may involve an error of 2–3 per cent. For special cases, reference should be made to B.S.S. 1042.

Where the orifice edge towards the down-stream side is bevelled off at an angle of 45° care must be taken that the orifice is inserted correctly. With the bevel facing up-stream, a considerable error may be introduced, the readings being 20 per cent. too low.

PERMANENT PRESSURE LOSS

In general the pressure differential should be made as large as possible to obtain greater accuracy on low rates of flow. The differential pressure that can be used is, however, generally limited by practical and economic considerations.

In determining the permissible maximum differential pressure, it is necessary to know the permanent pressure loss. This permanent loss varies according to the ratio of the orifice to the pipe diameter.

The percentage of the differential pressure which is permanent loss can be calculated with sufficient accuracy for practical purposes from the equation :—

$$\% \text{ loss} = \left[1-\left(\frac{d}{D}\right)^2\right] \times 100$$

$$= (1 - m) \times 100 \text{ where } m = \left(\frac{d}{D}\right)^2 \text{ (cf. Table 58).}$$

Thus if $m = 0{\cdot}4$, approximately 60 per cent. of the differential pressure is permanent loss.

MEASURING INSTRUMENTS

The differential pressure may be measured by a manometer or recording meter according to the requirements. The type of instrument used depends upon the static pressure of the fluid and the range of differential pressure to be measured. When measuring the rate of flow of steam the manometer or " U " gauge must be capable of withstanding the maximum steam pressure.

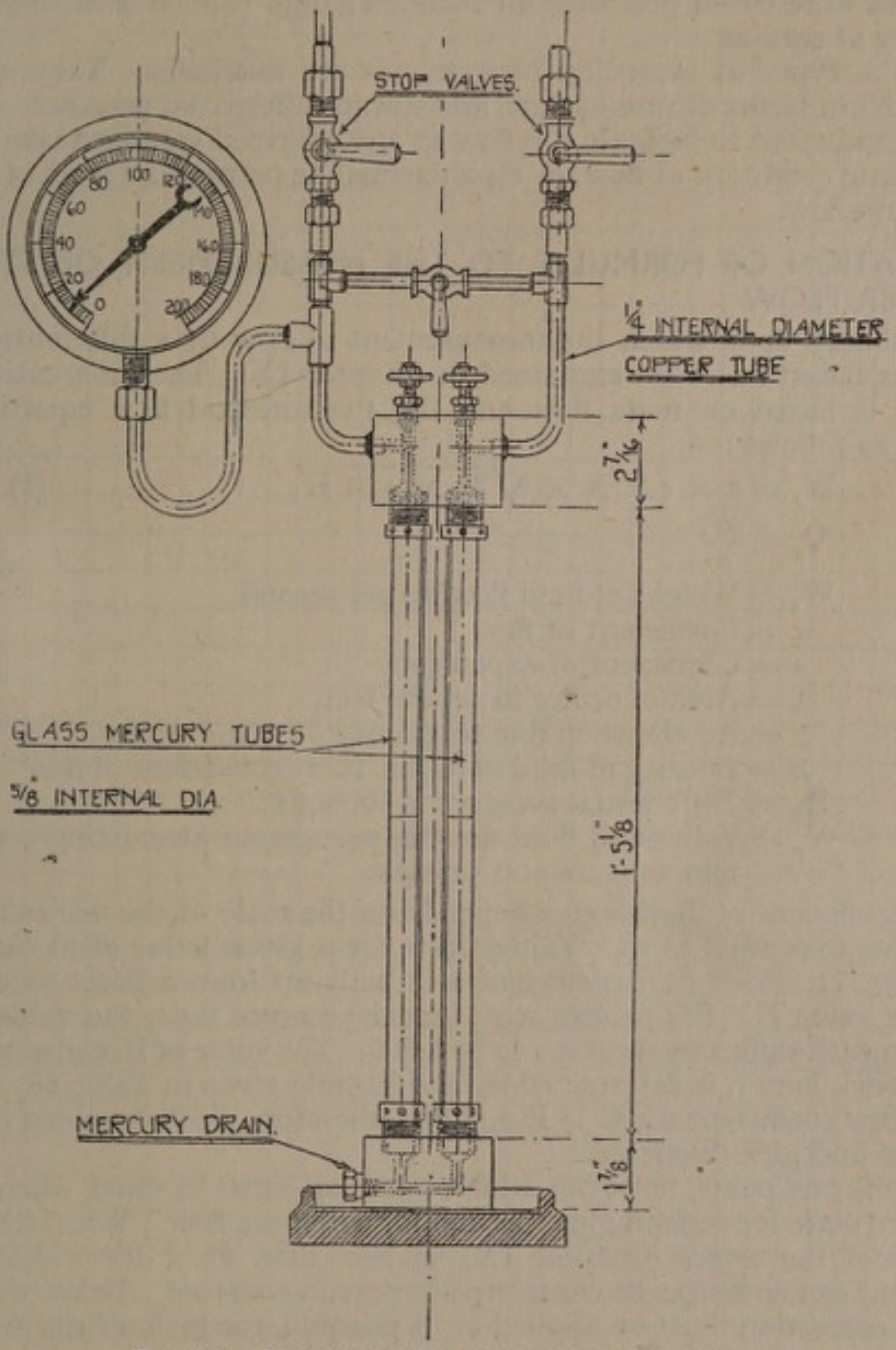

FIG. 70. Manometer for steam measurement.

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All metal parts liable to come into contact with the mercury should be made in steel. The tubes should be made sufficiently long to give adequate range to cover extreme conditions of flow. The manometer should be fitted with a valve to each limb and a third equalising valve on a connection below the other two. This enables the zero to be checked, and also safeguards the manometer from losing its indicating liquid should a sudden excess pressure occur on one side, when the limb valves are opened. Spindle screw valves should

preferably be used. A gauge to show the up-stream pressure should also be included. This is conveniently fixed to the high-pressure limb of the manometer. Such manometers may be purchased or made up in the workshop. A typical design is shown in Fig. 70.

For gaseous flow measurement, the differential pressure rarely exceeds 3 inches w.g., which allows the use of a " U " gauge of simple construction. For small differential pressures an inclined gauge can be used to give greater accuracy of reading.

Various types of recording instruments are available. They are usually calibrated in terms of rate of flow, and not in differential pressure. The scales can be graduated to indicate the flow at any specified standard condition for a given basic condition of flow, or on a decimal or percentage scale of a specified maximum flow.

APPLICATION OF FORMULÆ TO THE MEASUREMENT OF STEAM, GAS AND AIR FLOW

The theory upon which the measurement of flow by a differential pressure device is based has been explained in Chapter IX. The differential pressure method is based on mass flow and the fundamental flow equation can be written as follows :—

$$W_s = \alpha \times \epsilon \times A \times \sqrt{2g \times \rho \times H_1} \quad . \quad . \quad . \quad . \quad (1)$$

$$V_s = \frac{W_s}{\rho}$$

W_s = Weight of fluid flowing per second.
α = Coefficient of flow.
ϵ = Coefficient of expansion.
A = Area of orifice in square feet.
g = Acceleration due to gravity 32·2 ft./sec./sec.
ρ = Density of fluid in lb./cu. ft. at conditions of flow.
H_1 = Differential pressure in lb./sq. ft.
V_s = Volume of fluid flowing per second at existing conditions of temperature and pressure.

The coefficient of discharge α depends on the ratio of the orifice area to the pipe area, expressed as m. Values for α for a given value of m can be taken from Fig. 71. This flow coefficient α is built up from a basic value C and a variable value E. For orifices with corner pressure taps, the value for C can be taken with sufficient accuracy to be 0·605. The value of E, called the velocity of approach factor, is determined by the formula given in Table 58.

The flow coefficient α is C × E and can, therefore, be determined for any size of orifice and pipe diameter.

There is one point, however, which must be borne in mind when designing an orifice plate for a given pipe size and for a definite flow. When the Reynolds number of the *orifice* (Chapter IX) is less than $25 \times 10^4 \times d/D$, the flow coefficient can no longer be considered to remain constant. Below that limiting value a correction must be applied or, if possible, the ratio of the orifice to the pipe area must be reduced.

In technical literature the Reynolds number is very often based on the diameter of the pipe, and not on the diameter of the orifice.

$$\text{Re (orifice)} = \text{Re (pipe)} \times D/d.$$

The conversion of the fundamental flow formula in self-consistent units into a formula for practical use, where the area is replaced by the diameter of the orifice, can be written as follows :—

$$W = 1{,}273\, \alpha\, \epsilon\, d^2 \sqrt{H\rho} \text{ lb./hr.} \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

or $$W = 359\, \alpha\, \epsilon\, d^2 \sqrt{h\rho} \text{ lb./hr.} \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

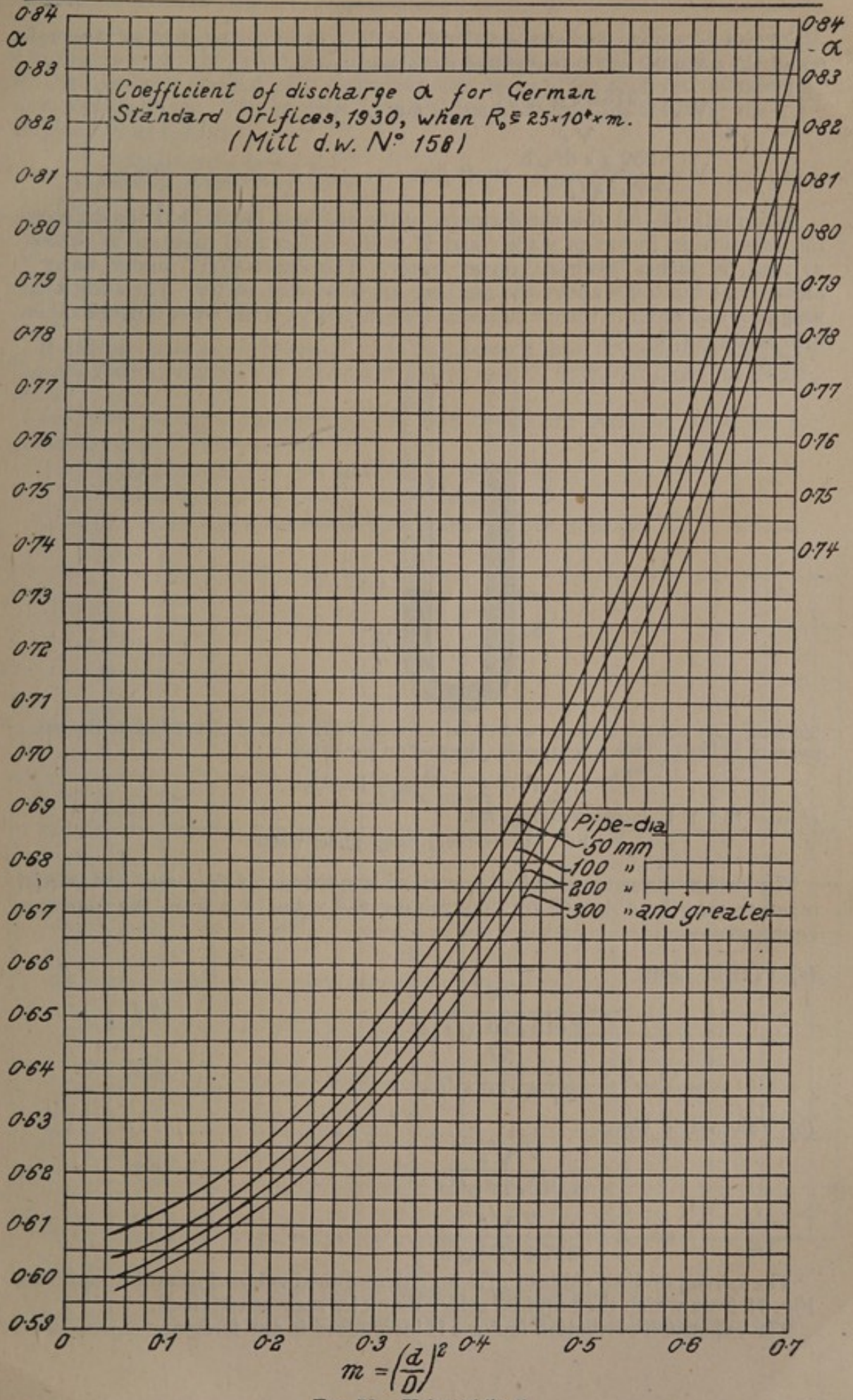

FIG. 71. Value of "α."

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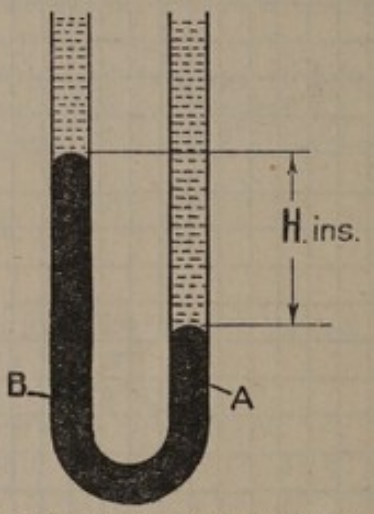
For the determination of the volumetric flow, as in gas or air flow measurement, the formulæ can be expressed as follows :—

$$V = \frac{1{,}273\,\alpha\,\epsilon\,d^2\sqrt{H}}{\sqrt{\rho}} \text{ cu. ft./hr.} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (5)$$

or $$V = \frac{359\,\alpha\,\epsilon\,d^2\sqrt{h}}{\sqrt{\rho}} \text{ cu. ft./hr.} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (6)$$

In formulæ (3)–(6), d is the diameter of the orifice in inches, h is the differential pressure in inches of water, and H is the differential pressure at orifice, as read, in inches mercury, under water. The expression "under water" means that the tubes above the mercury are filled with water. The additional weight of this water in the limb in which the mercury stands lower must be taken into account (Fig. 72).



Since mercury in limb A has an additional pressure of H ins. w.g. upon it above the water pressure in limb B, the true pressure showing on the gauge is :

$$H - \frac{H}{\text{sp. gr. of Hg.}} \text{ ins. Hg.}$$

or h = 12·6 × H inches water gauge.

FIG. 72. Mercury gauge under water.

The orifice diameter required for given conditions of flow and differential pressure may be obtained by determining the value of x from the following equations :—

(*a*) Weight Units

$$x = \frac{W}{1{,}018\,\epsilon\,D^2\sqrt{H\rho}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (7)$$

or $$x = \frac{W}{287{\cdot}4\,\epsilon\,D^2\sqrt{h\rho}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (8)$$

(*b*) Volumetric Units

$$x = \frac{V\sqrt{\rho}}{1{,}018\,\epsilon\,D^2\sqrt{H}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (9)$$

or $$x = \frac{V\sqrt{\rho}}{287{\cdot}4\,\epsilon\,D^2\sqrt{h}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (10)$$

The corresponding value of d/D for a determined value of x can be read from Fig. 73 from which the required orifice diameter may be obtained.

The expansion correction factor ϵ is introduced in the foregoing equations to

allow for the change in density between the two sides of the orifice. For normal conditions where the pressure drop across the orifice does not exceed 5 per cent. of the absolute pressure the correction can be omitted without introducing any serious error. The approximate errors introduced by omitting the correction are as follows :—

Differential pressure as percentage of absolute pressure, i.e. $\frac{p_1 - p_2}{p_1} \times 100$	Error per cent. fast.
1·25	0·5
2·50	1·0
5·00	2·0

If greater accuracy is required, or if the pressure drop exceeds 5 per cent. of the absolute pressure, the correction factor can be obtained from the B.S.S. 1042.

For more accurate calculations the flow coefficient must be corrected for certain factors. These include :—

An extra tolerance for the Reynolds number, Re.

An extra tolerance for pipe or main diameter, D.

Complete information covering these points will be found in B.S.S. 1042.

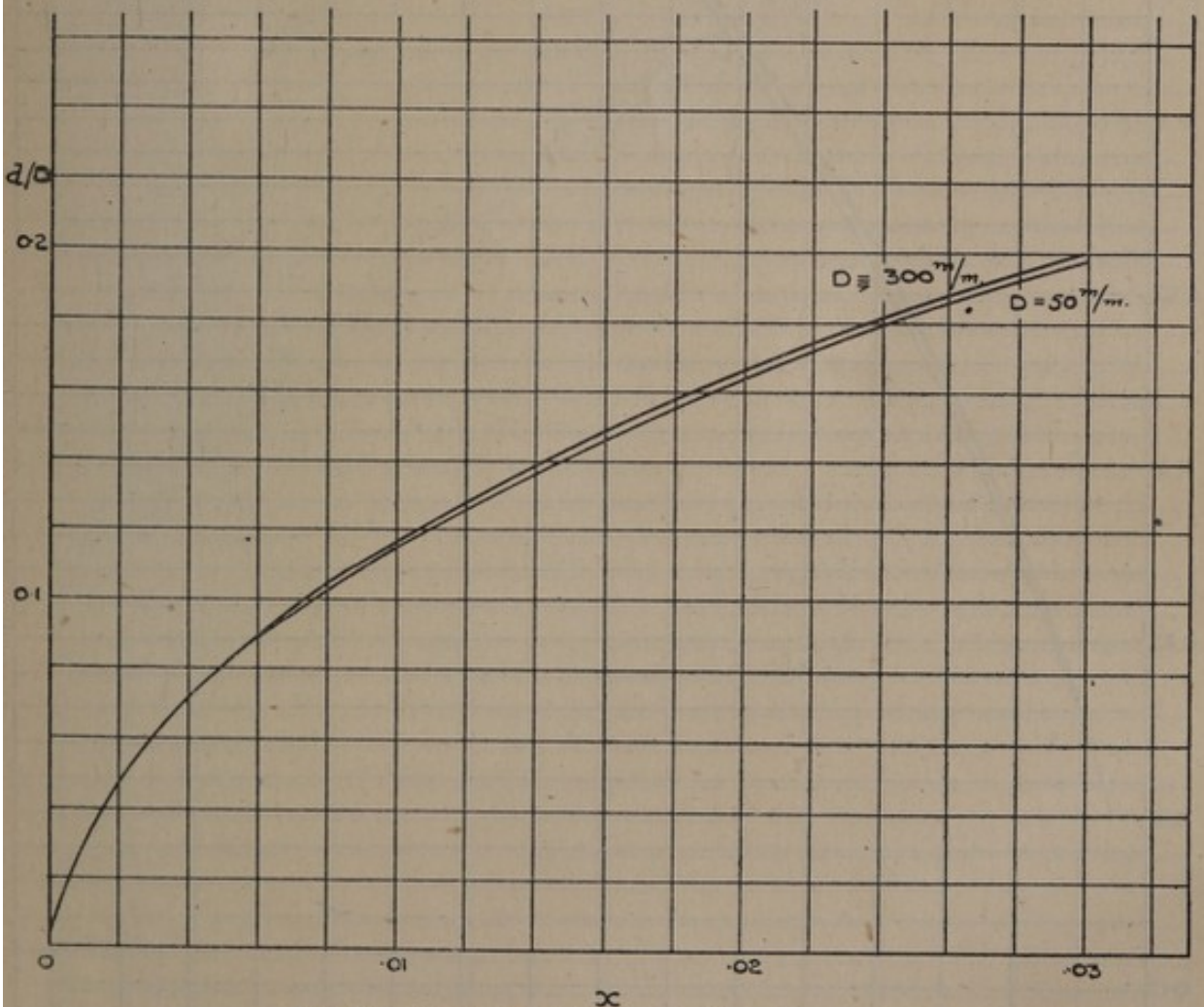


FIG. 73. Chart for determining orifice diameter.

(Reproduced from " Fuel Saving in the Iron and Steel Industry " by permission of the Iron and Steel Industrial Research Council.)

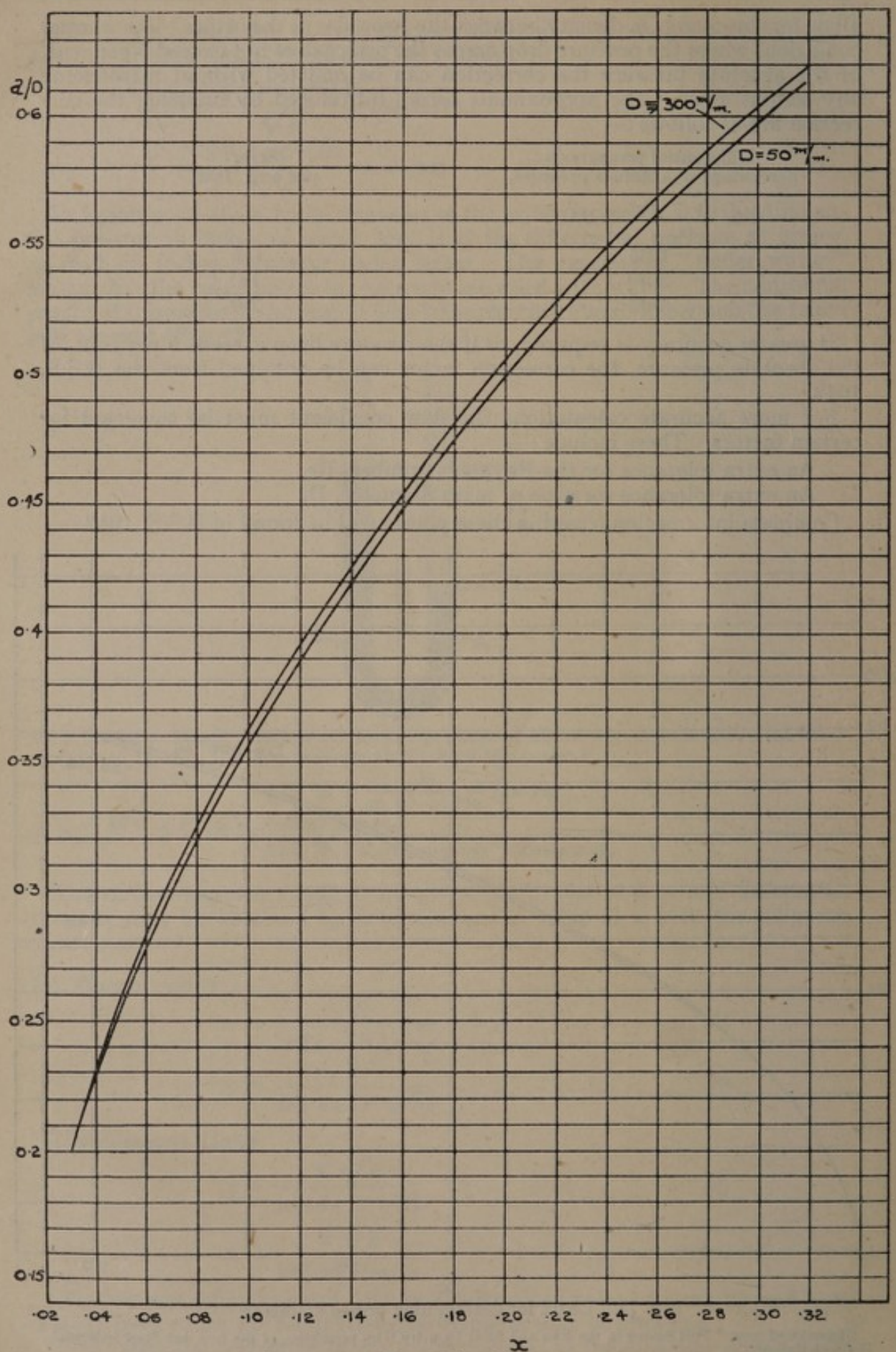

d/D
0·6
0·55
0·5
0·45
0·4
0·35
0·3
0·25
0·2
0·15
D = 300 m/m.
D = 50 m/m.
·02 ·04 ·06 ·08 ·10 ·12 ·14 ·16 ·18 ·20 ·22 ·24 ·26 ·28 ·30 ·32
x

FIG. 73 (*contd.*)

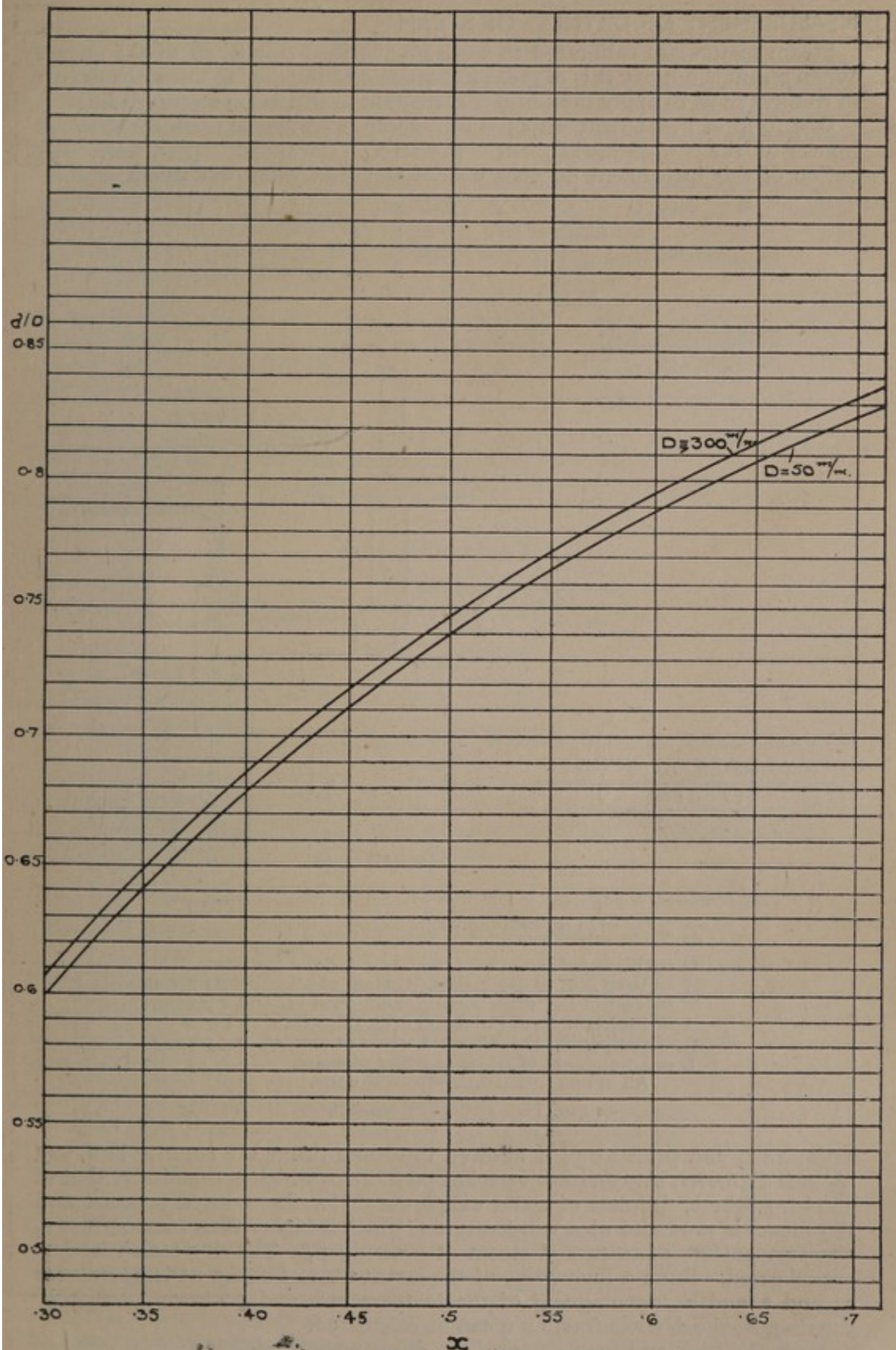
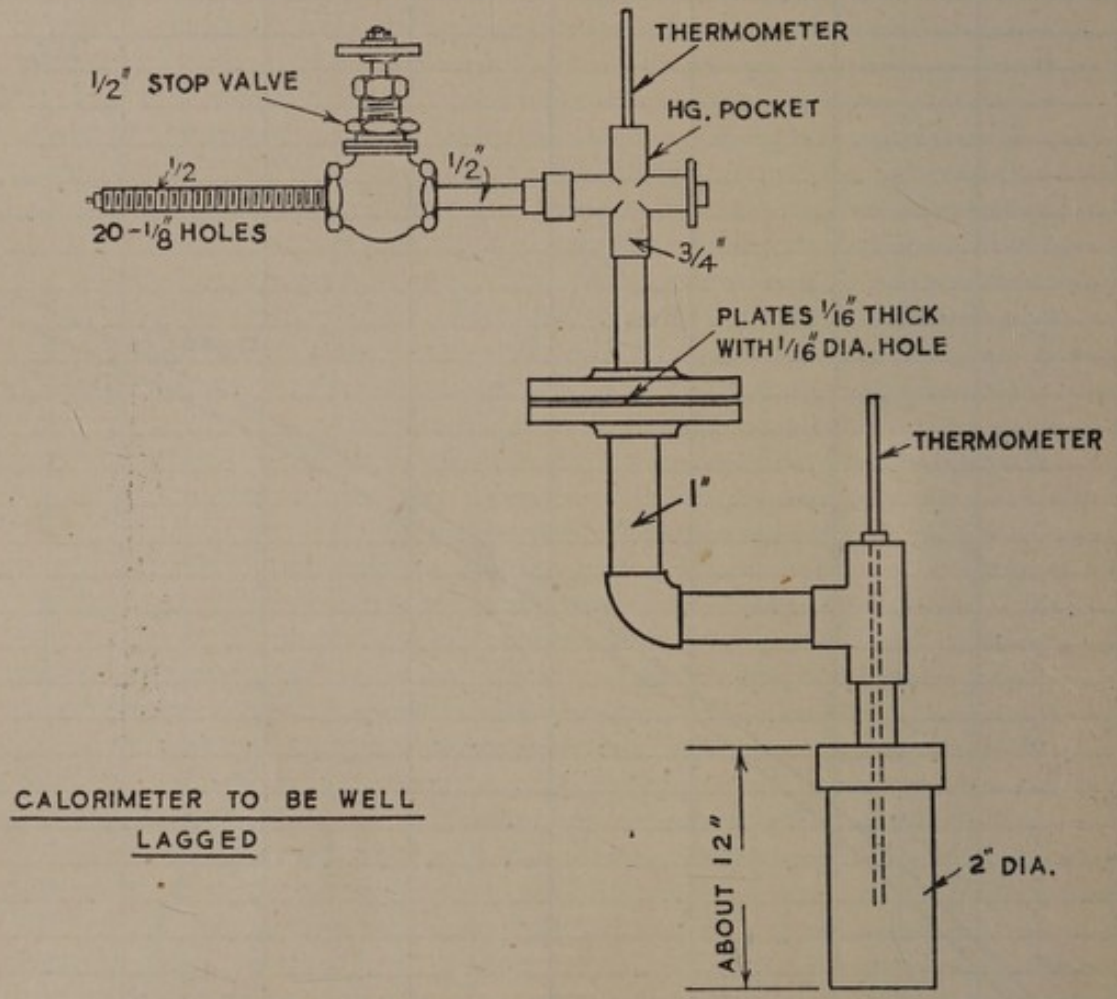


FIG. 73 (*contd.*)

MEASUREMENT OF DRYNESS OF STEAM

Steam meters are calibrated to work on specified conditions of the steam. When steam is not dry it is of great importance to the user to know the degree of wetness so as to apply the suitable correction to the steam meter readings.

One form of instrument which is commonly used and easy to construct is known as the " wiredrawing " or " throttling calorimeter " (Fig. 74). The action of this instrument depends on the fact that when wet steam at tem-



$$Q = \frac{(H - h) + 0{\cdot}46(t - t_1)}{L}$$

where : Q = Quality of steam expressed as a decimal fraction.
H = Total heat of dry saturated steam at atmospheric pressure.
h = Heat of the water corresponding with the boiler pressure.
t = Temperature of the steam after the orifice.
t_1 = Temperature of saturated steam at atmospheric pressure.
L = Latent heat of steam at boiler pressure.
All values in B.Th.U.—lb.—°F units.

FIG. 74. Throttling calorimeter.

perature t_2 and pressure p_2 is allowed to escape through a small orifice, say $\frac{1}{16}$ inch diameter, into another vessel at pressure p_1 which is practically that of the atmosphere, droplets of water evaporate under the reduced pressure and dry steam is produced at a temperature t, which is higher than temperature t_1 the temperature of saturated steam at pressure p_1. The dry steam is thus superheated. From a knowledge of the latent heat of steam at temperatures t_1 and t and a measurement of the temperatures and pressures previously mentioned, the dryness fraction q can be calculated.

Both the high and low pressure chambers must be well lagged to prevent

heat loss or jacketed with steam at the same pressure and temperature. Steam should always be passed for a long enough period to establish a steady state before readings are taken. With these precautions, this instrument should be capable of considerable precision, but can only be used when the steam has a fairly high dryness fraction, e.g. when the wetness of the steam is below about 4 per cent.

The method of calculation can best be explained by a numerical example. Steam is obtained from a boiler operating at 100 lb. per square inch absolute and is expanded to atmospheric pressure, 14·7 lb. absolute. The temperature in the expansion chamber is 260° F. What is the dryness of the steam ?

From steam tables,

	At 100 lb.	At 14·7 lb.
Temperature of steam—°F. ..	.. 327·8	212
Sensible heat in water—B.Th.U./lb.	.. 298·4	180·1
Latent heat of steam—B.Th.U./lb.	.. 888·8	970·3
Dryness of steam—per cent. ..	.. x	Superheated to 260° F.

When the steam is expanded, the sensible heat in the water is reduced from 298·4 to 180·1, i.e. by 118·3 B.Th.U. ; this heat is evolved.

The latent heat of the steam increases from 888·8 to 970·3 and this absorbs heat ; the heat so absorbed is

$$\frac{x}{100}(970{\cdot}3 - 888{\cdot}8)$$

Any water present as such in the steam is converted into steam at 14·7 lb. and absorbs 970·3 B.Th.U./lb. of water, this being equal to

$$\left(1 - \frac{x}{100}\right) 970{\cdot}3$$

The net result is that the heat evolved

$$= 118{\cdot}3 - \frac{x}{100}(970{\cdot}3 - 888{\cdot}8) - \left(1 - \frac{x}{100}\right) 970{\cdot}3 \text{ B.Th.U./lb. of total steam.}$$

This heat is given to the steam and superheats it. The degree of superheat is 260°–212° F., and since the sp. ht. of steam at atmospheric pressure is 0·46 B.Th.U./lb./°F. the heat measured as given to the steam is 0·46 (260—212).

Since heat evolved = heat given to steam

$$118{\cdot}3 - \frac{x}{100}(970{\cdot}3 - 888{\cdot}8) - \left(1 - \frac{x}{100}\right) 970{\cdot}3 = 0{\cdot}46\ (260–212)$$

whence $8{\cdot}888x = (970{\cdot}3 - 118{\cdot}3) + 0{\cdot}46\ (260–212)$ . . (1)

and $x = 98{\cdot}33$

From (1), by adding 180·1 on to 970·3 and 118·3, it will be evident that if

Q = dryness fraction of the steam expressed as a decimal
H = total heat of dry, saturated steam at atmospheric pressure
h = sensible heat of water corresponding with boiler pressure
t = temperature of steam after orifice
t_1 = temperature of saturated steam at atmospheric pressure
L = latent heat of steam corresponding to boiler pressure

$$Q = \frac{(H - h) + 0{\cdot}46\ (t - t_1)}{L}$$

PULSATING FLOW

The orifice method described provides an accurate method of measurement of what may be termed steady flows, that is, when the indicating gauge or meter is able to follow all changes of velocity. This, however, does not apply when the orifice is placed near a reciprocating engine, when serious errors in the meter readings may be introduced due to velocity pulsations.

Whenever a pulsating flow has to be measured it is advisable to consult the meter manufacturers.

CORRECTION FACTORS FOR APPLICATION TO METER READINGS *

A meter, unless fitted with an integral correction device, will give a true record of differential pressure only, and the conversion of the record into units of flow by a single multiplying constant can be carried out for one set of fluid conditions only.

For other conditions, the true rate of flow

= Flow obtained from meter readings at design conditions, multiplied by a correction factor F.

If F = Correction factor
P = Absolute pressure, up-stream tapping, in inches Hg
T = Absolute temperature of the gas in °F. at the same point as P
w = Vapour tension of water in inches Hg (taken from tables)
s = Specific gravity of gas (dry gas to dry air)
0·624 = Specific gravity of water vapour relative to dry air
b = Suffix to denote basic conditions for which meter was designed.
m = Suffix to denote metering conditions,

the factor F for various units and fluids is determined as follows :—

(*a*) *Steam and liquids* (*lb. per hour*).

$$F = \frac{\sqrt{\rho_m}}{\sqrt{\rho_b}}$$

Correction factors for steam may be read from curves given in Figs. 75–78.

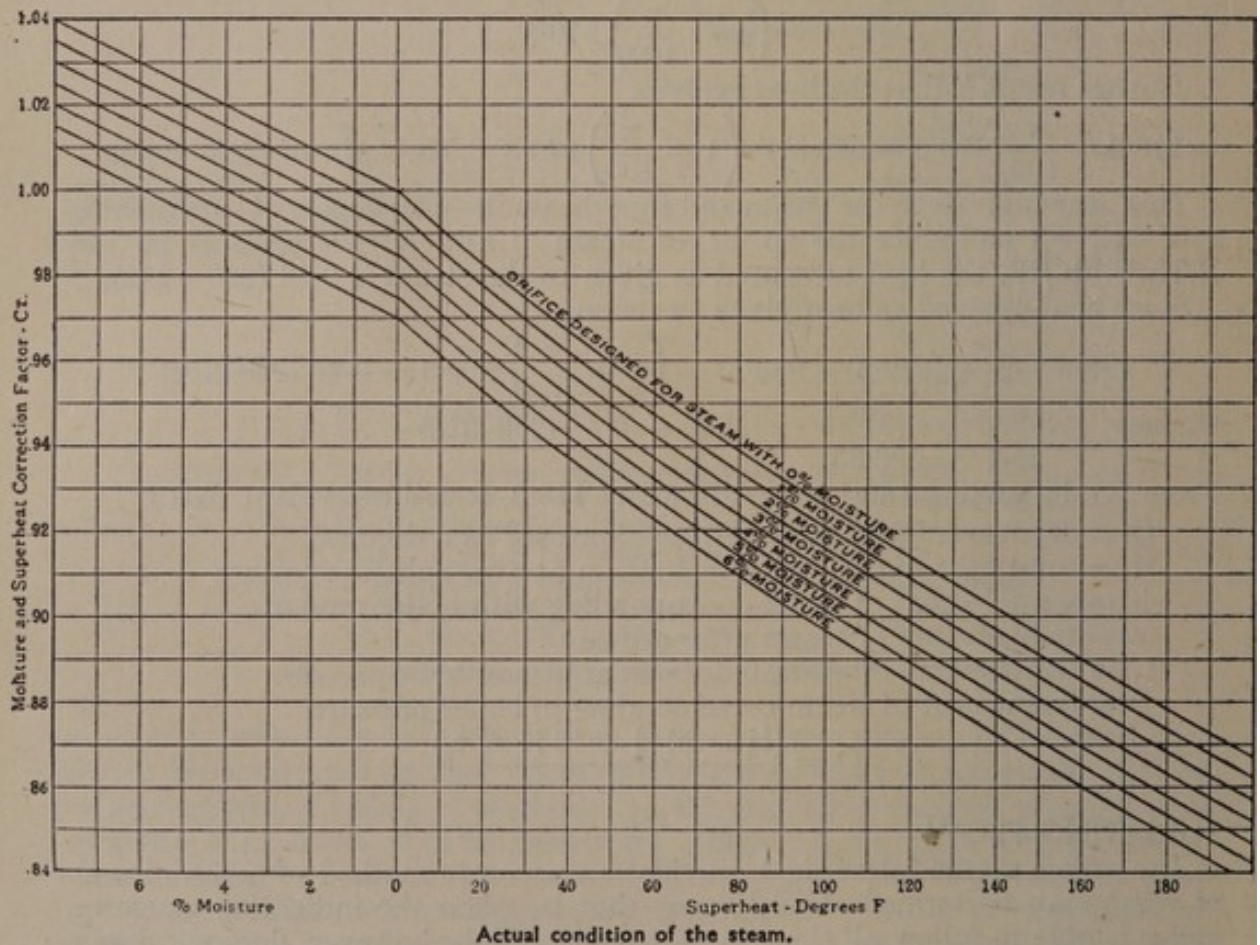


FIG. 75. Correction factors for steam meters.

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* B.S.S. 1042.

Correction factors for steam. Orifices designed for superheat or moisture content shown by curve.
Weight (in lb. per hour) = Meter reading × Constant × C_P × C_T where
C_P = Pressure correction factor.
C_T = Superheat or moisture correction factor.
Example: Temperature correction for a meter designed for 190° superheat but used at actual average operating conditions of 120° superheat correction factor = 1·045 as shown.

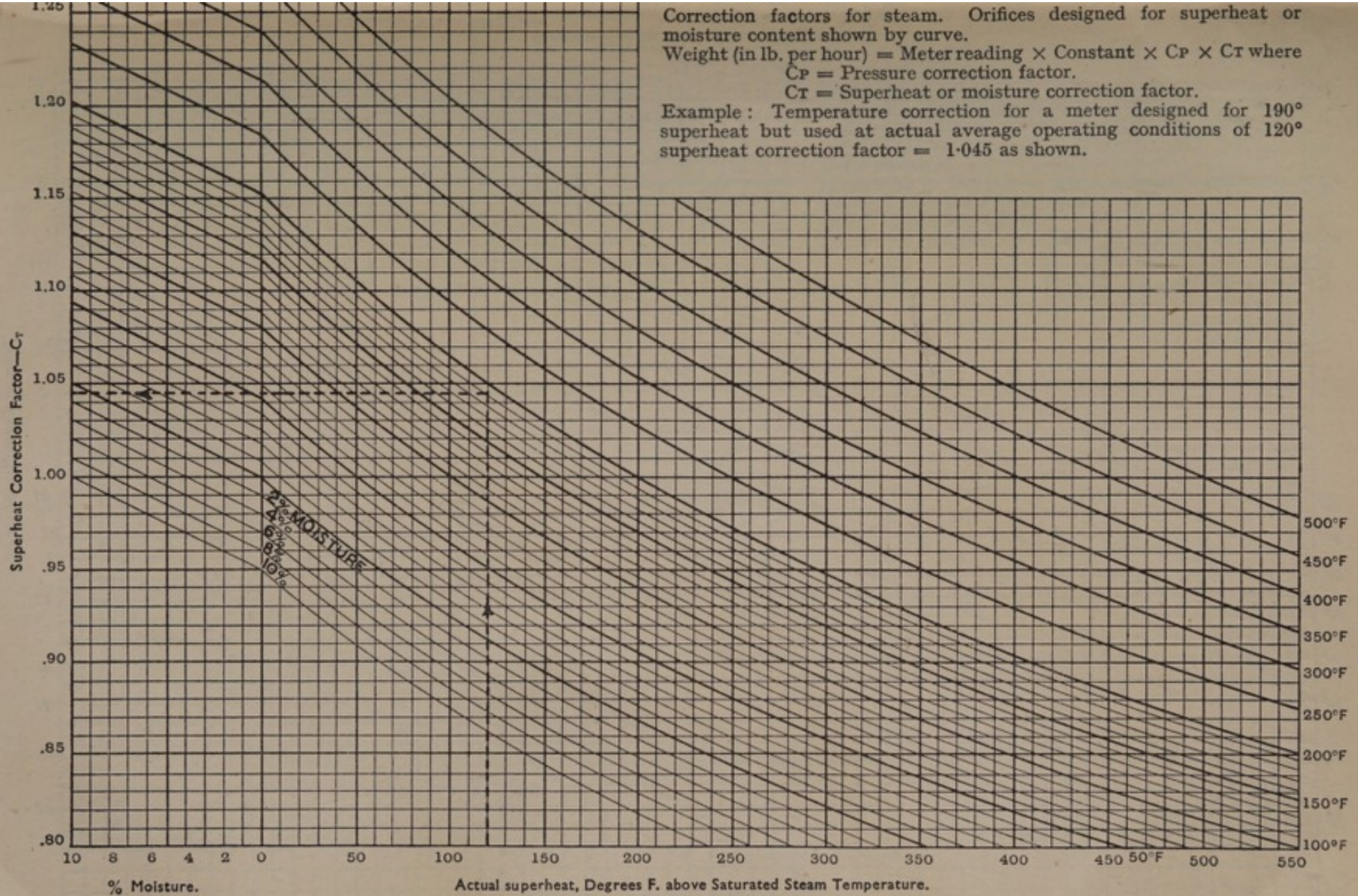

FIG. 76. Correction factors for steam meters.

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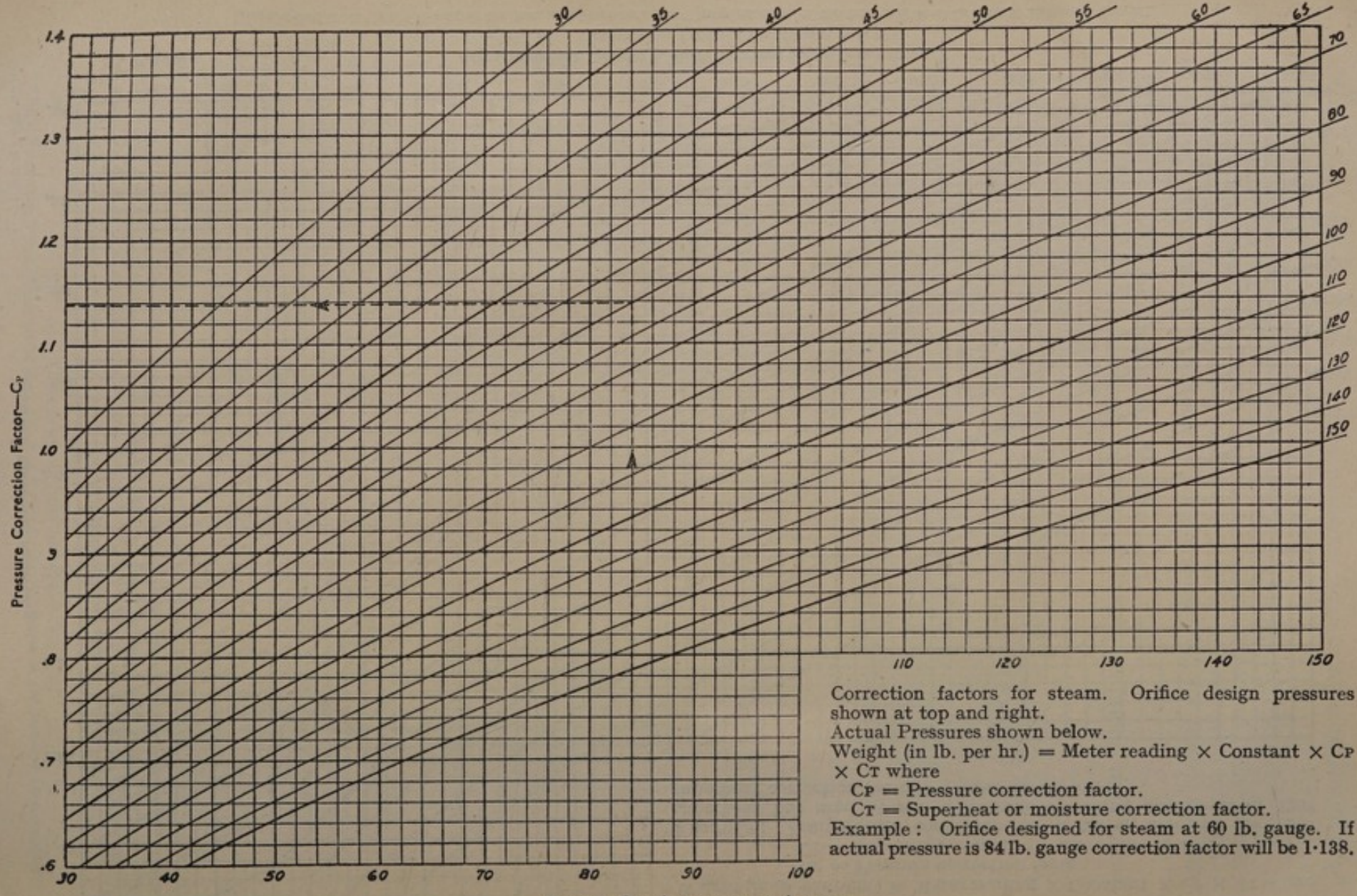
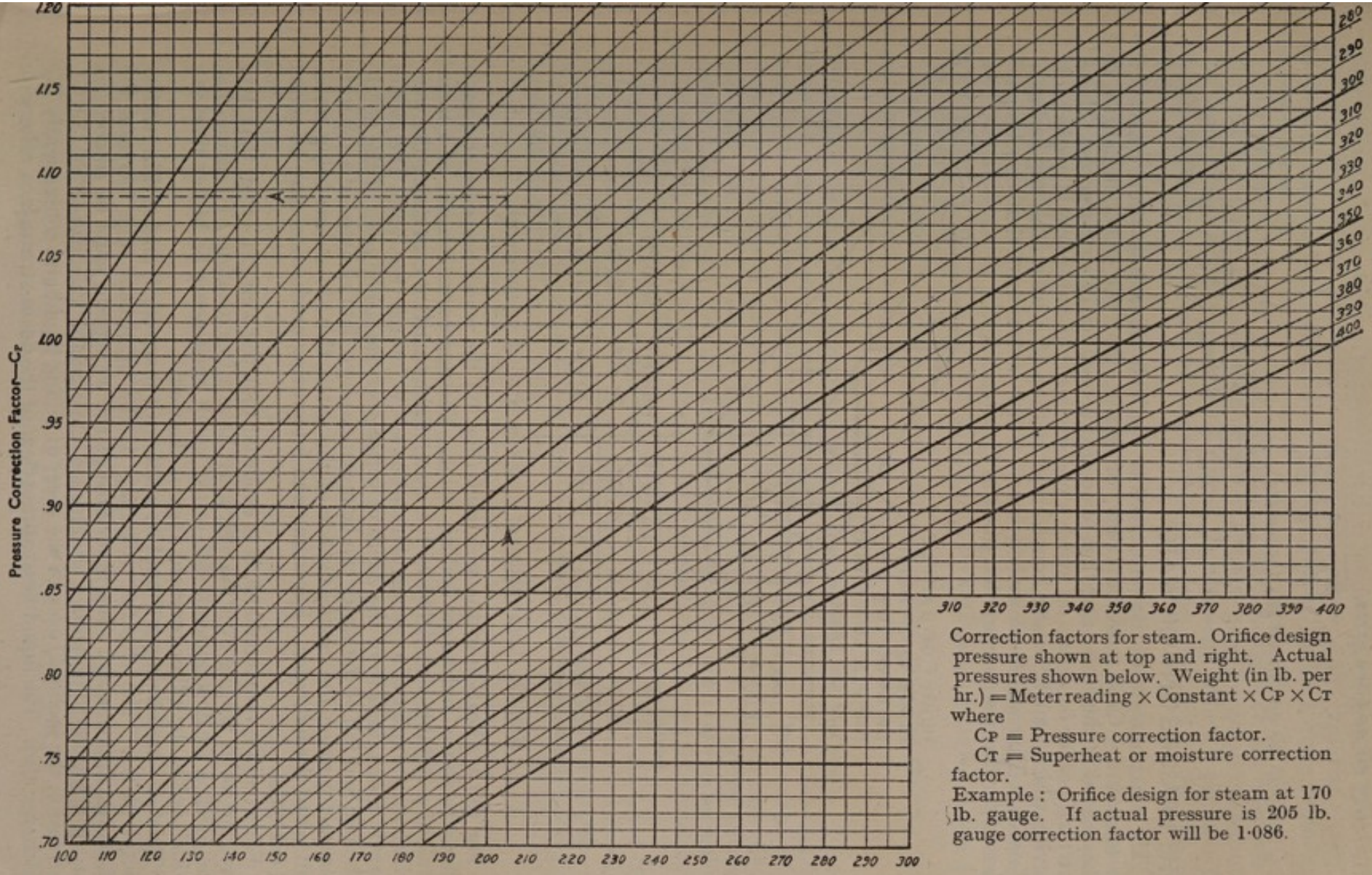


Correction factors for steam. Orifice design pressures shown at top and right.
Actual Pressures shown below.
Weight (in lb. per hr.) = Meter reading × Constant × C_P × C_T where
C_P = Pressure correction factor.
C_T = Superheat or moisture correction factor.
Example : Orifice designed for steam at 60 lb. gauge. If actual pressure is 84 lb. gauge correction factor will be 1·138.

FIG. 77. Correction factors for steam meters.



Correction factors for steam. Orifice design pressure shown at top and right. Actual pressures shown below. Weight (in lb. per hr.) = Meter reading × Constant × C_P × C_T where

C_P = Pressure correction factor.

C_T = Superheat or moisture correction factor.

Example : Orifice design for steam at 170 lb. gauge. If actual pressure is 205 lb. gauge correction factor will be 1·086.

FIG. 78. Correction factors for steam meters.

(Reproduced from "Fuel Saving in the Iron and Steel Industry" by permission of the Iron and Steel Industrial Research Council.)

(*b*) *Dry air and dry gases.*

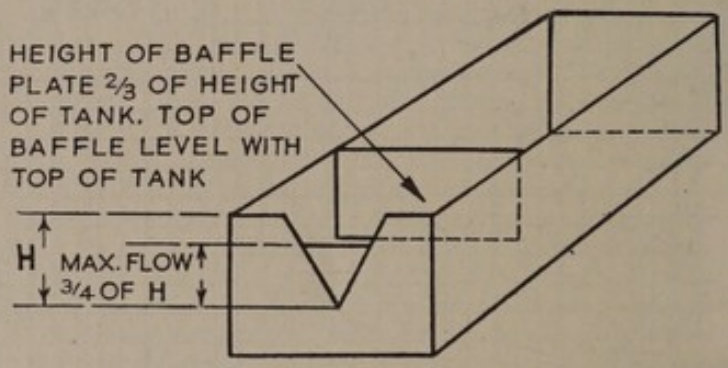
$$F = \sqrt{\frac{P_m \times T_b \times s_b}{P_b \times T_m \times s_m}}$$

(*c*) *Moist air and moist gas* (*saturated*).

$$F = \sqrt{\frac{T_b}{T_m}} \times \sqrt{\frac{s_b(P_b - w_b) + 0{\cdot}624\, w_b}{s_m(P_m - w_m) + 0{\cdot}624\, w_m}} \times \frac{P_m - w_m}{P_b - w_b}$$

THE MEASUREMENT OF WATER

The measurement of the water feed to the boiler is an important control of performance. Care must be taken, of course, that the water measured is evaporated in the boiler and that none of it is used for other purposes. There are many other occasions on which a measurement of water flow is needed. A device which can be easily constructed is that of a weir tank employing a V-notch. This will be described here. Other instruments that can be purchased for this purpose are described in Chapter XXIX.



RATE OF FLOW	LENGTH	HEIGHT	WIDTH
4000 GALS./HR.	4'. 0"	2' 6"	2'. 0"
50,000 GALS./HR.	7'. 0"	3'. 0"	4'. 6"

FIG. 79. Construction of a tank for V-notch measurement.

For the application of the weir tank (Fig. 79) the water must be at atmospheric pressure and below boiling point. The apparatus consists of a metal tank of a suitable size, fitted with a water-tight partition containing the weir, which may consist of a rectangular slot or a " V " notch (Fig. 79). The rate of flow is ascertained from the depth of water over the sill or apex of the notch. The rise and fall of the water can be measured by a float situated in a separate compartment on the up-stream side of the notch, this movement being transmitted to the indicator. Recording and integrating mechanism may be added if desired.

For accurate measurement, it is essential that the weir runs free, that is to say, the water flowing over the weir must have a free overfall. To ensure this, the level of the down-stream side of the notch must never be higher than say 1 inch below the sill or reversed apex of the weir.

The " V " notch proper should be made with sharp edges and of thin rustless metal plates. The slightest rounding of the up-stream edge increases the discharge appreciably. The inner face of the weir plate must be smooth, as any roughness or projection diminishes the upward velocity of the liquid, increasing the sectional area of the sheet of falling liquid and consequently the discharge also. The liquid must spring clear of the down-stream face, otherwise

the discharge is increased. The head (H or h) is the distance (Figure in Table 60) between the crest of the notch or weir and the surface of the liquid at a point as near as possible to the outlet from the tank, but before the surface begins to curve downwards.

The tank may be fitted with an automatic regulating valve on the inlet side. This may be of the float-operated equilibrium type, with the float situated on the down-stream side of the notch. With this arrangement, if there is a tendency to drown the weir, the inlet valve automatically closes and reduces the rate of flow.

It will be readily appreciated that for accurate metering, steady flow conditions are required on the up-stream side of the notch ; thus the measuring tank should not be used as a surge tank, and live steam should not be allowed to enter.

An installation of this description will always register the correct volume of flow irrespective of the temperature. It can, however, be calibrated to read the correct weight of flow at any desired temperature, and corrective factors are supplied by which the meter readings must be multiplied if the set temperature varies from that specified.

The general theory of the flow of a liquid over a weir, indicates that for the arrangement shown in Fig. 79,

$$Q = C\,h^{2.5}\sqrt{2g}$$

where Q is the quantity of liquid flowing in unit time and h the head. C is a constant depending on the density, viscosity and surface tension of the liquid, and on the angle between the sides of the notch (Table 60). For water, the density, viscosity and surface tension may be regarded as known—though they will vary somewhat with the temperature—and can be incorporated in the constant, leaving only the angle between the sides of the notch as the variable. For any given instrument this angle also is fixed when the arrangement is set up, and the flow can be read directly from a chart when h is measured.

Table 60 gives three formulæ and indicates their value, with the head (H or h) as the only variable, for four selected values of the angle between the arms of the notch (θ). Table 61 gives for one of these formulæ complete details of the discharge under various conditions. Both these tables are taken from the handbook issued by George Kent Ltd.

TABLE 60. DISCHARGES OVER " V " NOTCHES

Symbols used :—

H = height over notch in feet.
h = height over notch in inches.
Q = flow in cubic feet per second.
G = flow in gallons per hour.
θ = angle of " V " notch.



Authority	General formulæ	Values of θ			
		90°	60°	45°	30°
Approx.	$Q = 2{\cdot}48\ H^{2.5} \times \tan\theta/2$ $G = 111{\cdot}5\ h^{2.5} \times \tan\theta/2$	$2{\cdot}48H^{2.5}$ $111{\cdot}5\ h^{2.5}$	$1{\cdot}43\ H^{2.5}$ $64{\cdot}4\ h^{2.5}$	$1{\cdot}03\ H^{2.5}$ $46{\cdot}2\ h^{2.5}$	$\cdot 665H^{2.5}$ $29{\cdot}9\ h^{2.5}$
Gourley & Crimp	$Q = 2{\cdot}48\ H^{2.47} \times \tan\theta/2$ $G = 120{\cdot}1\ h^{2.47} \times \tan\theta/2$	$2{\cdot}48H^{2.47}$ $120{\cdot}1\ h^{2.47}$	$1{\cdot}432H^{2.47}$ $69{\cdot}34\ h^{2.47}$	$1{\cdot}027H^{2.47}$ $49{\cdot}74\ h^{2.47}$	$\cdot 6645H^{2.47}$ $32{\cdot}17\ h^{2.47}$
Barnes	$Q = 2{\cdot}462H^{2.48} \times Z$ $G = 116{\cdot}3\ h^{2.48} \times Z$	$2{\cdot}48H^{2.48}$ $117{\cdot}1\ h^{2.48}$	$1{\cdot}435H^{2.48}$ $67{\cdot}8\ h^{2.48}$	$1{\cdot}032H^{2.48}$ $48{\cdot}73\ h^{2.48}$	$0{\cdot}669H^{2.48}$ $31{\cdot}61\ h^{2.48}$

where $Z = \tan\theta/2/(\frac{1}{2}\sin\theta/2)^{\cdot 00703}$

TABLE 61. DISCHARGES OVER "V" NOTCHES
(Calculated from Barnes' Formula)

Head in in.	Discharge in gallons/hour for angles of notch equal to :—										
	5°	10°	20°	30°	40°	45°	50°	60°	70°	80°	90°
2	29·10	58·03	116·4	176·4	239·1	271·9	305·9	378·2	447·9	548·8	653·6
4	162·3	323·7	649·3	983·7	1,334	1,517	1,706	2,110	2,449	3,061	3,646
6	443·8	884·9	1,775	2,689	3,646	4,146	4,664	5,768	6,830	8,369	9,967
8	905·7	1,806	3,622	5,489	7,442	8,462	9,520	11,770	13,940	17,080	20,340
10	1,575	3,141	6,300	9,547	12,940	14,720	16,560	20,480	24,240	29,710	35,380
12	2,476	4,937	9,902	15,005	20,340	23,130	26,020	32,180	38,110	46,690	55,600
14	3,628	7,236	14,510	21,990	29,810	33,900	38,140	47,170	55,850	68,430	81,500
16	5,053	10,070	20,210	30,620	41,520	47,210	53,110	65,680	77,770	95,290	113,500
18	6,767	13,490	27,070	41,010	55,600	63,230	71,130	87,960	104 100	127,600	152,000
20	11,060	17,520	35,150	53,260	72,210	82,110	92,370	114,200	135,300	165,700	197,400
24	13,810	27,540	55,240	83,710	113,500	129,000	145,200	179,500	212,600	260,500	310,100

PRACTICAL APPLICATION OF THE FLOW FORMULÆ

The problems usually met in practical flow measurement with differential pressure devices, can be stated to be as follows :—

(1) Determination of the rate of flow, for a specified orifice diameter and for a given or permissible differential pressure.
(2) Determination of orifice diameter required to measure the flow for a specified plant. Here it is first necessary to make an approximation of the maximum quantity of the fluid to be measured and to decide on the allowable pressure loss in the pipe line.
(3) For a specified maximum pressure loss (permanent loss), the d/D ratio must be carefully chosen, in order to be above the limiting value of Re at a considerably lesser flow. This applies especially to gaseous flow, where the available static pressure is very often less than 6 inches w.g. and where the permanent pressure loss must be kept at a minimum.

PRACTICAL APPLICATION

The following examples will illustrate the practical application of the flow formulæ for various fluids.

EXAMPLE A. *Superheated Steam—Calculation of Flow*

Data obtained :—

Steam pressure p = 150 lb./sq. in. gauge
= 164·7 lb./sq. in. absolute

Temperature of steam, t .. .. .. = 400° F.
Density, ρ .. .. = 0·340 lb./cu. ft.
Differential pressure, H (as read) .. .. = 4 inches Hg.
Pipe diameter, D .. .. = 8 ,,
Orifice diameter, d .. .. = 4 ,,

It is required to calculate the rate of flow in lb./hr.
For this example formula (3) is used.

$$W = 1{,}273\,\alpha\,\epsilon\,d^2\sqrt{H\rho}\ \text{lb./hr.},$$

but the correction for the expansion factor ϵ is neglected.

Orifice ratio d/D = 0·5, $m = (d/D)^2 = 0{\cdot}25$.
Coefficient of flow α (read from Fig. 71) = 0·627.
Rate of flow $W = 1{,}283 \times 0{\cdot}627 \times 4^2 \times \sqrt{4 \times 0{\cdot}34} = 14{,}900$ lb./hr.

EXAMPLE B. *Saturated Steam—Determination of Orifice Diameter*

Data :—

Rate of flow, W (known or assumed) ..	= 16,000 lb./hr.
Absolute pressure, p	= 134·7 lb./sq. in.
Temperature, t	= 350° F.
Density, ρ	= 0·3 lb./cu. ft.
Differential pressure, H (as read)	= 7·0 inches Hg
Expansion factor, ϵ	neglected
Pipe diameter, D	= 6 inches

It is required to determine what should be the diameter of the orifice, d. Here, formula (7) is used,

$$x = \frac{W}{1{,}018\ D^2 \sqrt{H\rho}}$$

$$= \frac{16{,}000}{1{,}018 \times 36 \times \sqrt{7 \times 0{\cdot}3}} = 0{\cdot}301.$$

Reading from Fig. 73, d/D for x = 0·301 is found to be 0·605.
Since D = 6 inches, the orifice diameter d = 6 × 0·605 = 3·63 inches.
When designing an orifice for use with a ring balance recorder equation (8) is used, the calculation being similar.
The Reynolds number for the foregoing example is determined as follows :—

$$\text{Re} = \frac{vD\rho}{\eta}^{*}$$

$$\text{Here } D = \frac{d}{12} = \frac{3{\cdot}63}{12} = 0{\cdot}3025 \text{ feet}$$

$$v = \frac{16{,}000}{0{\cdot}3 \times 3{,}600} \times \frac{144 \times 4}{3{\cdot}63^2 \times 3{\cdot}14} = 200 \text{ ft./sec.}$$

η (poises) from Table 50 = 160 × 10^{-6} C.G.S. units.
Conversion factor into F.P.S. units from Chapter IX = 0·0672.
η (British physical units) = 160 × 10^{-6} × 0·0672 = 10·7 × 10^{-6} F.P.S. units

$$\text{Re} = \frac{0{\cdot}3025 \times 200 \times 0{\cdot}3}{10{\cdot}7 \times 10^{-6}} = 1{,}690{,}000$$

$$\text{Re limit value}^{\dagger} = 250{,}000 \times \frac{3{\cdot}63}{6} = 151{,}000.$$

For the conditions specified above for temperature and pressure, the coefficient of flow will remain constant down to 10 per cent. of the maximum flow of 16,000 lb. per hour.

When the orifice has been designed for a given pressure and steam quality and actual conditions vary from this, correction factors to apply will be found from the curves in Figs. 75–78.

* This is the nomenclature given in Chapter IX. For application here, " D " is the diameter of the orifice in feet, i.e. d/12.

† When the values of Reynolds number are sufficiently high the flow co-efficients remain substantially constant for standardised nozzles and orifice plates, i.e. they remain unaffected by the velocity, density and viscosity of the flowing fluid. The values of Re at which the flow coefficients can no longer be considered to remain constant are known as "limiting values" (B.S.S. 1042, p. 14).

EXAMPLE C. *Water—Calculation of Flow from Orifice Measurement*

Data :—

Diameter of main, D	= 10 inches
Diameter of orifice, d	= 7 inches
Temperature, t	= 90° F.
Density at 90° F.	= 62·11 lb./cu. ft.
Maximum differential pressure, H ..	= 6 inches Hg (as read)

It is required to determine the rate of flow, W, in lb. per hour and gallons per hour.

Formula (3) is used for this purpose.

$$W = 1{,}273\,\alpha\,d^2\sqrt{H\rho}\ \text{lb./hr.}$$

Orifice ratio d/D = 0·7, whence m = 0·49.
Coefficient of flow α (from Fig. 71) = 0·691.
Rate of flow $W = 1{,}273 \times 0{\cdot}691 \times 49 \times \sqrt{6 \times 62{\cdot}11}$

$$W = 831{,}880\ \text{lb./hr.}$$

Since 1 gallon of water at 90° F. temperature can be found from standard tables to weigh 9·961 lb.,

$$W = \frac{831{,}880}{9{\cdot}961} = 83{,}500\ \text{gallons per hour.}$$

EXAMPLE D. *Coke Oven Gas (or Town Gas)*

Data :—

Maximum rate of flow, V ..	= 250,000 cu. ft./hr. at S.T.P.
Maximum differential head, h	= 2·0 inches w.g.
Diameter of main, D	= 22 inches
Specific gravity (dry gas and dry air), s	= 0·42
Density of gas, ρ	= 0·0342 lb./cu. ft. at S.T.P.

It is required to determine what should be the diameter of orifice, d, for these conditions.

Formula (10) is used.

$$x = \frac{V\sqrt{\rho}}{287{\cdot}4\,\epsilon\,D^2/\sqrt{h}}$$

neglecting ϵ, $x = \dfrac{250{,}000 \times \sqrt{0{\cdot}0342}}{287{\cdot}4 \times 484 \times \sqrt{2{\cdot}0}} = 0{\cdot}235$

From Fig. 73 (*b*) for x = 0·235, d/D = 0·545,
and therefore d = 0·545 × 22 = 12 inches.

An example of the calculation of the density of a gas is given in Chapter IX (pages 199–200).

EXAMPLE E. *Measurement of Water by "V" Notch*

Data :—

Angle of notch, θ	= 45°
Height over notch (inches)	= 6 inches.

It is required to ascertain the flow in lb./hr. when the water is at 140° F.
From Table 61 the flow is 4,146 gallons per hour.
From standard tables the weight of 1 gallon of water at 140° F. is 9·8537 lb.
Therefore, flow of water = 4,146 × 9·8537
= 40,850 lb./hr.

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CHAPTER XI

STOKING AND BOILER OPERATION

The air supply—The management of draught—Passage of air through a fuel bed—Methods and practice of hand firing—Methods and practice of mechanical firing—Water level—Blow-down—Cleaning fires—Banking fires—Cleaning heating surfaces—Carbon losses—Clinker formation.

THE principles of combustion were discussed in detail in Chapters IV, V and VI. In the present chapter the application of these principles to practice is discussed with special reference to the burning of solid fuel. Stoking and boiler operation are here combined since most of the practical operation of the boiler is performed by the boiler fireman.

THE AIR SUPPLY

Adequate draught and proper management of draught are the first requisites for proper combustion. The draught may be secured in one of the various ways mentioned in Chapters VI and XIV, but in whatever way it is obtained it is essential that proper use shall be made of it. The function of draught is to secure a sufficient flow of air through the furnace. It is the duty of the furnace operator to see that the air enters in the right place and flows through the right channels.

The principles of correct draught utilisation have been discussed in Chapter VI. It was there pointed out that between the inlet and outlet of a boiler or furnace system, including any auxiliary plant such as economisers and air heaters, there is a pressure difference which may amount to only a few tenths of an inch. This pressure difference is the motive power which moves the air and gases through the system. Flues and particularly auxiliary plant will absorb part of this pressure difference by their resistance. The firebed will also provide resistance. The importance of keeping the flues and other parts of the plant free from accumulations of dust, soot or other extraneous matter will be self-evident.

Air will flow by the path of least resistance and if leaks exist in the brickwork, air will flow through these rather than through the furnace. Leakage of cold air into the setting is one of the chief sources of fuel wastage, the waste arising from the lowering of the temperature of the gases with consequent reduction in the rate of heat transmission ; it also reduces the available draught. The first step towards successful stoking is to make the setting air-tight. Methods of testing for leakage have been given at the end of Chapter VI. The objective should be to make the whole setting tight by putting it in thorough repair, and to do the work so well that it will stay tight. Nevertheless a thorough test should be made periodically, say once a month even under the best conditions.

The CO_2 content of the gases, as indicated in Chapter IV, is the best general indication of the presence of undue quantities of excess air.

Leakages may occur in the following places :—

(1) *Economisers.* When scraper gear is fitted, a certain amount of air leaks through the holes through which this gear passes. Proprietary air excluders are on the market, but opinion seems to be somewhat divided as to their value.

(2) *External Brickwork.* Any expansion cracks or spaces in the brickwork, or between the brickwork and boiler shell, should be caulked with asbestos rope or pointed with fireclay mortar or caulked with a very thin mixture of fireclay and cotton waste, first pulling the waste apart so that the fibres will be covered with the clay.

In the smaller type of water-tube boiler, cracks may occur in the seating of

the drums or in the front doors. The brickwork is examined by holding against it an open flame as from a candle or oil lamp. Inleakage of air is indicated by the flame being drawn in. Places where this is found to occur are marked with chalk for attention.

The natural porosity of unglazed brickwork may be largely overcome by applying one or two coats of a mixture of hot tar and Portland cement of suitable consistency, or by coats of tar only, or by two or three coats of white-wash. The brickwork of the blow-down recess should be examined.

(3) *Dampers.* Sliding dampers often leak air, but there are various air excluders on the market which largely minimise this defect.

(4) *Ash Doors, etc.* Loss of effective draught may be caused through the door sealing the ash pit or the fire-door having become warped or burned. Damaged doors should be replaced as soon as possible. Furnace fronts, firing, inspection and cleaning doors should not be overlooked.

Leakages have a cumulative effect on draught because, first they increase the total volume of gases to be handled, and second they reduce the temperature of the gases and therefore reduce the chimney draught.

MANAGEMENT OF DRAUGHT

The damper is the control valve of the combustion system, whether the draught be produced by a chimney or a fan. It is essential for proper control that :—

(1) The damper shall be controllable from the firing floor so that the boiler attendant or furnace man does not have to walk some way (or even to climb over ladders, pipes, and so forth) to reach it.
(2) The damper must be controllable inch by inch so that exact setting is possible.
(3) A draught gauge should be so situated that when the damper is moved, the operator can see simultaneously the effect on the draught.

Where an induced draught fan is used, speed control should be adopted, thereby saving fan power, or alternatively the fan discharge damper should be used. All these methods lend themselves to some easily devised form of boiler front control. As an example, a boiler plant may be fitted with economiser swivel dampers provided with an arm on each spindle, the two linked by a flat iron link, a length of $\frac{1}{2}$-inch diameter wire rope passing over suitable pulleys to the boiler front and the dampers counter-balanced. The same arrangement can be applied to induced draught fans. Many existing boiler plants are not fitted with front damper control, but with a little ingenuity this can always be arranged. Without it, the damper is too often neglected.

DRAUGHT GAUGE.

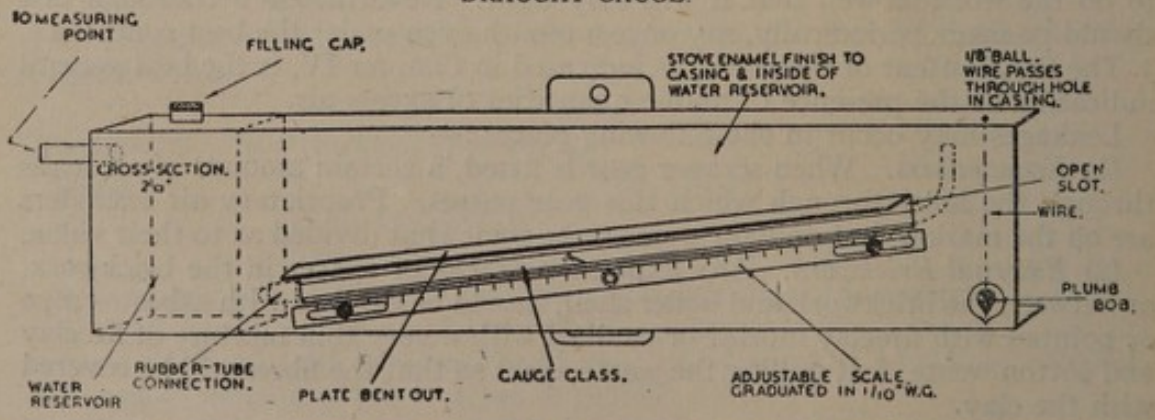

FIG. 80. Simple inclined draught gauge.

Quadrants with pin-and-hole adjustments are much too coarse for correct control. Where the main damper is the only control point and is of the usual rise-and-fall type and very heavy, a good plan is to instal a light swivel damper on the boiler side and use this as a control.

The next point is to have means for measuring the draught. The extent to which the damper is closed is not an accurate measure of the draught, nor of the amount of air passing. Closing the damper by half will not reduce the draught to the same extent. Consequently a draught gauge is essential and the operatives must be taught how to use it. A simple draught gauge is shown in Fig. 80. Connections from all points at which draught is measured should be led to gauges placed near the firing floor so that the operatives can see the draught readily. The position of the gauges should also be arranged so that they can be easily read by the operator while adjusting the draught.

The correct draught for various conditions is a matter for trial, but control to less than 0·1 inch w.g. is necessary. If, for example, a draught at the chimney of 0·7 inch w.g. is correct, a draught of 0·8 inch may mean that the air supply is 12 per cent. greater than it should be.

Draught should be measured at the following points :—

(1) Under the grate or other convenient point when forced draught is used.
(2) In the boiler furnace or as near as convenient : for instance, in the Economic type of boiler the combustion chamber is a convenient position.
(3) Either on the furnace side of the induced draught fan or at the base of the chimney, on the boiler side of the damper.

The correct draught can be established after seeing that all fires are in good condition, the fuel bed of the right depth for the type of fuel being burnt, and the grates well covered from back to front. The draught is then adjusted to be the minimum for the load required. When this point has been established it should be marked on the damper control. This figure will, of course, hold good only for the load operating at the time, and will require adjustment whenever the load varies. The air supply must be adjusted at all times to keep in step with the load.

Other methods of air control, such as closing the ashpit, are crude and ineffective and lead to other troubles, such as damage to firebars and clinker formation.

In furnaces equipped with forced draught, control of combustion will be effected by adjustment of the steam jets or of the damper controls provided, but it must be kept in mind that the chimney or induced draught must also be controlled so that the furnace is under either a very slight pressure or (for boiler furnaces) a very slight suction—in other words, in an almost balanced condition. Controlling the forced draught is only half the problem, and control of both forced and induced draught must go together.

The use of a draught gauge above the firing door (Fig. 81) indicating the draught in the furnace can be illustrated by assuming that a gauge is so connected as to register the draught in this position. If the firebed is allowed to burn too thin, or bare patches develop, the resistance of the firebed will decrease and the gauge reading will fall. If the fire is too thick, or becomes dirty, the gauge reading will increase, since the resistance of the fire has increased. Thus the gauge in this position will indicate the conditions in the firebed. A sensitive gauge reading to, say, 0·01 inch of water is desirable at this point.

The losses that may occur through wear of steam jets used for forced draught production has been discussed in the early part of Chapter VI. The steam jets should never be subjected to the full boiler pressure, and in general a reduction to about 30 lb. gives sufficient pressure to induce the necessary air.

Jets should be replaced when any appreciable wear is found to have occurred. It is advantageous to have the nipples made from stainless or non-corroding steel, phosphor-bronze or other metal which offers a high resistance to erosive wear. Jets must be correctly centred.

A valve should be provided to regulate the steam supply to the jets, and a

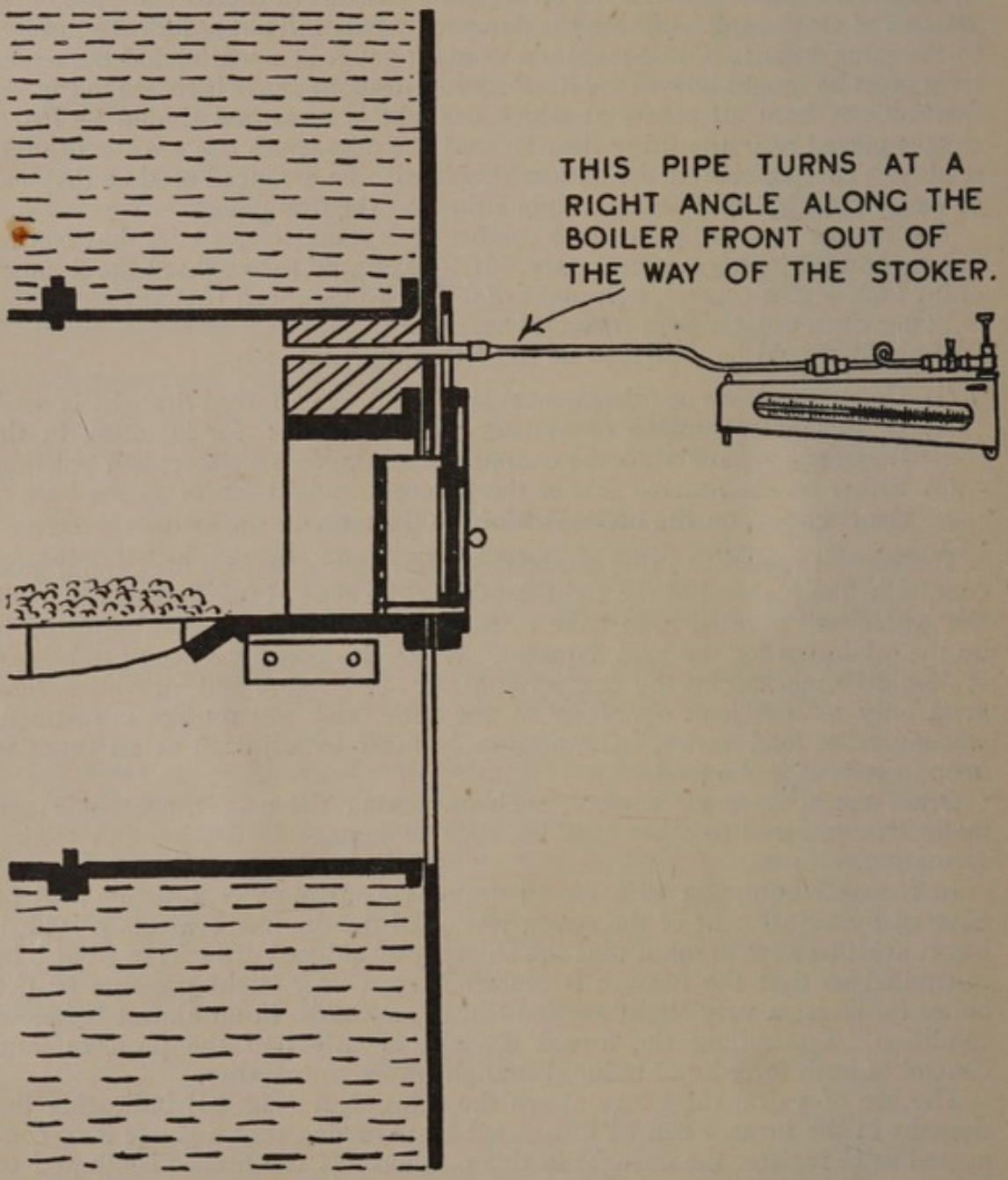

FIG. 81. Gauge in position on boiler front for taking draught over the fire.

pressure gauge fitted between the valve and the jets. The provision of a run of piping in the boiler flue is helpful in obtaining dry or superheated steam, thus improving the working of the jets.

Fans are discussed in Chapter XIV.

THE PASSAGE OF AIR THROUGH THE FUEL BED

When all the sources of loss due to friction and air leakage have been reduced to a minimum, the effective draught available will induce primary air through

the fuel bed and will draw in secondary air above the fuel bed. The air will take the path of least resistance, and if the rate of combustion in the furnace is not adequate the fault may lie in insufficient draught or motive power, or in faulty application of the draught.

If the grate is not covered by fuel at the back—which can be observed by looking under the bars—or if there are holes or thin places in the firebed the air will flow through these paths in preference to taking the path of greater resistance where the fuel bed is of the full thickness. The weight of coal burnt is decreased and the amount of excess air is increased when this happens.

As an example of the effect of maldistribution of the fuel the experience of a works may be quoted at which the steam output was insufficient apparently on account of shortage of draught. The conditions observed were as follows :—

Draught over fire	0·64 inch
CO_2	12 per cent.
Boiler output	10,220 lb. per hour

Correction of the fire conditions, consisting largely in securing an even and well-distributed fuel bed was followed by *reducing* the draught, with the following results :—

Draught over fire	0·32 inch
CO_2	14·5 per cent.
Boiler output	13,000 lb. per hour

It is a good rule to use as little draught as possible for the load to be carried.

Reference to Table 22 (Chapter V.) will indicate the serious effect of reduction in CO_2 content on the fuel consumption, an effect which becomes still more serious the higher the outlet temperature of the flue gases.

RELATIONSHIP BETWEEN DRAUGHT AND RATE OF BURNING

The rate of combustion of a solid fuel will depend upon the rate at which air can be passed through the fuel bed. The size and coking properties of the coal as well as the thickness of the fuel bed will all have an effect upon this.

Some information on the draught required for various types of stokers is given later in this chapter.

RATES OF COMBUSTION

The method of quoting rates of combustion in lb. per square foot of grate area per hour is misleading because of the widely differing calorific values of solid fuels and of the widely differing amounts of primary and secondary air required. A better principle would be to express the rate of combustion of solid fuel in therms per square foot per hour or B.Th.U. per square foot per hour. Since the weight basis is still in common use it is necessary to retain it here.

The limit to the economic rate is dictated by efficiency as reflected in the loss of carbon blown off the grate and up the chimney, loss of carbon in the ash, by maintenance problems connected with the very high temperatures developed at the high rates of burning, and by the use of power involved in the generation of draught.

HAND FIRING

Three methods of hand firing are in use :—

(1) Spreading or sprinkling method.
(2) Side firing or wing firing method.
(3) Coking method.

(1) *Spreading or Sprinkling Method.* This method consists of throwing an even layer of coal practically over the whole of the fire at each firing charge. Usually slightly less coal is placed on the back of the fire than upon the front, so as to prevent the production of smoke due to the cooling of the whole of the surface of the fire by the fresh coal. The back of the fire should never be bare, and it is necessary to use the rake to level up occasionally in order to make up for the lesser fuel charge upon the back portion of the furnace.

With care this method produces more steam than any other, and more uniformity, and should give a higher CO_2, but an unskilled fireman using it will produce more smoke. A good fireman is thoroughly master of the art of using his shovel and can place the coal exactly where he wishes on the grate. Strength is not the only requisite. The shovel should be used with a peculiar motion which only comes with practice. When the shovel is almost at the end of its throw, the extreme end of the handle should be smartly tilted downward, so as to "spray" the coal over a wider area. If the shovel is simply jerked from below the coal upon it, then the coal falls in a heap and gives the fire a very uneven appearance. This uneven thickness does not assist good combustion, and the rake has to be brought frequently into use to level the fire so as to allow the thicker portion to be thoroughly burned. A shovel of suitable size should be provided.

The spreading method is very useful for responding quickly to variations in steam demand, and for dealing with low-volatile fuels.

(2) *Side Firing or Wing Firing Method.* This method consists of throwing the coal first on the right-hand half of the fire and then allowing a period of time to elapse before the left half is fired. The left-hand side should not receive its charge until all the gases have been driven off from the first charge placed on the right-hand half. This method is widely used, and the advantages are that one half of the fire is always in good condition, and it ensures the combustion of the volatile matter given off by the other half. It helps to avoid black smoke.

Rules for Hand Firing by the Spreading and Side Firing Methods. Firing should be done lightly and often, the fire-door being kept open for the minimum time necessary. The firebars should be kept completely covered with coal, particular attention being paid to the back of the grate and the corners, right and left of the fire-door.

A level fire must be maintained, cracks and holes in it being filled with fresh charges of coal. The dampers are adjusted according to the load and secondary air admitted through the air-grids in the fire-doors as required. More secondary air is needed just after a fresh charge of coal than when the volatile matter has burned off. Excessive use of the poker, slice bar and rake must be avoided as it helps to cause clinker; moreover, excess air is admitted when the doors are open while tools are being used.

Dry slacks and finer coals are preferably wetted before burning. The wetting must be uniform and should be done some hours before the coal is used so that the moisture may become thoroughly distributed. If some parts of a heap are wet and others are dry, little advantage is gained by wetting.

(3) *Coking Method.* This method consists of charging the fresh coal on to the front part of the furnace to a depth of about 10 inches, and after allowing time for the gaseous matter to be driven off, pushing it forward over the rest of the grate. With this method very little smoke is produced as the volatile matter is burned when passing over the hot fire at the back, but it may not produce as much steam in a given period as either the spreading or side firing methods.

It is a useful method for small and lightly loaded boilers where the fireman has other duties to perform besides that of stoking.

Thickness of Firebed. The thickness of the firebed must be decided by the

skill and experience of the fireman according to the conditions prevailing. The size and nature of the coal are important ; larger-sized fuels generally require thicker firebeds to prevent air passing through them unburnt.

The amount of fuel burnt, and, therefore, the amount of steam made, is determined by the quantity of useful air passed through a fire.

Too thin a fire leads to holes which cause (*a*) excess air, and (*b*) reduced output of steam.

Too thick a fire prevents air from passing through the fuel bed and so reduces the output.

For each coal there is a most suitable thickness of fuel bed which the fireman can discover by experiment. The way to do this is to fix a draught gauge above the furnace door in the manner previously described, and to determine by trials the relation between draught, fuel bed thickness and CO_2 content.

It must be recognised that the thickness of bed with hand firing cannot be kept constant because the fire will be thickest immediately after firing and thinnest just before firing. It is, therefore, necessary to consider the mean thickness.

When hand firing it is difficult to get good results from a fire with a mean thickness of 3 inches or less owing to the ease with which air can pass through it, and the greater liability to form air holes. This liability will be increased, the higher the draught. Mean thicknesses of 4–6 inches have proved satisfactory. The thinner bed should be used with small coals such as $\frac{1}{2}$-inch slack or pearls, the bed should be thicker with $1\frac{1}{2}$-inch slacks or singles or nuts. It will also be found necessary to carry somewhat thicker beds when the ash content is high—say of the order of 15 per cent.

So long as the fuel bed is thin, little difference in performance will be found between the free-burning coals of groups 1 and 2, and the caking coals of groups 4 and 5. Thick beds will emphasise caking properties and cause the formation of crusts.

In plants where the load fluctuates considerably and there may be sudden demands for more steam, it may be advisable to carry slightly thicker fuel beds in order to have a greater reserve of hot coal on the grate.

Use of Rake and Slice Bar. The rake is used for levelling the fire. It should be used as little as possible as it tends to increase the danger of volatile matter escaping unburned and produce black smoke. It should be passed lightly over the fire surface. It should never be driven down on the firebars and pushed along as this tends to press the soft clinker between the bars, making it difficult to remove ; it also mixes the ashes with the coal, and makes cleaning out longer and harder and more wasteful.

If the fire is heavy with ash, and air is not passing freely through it, the slice bar is run under the fuel and twisted slightly with a side movement. The ash must not be lifted into the upper zone and left there.

Smoke Prevention. To prevent smoke formation, the rich tarry gases evolved from the coal must pass over the incandescent firebed, and enough secondary air for combustion must be admitted through the openings in the fire-doors. This air must meet the volatile gases at a place where the temperature is sufficiently high for combustion to take place.

The coal should be fired in small amounts at short intervals and well distributed over the surface since in this way only small amounts of volatile matter are evolved per square foot of grate area at any one time. This does not apply to the coking method.

When a CO_2 recorder is not available, a light smoke haze from the chimney is generally a good indication of satisfactory combustion conditions, provided the temperature of the fuel bed is high. When burning coke, anthracite or smokeless coals, the chimney is, of course, always clear.

Cleaning Fires. Cleaning takes several minutes, and before cleaning, the dampers should be checked as much as possible to avoid an inrush of cold air when the doors are open. Fires should not be burned down too low before cleaning, but sufficient fire must be left on the grate to start up rapidly after cleaning.

The water level should be so manipulated as to avoid the need for feeding water to the boiler while cleaning the fires, as otherwise the steam will be rapidly reduced. Times of cleaning should be arranged to fit in with periods of low load. Where there is a bank of boilers, fires should be cleaned in rotation.

For cleaning the fires the slice bar and hoe are used. The fires should be cleaned out thoroughly, one fire at a time, all ash and clinker being removed so that they will not fuse to the bars or bridge.

There are two methods of cleaning hand fired furnaces, namely, the side method and the front-to-back method. In the side method good coal is pushed from one side to the other, and the clinker and ash scraped out by the hoe after being loosened, if necessary, by the slice bar. The burning coal from the other side is then removed to the clean side. A few shovelfuls of fresh coal are added in order to have enough burning coal to cover the entire grate when the cleaning is done. The clinkers are then removed from the second half of the grate, after which the fire is spread evenly over the grate and built up gradually with fresh coal.

When starting to clean a thin fire it may be necessary to put fresh coal on the side to be cleaned last in order to have enough burning coal left to start a hot fire quickly.

In the front-to-back method, the burning coal is pushed with the hoe or rake against the bridge wall. It is usually preferable to clean one half of the grate at a time. The clinker is loosened with the slice bar and pulled out of the furnace with the rake and the burning coal spread evenly over the bare grate. If this method must be used while the load is on the boiler, the side method should be employed after the day's run is over so as to prevent a large accumulation of thick and hard clinker at the bridge wall.

Reasonable care should be taken to see that unburned fuel is not drawn together with the clinker and ash.

The clinker and ash should be drawn from the furnace on to an iron or steel barrow, wheeled outside the boiler house and slaked with water. On no account should the ashes and clinker be drawn on to the plates in front of the boiler and slaked there, as this practice leads to corrosion and wasting of the plates and if allowed against the boiler may in time ruin it.

Special care should be taken to see that no coke is discarded with the clinker and ashes. Coke recovered is put back on the fire.

The small coal that falls through the grate bars should be collected and burned with fresh coal.

Dimensions of Grate. Considerable economies can often be affected by using a grate of size suited to the load. There is some difference of opinion as to what should be the maximum length of the grate, some authorities maintaining that the 6-foot grate is too long for the best operating practice, and preferring a 5-foot grate.

It is found, however, that whatever be the initial length of the grate it should be used to the full capacity in order to provide a sufficiently hot fire for adequate radiation heat transmission in the combustion chamber. Economies have followed the practice of shortening the grate by building up the end near the bridge with refractory bricks when a reduced seasonal load requires less coal to be burnt for several weeks.

Hand Firing with Slurry and other Low-grade Fuels. The term "low-grade" fuel may be considered to refer to those products having a calorific value

of between 8,000 and 10,000 British Thermal Units per pound " as received," and includes belt pickings, middlings, duff, fines and slurry.

The behaviour of low-grade fuels varies very much in different types of grates and furnaces, but generally the best results will be obtained with furnaces of the forced draught type. Low-grade coals can usefully be blended with others of better quality.

With duffs, fines and slurries the fires must be raked at frequent intervals in order to keep the fuel bed open and to prevent the fires dying down. With inferior slurries or filter cake the firebed should be of good depth, say 10–12 inches to give the necessary " body " of heat to ignite the fresh raw fuel when fired. Good under-grate air pressure is necessary, not less than 3–4 inches water gauge.

Outcrop Coals. Considerable quantities of outcrop coals are coming on to the market. They vary from low grade to quite good fuels. No hard and fast rules can accordingly be laid down for their treatment.

MECHANICAL FIRING

Mechanical stokers may be grouped under the following heads :—

I. Stokers generally used for shell boilers (including vertical boilers).

II. Stokers generally used for water-tube boilers.

It is only possible to give general information regarding the use and characteristics of mechanical stokers.

I. *Stokers for Shell Boilers.* There are three generic types : the Sprinkler stoker ; the Coking stoker ; and the Underfeed stoker.

(1) *Sprinkler Stokers.* These throw the fuel all over the grate by means of shovels, flippers, or rotary distributors (cf. Figs. 27 and 82). They imitate the spreading method of hand firing previously described. Sprinkler stokers may be, again, subdivided into two classes :—

(*a*) When the coal is spread by a shovel working intermittently, the grate bars generally move, i.e. the grates are self-cleaning. The coal is thrown to different sections of the grate in turn by means of the differential lengths of the cams of the mechanism ; this type also imitates in some measure the side firing system of hand firing (Fig. 83).

(*b*) When the coal is spread continuously by a rotary distributor the grate is generally stationary. In this system the coal is spread over the whole area of the grate evenly and continuously (Fig. 84).

(*a*) *Shovel Sprinkler.* The thickness of the fuel bed should be approximately as follows :—

	Inches
Small slacks (½-inch) and small sized coals (pearls) of relatively low ash content, say under 10 per cent. .	3–4
Larger slacks (1½ inches) and larger sized coals (nuts) of relatively low ash content, say under 10 per cent. .	4–5
Coals higher than 10 per cent. ash content . . .	slightly thicker.

The shovels and springs should be inspected at regular intervals and properly adjusted.

(*b*) *Rotary Sprinklers.* The thickness of the fuel bed should be approximately 2–3 inches with slacks, rising to 4 inches with larger coal.

Sprinkler Stokers in General. It will be noted that the suggested fire thicknesses are less than those used for hand firing, and consequently it is necessary to watch the fire carefully to guard against uncovered or unevenly covered

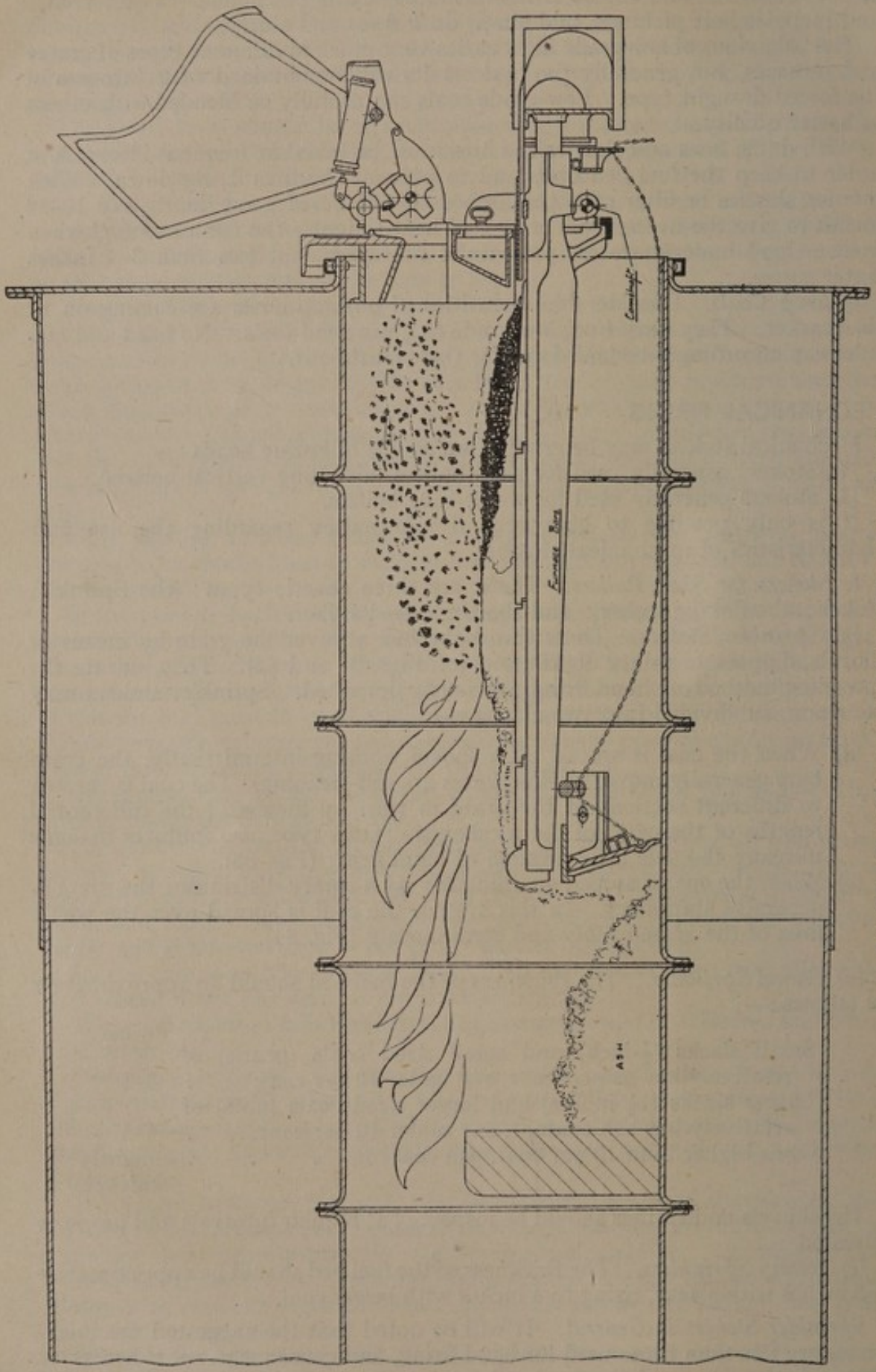

FIG. 82. Sprinkler stoker with variable cam mechanism.

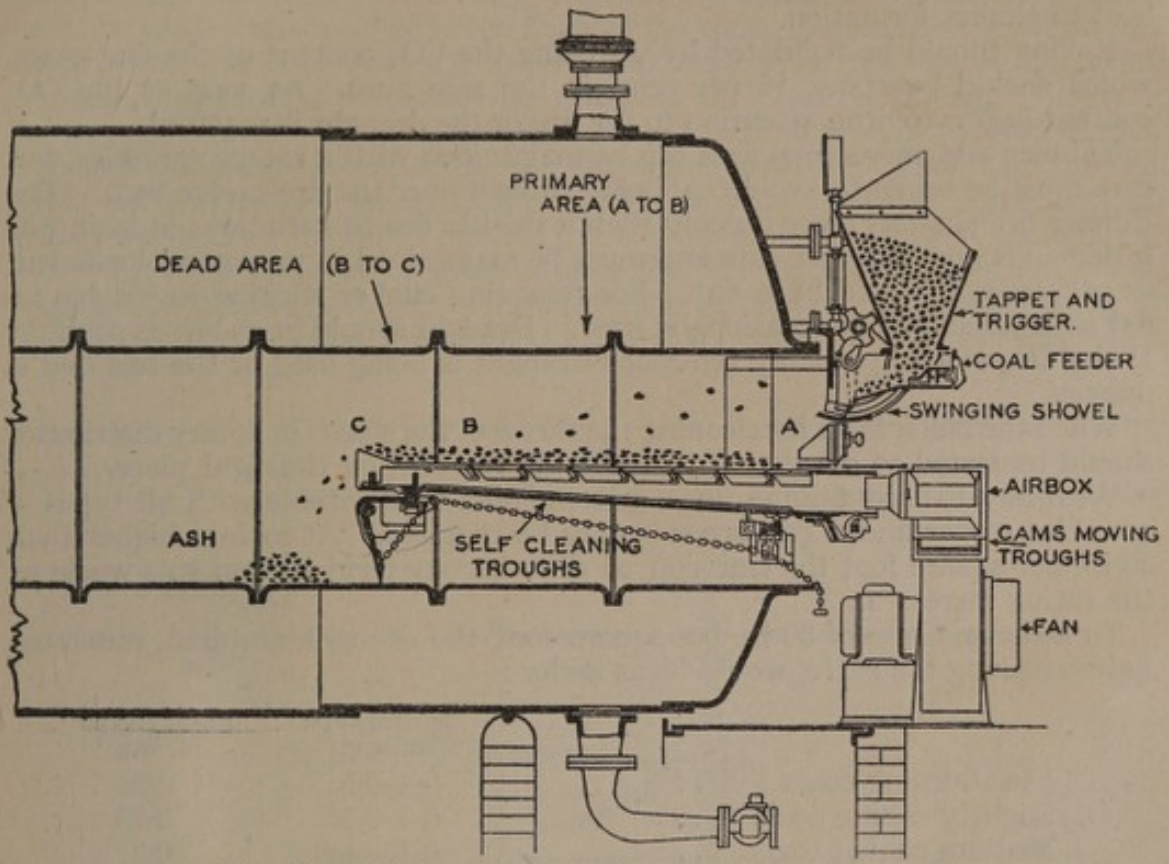

FIG. 83. Moving grate sprinkler stoker, shovel charging.

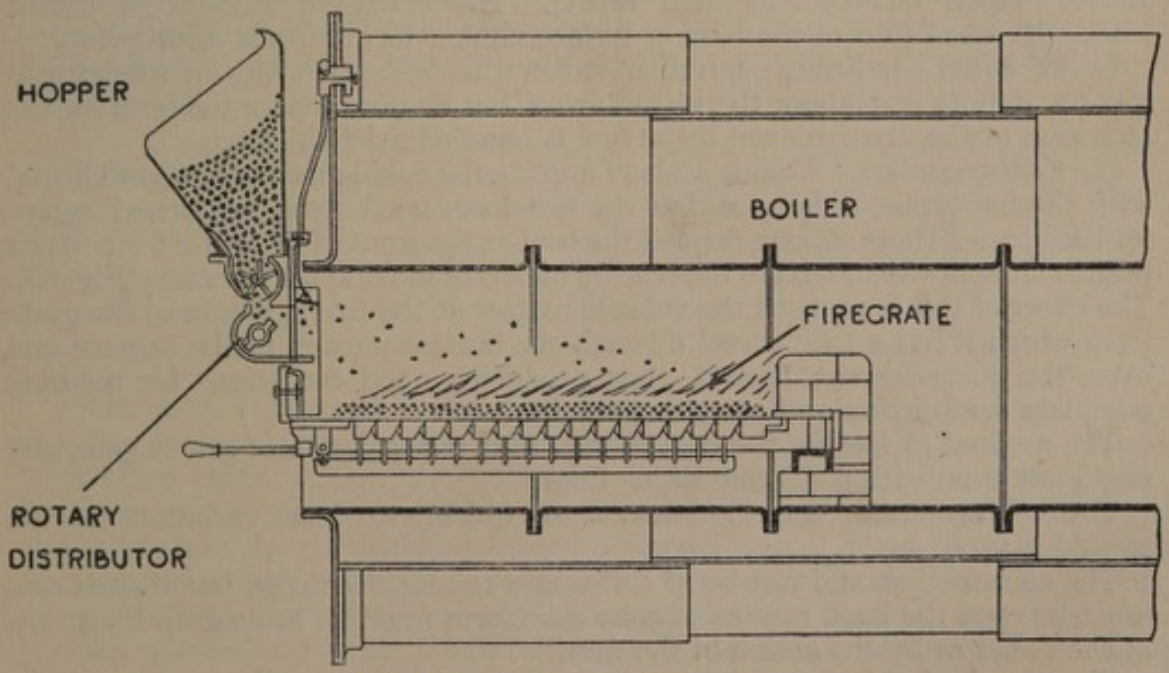

FIG. 84. Rotary sprinkler stoker.

firebars. If these defects occur they can be remedied by a few light strokes with the rake, care being taken not to disturb the whole firebed as this might lead to clinker formation.

Raking should be regulated by watching the CO_2 content of the flue gases, which should be about 12 per cent. in the side flues. As soon as the CO_2 content begins to drop attention to the fire or the draught is required.

Thinner and more even fires can be maintained with a rotary sprinkler, but care must be taken to avoid coal being thrown over the fire-bridge wall. The thinner fire answers more quickly than a thicker fire to variations in load, but it demands much greater care and must be maintained in very good condition, as otherwise it would burn out. For removing clinker sticking to the bars a flat slice bar should be occasionally used. Draught should be as low as possible. If the CO_2 is low, probably too much draught is being used or the fuel bed is uneven.

When the boiler is off for cleaning the throw of the shovel or rotary distributor should be tested to see that it is delivering the coal on the right place.

With any fuel containing fines, grit emission is inevitable with all types of sprinkler stoker and a grit arrester may be required. At ratings higher than 30 lb. per square foot the emission of grits becomes serious and gets worse as the rating increases.

To achieve a rate of 30 lb. per square foot the draught required, measured halfway along the grate, would be as under :—

	Shovel type in. w.g.	Rotary type in. w.g.
Free-burning coals	0·4–0·5	0·25
Slightly caking coals	0·4–0·55	0·25
Medium caking coals	0·45–0·6	0·3
Strongly caking coals	0·65–0·85	0·3

There seems to be little difference in performance with this type of stoker on free-burning and caking coals, or between slacks and sized coals, though slacks require greater draught for the same burning rate. Both give similar CO_2 content in the flue gases and similar output of steam, but there is a reduction in efficiency with fuels containing fines on account of grit emission as mentioned above, which increases at high rating. The excessive amount of steam frequently used to cool the bars or reduce clinker formation is unnecessary.

Under equal conditions, sprinkler stokers, as with all types of mechanical stokers, require less labour than hand firing, but do not obviate the need for the attention of the fireman even if the fuel is handled mechanically.

(2) *Coking Stokers.* Coking stokers imitate the coking method of hand firing, with the important difference that the fire-doors need never be opened except for banking. These stokers deposit the coal at the front of the grate from which position it is slowly moved to the rear by means of moving bars or rams (Fig. 85). The effect of this is to distil the volatile matter at the front portion of the grate from which it has a long travel through the hottest portion of the furnace and over the incandescent firebed, thus producing good conditions for securing complete combustion and obviating smoke.

The method of feeding the coal is such that the emission of grit is generally negligible even with coals containing fines.

The coking stoker does not answer so quickly to load variations as the sprinkler type.

The coal used should not be of a strongly caking character, but if such coal must be used the hard masses of coke that form must be broken up by means of the poker or by the action of the ram.

The standard thickness of fire is 12 to 14 inches at the front of the grate,

tapering toward the back (Fig. 85). Care should be taken to see that the fuel covers all parts of the grate. Bars too thinly covered at the back will admit cold air, while if the fuel bed is too thick at the back, live coal only partly burnt may be carried over with the ashes. The cure when the end of the grate becomes uncovered is to increase the coal feed or reduce the draught ; the cure for excessive carbon in ashes is the reverse ; which method is adopted in each instance depends, of course, on the steam requirements. The back of this grate can be seen from the underside of the bars while the stoker is in operation, and if the bars are found to become red hot the probable cause is some factor which prevents sufficient air from passing through the bars to keep them cool, such as insufficient draught or the formation of clinker. The coal should be completely burnt before it falls off the end of the grate into the ash pit.

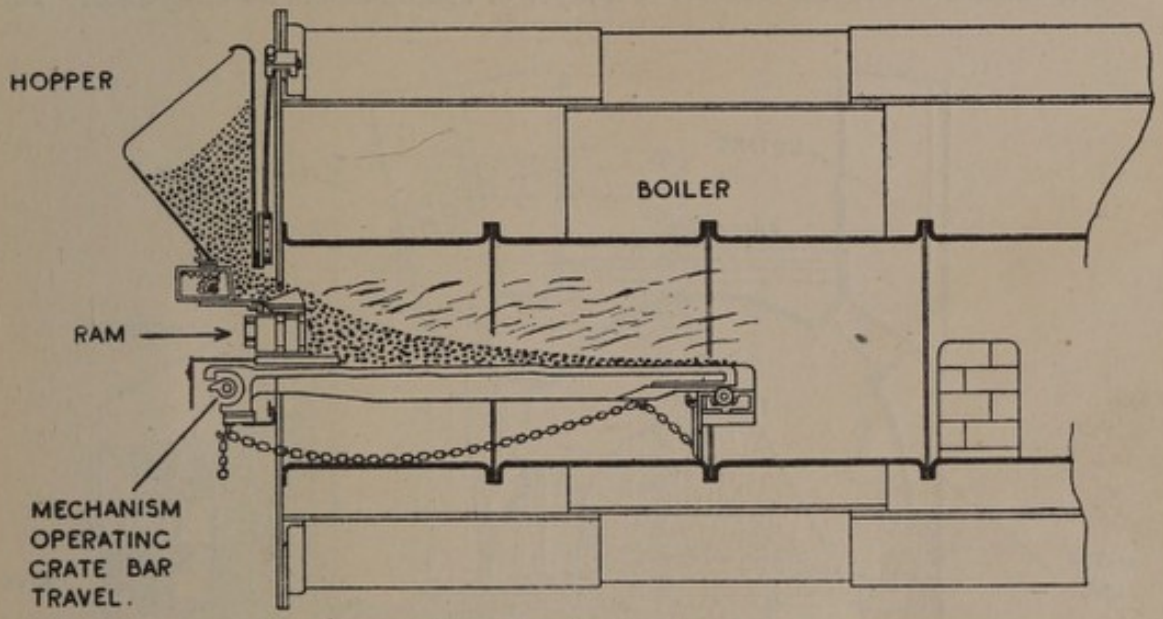

FIG. 85. Coking stoker.

This type of stoker requires more draught than the sprinkler, because of the greater thickness of firebed. The approximate draughts required in order to burn a variety of bituminous coals with induced draught at the rates stated are given below.

Burning rate lb./sq. ft. grate/hr.	Draught in. w.g.
20–25	0·25–0·3
25–30	0·4 –0·5
30–35	0·5 –0·6
35–40	0·7
40–45	0·8 –1·0

Apart from occasional slicing through the fire-doors, which are not opened for this purpose, this type of stoker requires very little manual attention ; with some coals, apart from removing ashes and, with poor quality coals, breaking up the clinker, no labour is required. This, of course, does not mean that attention from a skilled operator is unnecessary. From the labour angle this type of stoker is claimed to require less manual attention, provided reasonably good coal is available for full load requirements, than the sprinkler type.

(3) *Underfeed Stokers.* Underfeed stokers have no counterpart in hand firing. They fall into two types :—

(*a*) The pot type (Fig. 86), the larger sizes of which are also used for small water-tube boilers up to 25,000,000 B.Th.U. per hr. ; the smaller sizes are used for vertical boilers and C.I. sectional boilers (Chapter XXIV).
(*b*) The grate type (Fig. 87).

The coal is fired mechanically from the bottom of the retort by means of rams or screws (cf. Fig. 29), the air for combustion being supplied by forced draught fans. As the coal rises in the retort, volatile matter is driven off and when conditions are right burns on the top of the fire with a short, intense flame.

In the pot type the ash falls over the brickwork down the sides and back ; in the flue type it falls towards the sides and remains on the grate. In both types it is withdrawn by hand.

The surface of the fire consists of a layer of incandescent burning coke. The

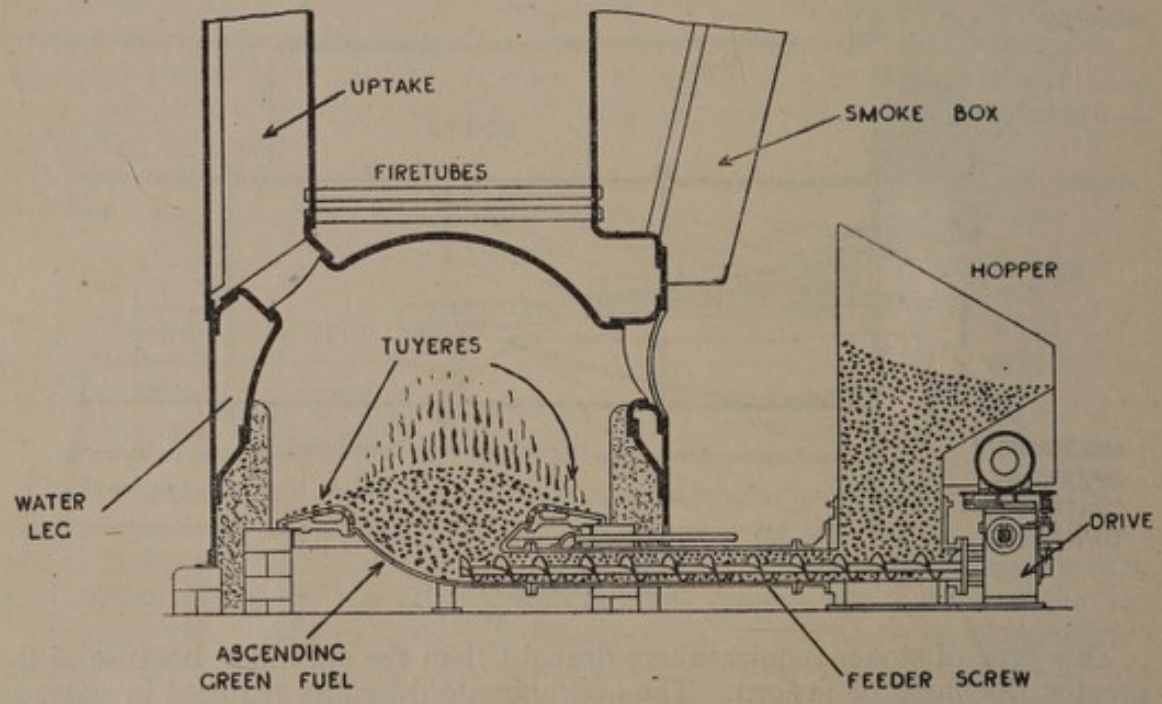

FIG. 86. Underfeed stoker—pot type.

air for combustion is introduced near the point where the fuel emerges from the retort or pot. As the volatile gases are liberated they are thoroughly mixed with air. The mixture passes up through the fuel bed where it encounters higher temperature zones and complete combustion of the volatiles occurs when the mixture passes through the intensely hot coke on the surface. All the air is introduced through the fuel bed from below. If smoke should be produced the air supply must be adjusted according to the analysis of the flue gases ; high CO_2 means that more air is needed, and low CO_2 that the air supply should be reduced.

The adjustment of the stoker must be such that coal emerges from the retort throughout its entire length (flue type), or over its entire area (pot type) ; when properly adjusted the fuel bed is automatically maintained clean as the ash is floated away from the retort.

These stokers are adapted for burning free-burning bituminous coals and semi-anthracites, but are more selective than the overfeed types.

II. *Stokers for Water-tube Boilers.* For water-tube boilers stokers fall into three main classes :—

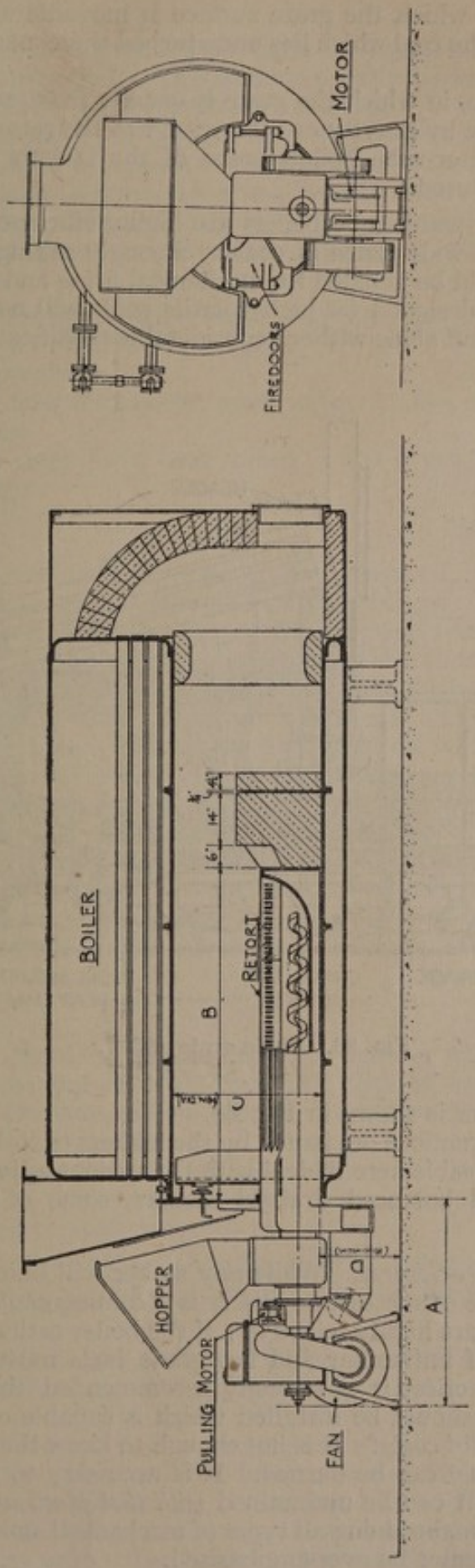

FIG. 87. Underfeed stoker—grate type.

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(*a*) Progressive, in which the grate surface is movable and travels at the same speed as the coal which lies undisturbed thereon, e.g. the chain-grate type.
(*b*) Non-progressive, in which the grate is usually fixed and the coal travels over the surface by gravity or otherwise, e.g. the retort type.
(*c*) For smaller water-tube boilers some of the stokers designed for shell boilers are also used.

Modern stokers for water-tube boilers are highly efficient and very flexible, taking, as they do, a wide range of coals. 'Strongly caking coals with ash of low fusing point should be avoided for mechanical firing and obviously a stoker and setting which is designed for high-volatile coals will not operate satisfactorily on anthracite duff alone without considerable modification to the machine and the setting.

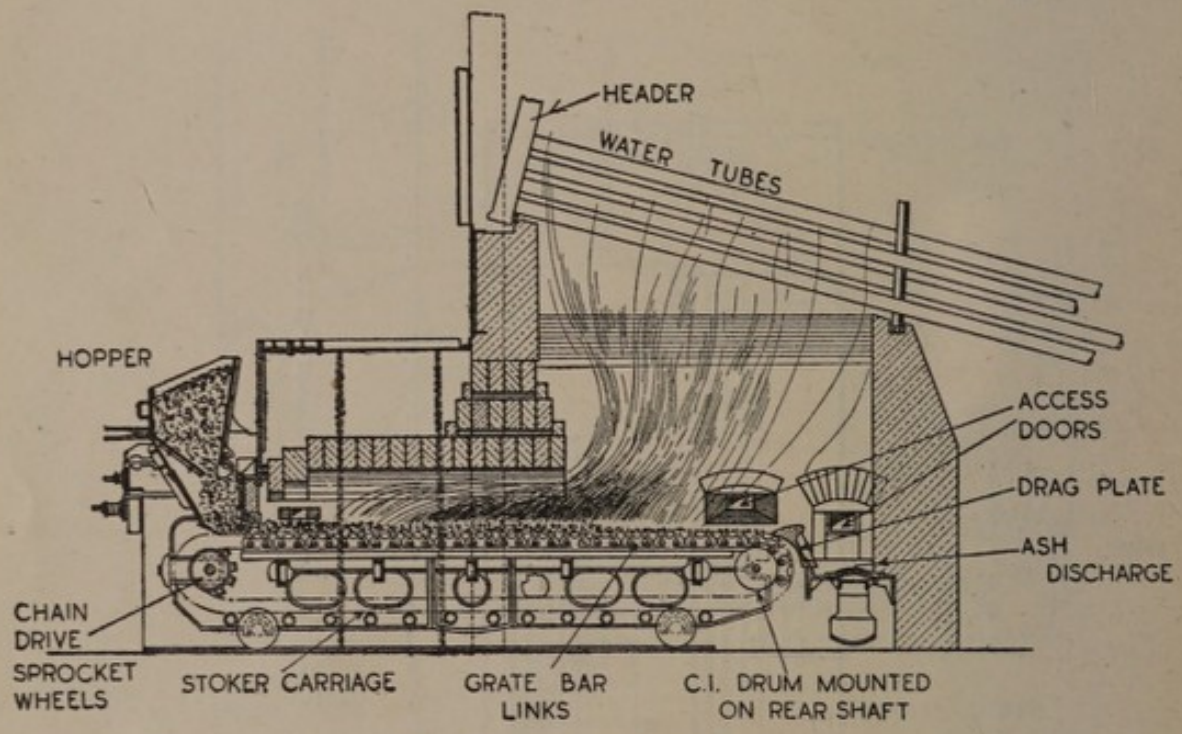

FIG. 88. Chain grate stoker.

A chain-grate stoker is shown in Fig. 88.

Full operating instructions are issued by the makers of stokers for water-tube boilers and it is impossible here to deal with the various adjustments for retort, natural, induced and balanced draught stokers, some of which are of the compartment type.

Mechanical Stokers—General. While any stoker will burn practically every coal with some degree of success, no stoker is a commercial success with every coal. Many of them are highly selective ; if the boiler rating is to be obtained, so long as the use of unfamiliar and low-grade fuels must be regarded as a feature of boiler practice, it is strongly recommended that when adopting machine firing plant should be installed which is capable of dealing with the widest possible range of coals. It is not enough to know that on such and such a stoker low-grade fuel *can* be burned. It is necessary to know whether the required steam output can be maintained *with that plant and fuel.*

The importance of maintaining all types of mechanical stokers well lubricated and in good repair cannot be over-emphasised.

WATER LEVEL

If the water level is maintained too high the steam passing from the boiler is liable to carry water with it. This water is usually discharged from separators, traps and drains and therefore represents a wastage. It is, therefore, important to keep the water as nearly as possible at working level by careful regulation of the feed pump. Excess of impurities in the boiler water aggravates priming.

Particular care must be taken never to allow the water level in the gauge glass to sink out of sight or get dangerously low. If this should happen and no water is blown out when the bottom gauge glass drain cock is opened, prompt action must be taken :—

(*a*) Shut forced draught dampers and stop fans, but it may be advisable to leave the outlet boiler damper open.
(*b*) Close all stop valves.
(*c*) Smother the fires with earth, sand, ashes or even coal ; as a last resort, draw the fires.
(*d*) When these steps have been taken, warn everyone in the vicinity of possible danger.

BLOW-DOWN

Whilst a certain amount of blowing down is essential, if this is excessive or if the blow-down valves leak, avoidable heat losses occur. The amount of water rejected should therefore be carefully controlled and reduced to the minimum necessary to maintain the boiler water sufficiently free from impurities.

The pipes leading from the blow-down valve should be examined to see that no leakage is occurring when the valves are closed. A general indication that the boiler should be blown down is the surging of the water in the gauge glass.

Let x be the gallons of feed water added per day per gallon of boiler capacity,
a, the total dissolved solids (T.D.S.) in grains per gallon in the feed water,
y, the gallons of water blown down per day per gallon of boiler capacity,
and b, the T.D.S. in this blow-down.

Since to maintain the solids in solution at a predetermined level the amount of dissolved solids removed from the boiler at blow-down must equal the amount of dissolved solid added per day in the feed water,

$$ax = by, \text{ whence } y = \frac{ax}{b}.$$

Clearly, the total dissolved solids should be kept as high as is safe to avoid priming and the amount of blow-down will depend on this, on the rate of steaming and on the composition of the feed water (including any returned condensate).

BANKING FIRES

Fires are banked for two purposes :—

(1) So that the boiler can be restarted and brought up to full steaming quickly.
(2) To give a continuous, though small, supply of steam over a period without attention.

The object must be to cause all the small amount of air that enters the boiler to pass through the hot part of the firebed so that it consumes the coal at a slow rate regulated by the main damper. Any air which does not pass through the hot part of the firebed comes in cold and leaves hot. The heat it acquires has been stolen from the boiler and is a direct waste of coal.

The bulk of the fuel should first be made red hot. The back of the grate is then cleared and all the good, live fire is pushed on to the back against the brick bridge. This fire is then built up with fresh fuel, any clinker on the

remainder of the grate being loosened, but left lying on the grate with the ash to help to prevent the passage of air through the front and middle of the grate.

All dampers are then closed to the minimum necessary to prevent the escape of fumes into the boiler house. Ash-pit doors and secondary air openings are also closed as far as possible.

The water in the boiler should be left 3 or 4 inches higher than the working level and care taken that the check valve, blow-off cock and water gauge drain cock are tightly closed, as otherwise the boiler may empty and damage result. Occasional inspection is desirable to ascertain that the water level is correct as shown in the gauge glass.

When restarting after the fires have been banked or after the dampers have been closed for any length of time, great care should be taken before opening the fire-door or breaking the fire to raise the damper and open the air regulators on the fire-doors. This is very important as it allows any unburnt gases which may have accumulated to be swept out of the boiler and thus avoids the risk of explosion and back-firing which at least are a danger to the fireman.

The clinker is then cleaned out and the live fuel drawn forward from the back. The fire is then gradually built up to the required thickness by firing lightly and often.

CLEANING HEATING SURFACES

The smoke tubes of tubular boilers should be brushed out every day whenever possible to ensure the highest evaporation and the best efficiency.

To do this by hand methods the dampers should be almost closed and the doors of the boiler house shut if necessary so that cold external air may not impinge upon the hot end plate when the smoke-box doors are opened. The smoke-box doors may then be opened, one at a time, and the tubes swept with tube brushes while the boiler is under steam.

With good coal and good firing, tube cleaning can be done less frequently, but the tubes must be kept clean.

To avoid this hand labour, steam jet blowers are sometimes fitted whereby the cleaning is done by well-directed steam jets, and this system does not require the smoke-box doors to be opened and interferes in no way with the operation of the boiler.

Combustion chambers of return tube boilers should be cleaned out once a fortnight or as required.

In Lancashire and other brick-set boilers the flues and the outside of the shell within the flues should be swept clean as required and at regular intervals.

FIREBARS

The space occupied by the solid portion of the firebars is not available for the passage of air, consequently firebars should be designed to give the maximum effective air space, and this should be as evenly distributed over the grate as possible. This does not preclude a gradual reduction from front to back, which is sometimes desirable.

Other essentials of firebars are :—

(1) Resistance to overheating and, therefore, avoidance of constant renewals. This is promoted by the use of special types of cast iron of a heat-resisting character.

(2) Such arrangement of the bars as will reduce to a minimum the amount of small coal falling through the grate.

Overheating can be reduced by so designing the bars that the heat which they receive from the fire is dissipated as rapidly as possible. This can be done

by reducing the area of absorption to a minimum and increasing the area of dissipation to a maximum, the area of absorption being that amount of metal directly exposed to the heat of the fuel bed, and the area of dissipation being that amount of metal exposed to the cooling effect of the incoming air. For example, a solid bar with an upper surface 1½ inches wide and only 1½ inches deep is more likely to become overheated and warped than a bar with an upper absorption surface of ⅜ inch and an elongated dissipating surface up to 5 inches, with or without additional heat dissipating surfaces in the form of fins.

The deep type of bar to which reference has just been made has the additional advantage that the space between the bars (available for air ingress) may be reduced and thus help in reducing to a minimum the amount of small coal falling through the grate. At the same time the number of spaces—although individually of lesser opening : say ⅜ inch for wide bars and ⅜₁₆ inch for narrow bars—is increased, resulting not only in a total overall increase of air space, but in much better distribution. This type of bar may be more expensive in initial cost.

A high burning rate per square foot of grate area tends in general to keep the bars cooler, because whilst a higher burning rate per square foot increases the temperature of the fuel bed, and therefore the heat received by the bars, this increase is relatively small compared with the greater cooling effect of the increased quantity of air sweeping the bars (cf. " Firebed cooling," Chapter VI). This would not apply to the same extent when preheated air is used. A high ash content in the coal will also tend to keep the bars cool. Conversely overheating and maintenance charges increase rapidly with " ultra-clean " coal.

Attempts to overcome some of the disadvantages of the wide surface type of stationary firebars have been made by making them hollow.

Where stokers are designed to work with moving bars—which mechanically reject the ashes into a back ash pit—the lower limit of width of bar is limited by this width being able to provide a cam strong enough to work with the propelling mechanism. Where this type of moving bar is employed its functions are said to be enhanced either by placing notches in the bar to give a more positive action to the ash removal, or by giving them a reciprocal action which is particularly useful in breaking up a strongly caking fuel. The correct relative movement of the bars may be an important factor in satisfactory combustion.

Forced draught can be applied either by closing the chamber under the bars or by the use of troughs into which either fixed grids or small loosely fitting bars are inserted.

CARBON LOSSES

In the operation of any stoker apart from reaching the best combustion conditions as previously described, losses must be reduced to a minimum. These losses are :—

(1) Loss of carbon in grit emission.
(2) Loss of carbon in riddlings through grate.
(3) Loss of carbon contained in the ashes.

(1) *Loss of Carbon in Grit Emission.* This source of loss is of relatively recent occurrence and arises out of the modern practice of burning coal at high rates of combustion per square foot of grate area. Apart from the nuisance which it causes to the surrounding property and actual danger in the damage to eyes of personnel and parts of machinery, it involves what may be an appreciable loss of carbon.

The amount of grit emitted depends, in the first instance, on the rate of combustion, i.e. the intensity of the draught, and it may be further influenced by various other factors, as follows :—

(*a*) The size of the coal.

The emission will be greater from slack than from sized coal.

(*b*) The type of coal.

The emission will generally be greater with a free-burning coal than with a caking coal.

(*c*) The type of stoker.

The emission will tend to be greater with the sprinkler type of firing than with the coking type, because with the sprinkler type the very fine particles are more readily caught up in the air stream as the coal is flipped on to the grate. Some alleviation may be obtained by thoroughly wetting the coal, which causes the smallest particles to cohere; this also applies to hand firing. The quantity of water added should be just snfficient to make the fuel "ball" when pressed in the hands.

(*d*) The operation of the stoker.

As far as possible disturbance of the fire should be avoided as this must necessarily tend to increase grit emission.

The emission of grit will tend to dirty the boiler and reduce the periods between cleaning. It will also erode induced draught fans.

(2) *Riddlings.* In all grates there is a tendency for small coal to fall through the bars. This tendency increases with free-burning coals and coals with a high percentage of fines. It is particularly prevalent in stokers with moving parts. It may be due to the bars being too widely spaced for the fuel burned or the actual movement of the bars. The riddlings should be returned to the coal in as uniform quantities as is convenient.

Excessive riddling may occur with chain-grate stokers if the lateral working clearance between the links is allowed to become too great. The clearance should be maintained at about the design figure, which with one type of chain-grate stoker is approximately $\frac{1}{8}$ inch per foot width of chain.

Again, with some types of chain-grate stoker, the bottom section of the hopper front consists of a hinged drag plate which rests on the chain. When in good condition this makes contact with the chain where the links have closed again after passing round the front sprocket. In the process of time this plate is liable to wear and assume a steeper angle, the bottom edge of the plate being advanced to the point where the break of the chain is not covered, so that small coal falls through direct into the riddling pit. These plates should always be maintained in good condition.

(3) *Loss of Carbon in Ashes.* The separation of clinker and ashes from the fuel bed invariably involves the entrainment of a certain amount of coke which, under bad conditions, may amount to 5–10 per cent. of the coal burned. The amount will tend to be greater when coals of high ash content are used.

The amount frequently depends on the skill of the firemen. A certain loss is unavoidable, and steps must be taken to recover as much as possible. To obtain a rough idea of its magnitude, an average sample of ash and clinker should be quenched in water immediately after leaving the fire, and the pieces of coke picked out by hand; the proportion may be more exactly ascertained by chemical analysis.

With a properly operated furnace using suitable coal, the ashes would not contain excessive amounts of unconsumed carbon. The first step, therefore, should be to investigate combustion conditions at the furnaces in order to reduce carbon losses to reasonable proportions. The chief causes of unconsumed carbon in ash are given below with some notes upon each of them.

(1) The coal is unsuited to the grate and the furnace.

Too frequently this is regarded as the prime cause of the trouble, and other causes are not investigated. There are circumstances—as for example when

the coal is too small or too large or contains quite excessive amounts of ash—under which the coal is the cause of the trouble. Generally, however, means can be found for burning most coals with reasonably satisfactory results.

This applies both to hand firing and mechanical firing, but more particularly to mechanical firing.

(2) High content of ash fusing at a low temperature.
(3) Too high a fuel bed temperature fusing ash to clinker.
(4) Too frequent cleaning of fires due to clinker.

Causes (2), (3) and (4) all arise from clinker formation. If clinker formation is excessive carbon particles will be enclosed in the clinker and thus cannot burn. The remedy is to prevent the clinker from fusing. Clinker formation takes place at the zones of highest temperature. When the fuel passes down the fire undisturbed at the zone of highest temperature so much unburnt material will be present that the ash particles will not be in contact with one another to any considerable extent ; large masses of clinker cannot normally form under these conditions unless the ash is very fusible. If the fuel bed is disturbed in such a way as to bring the bed of ash lying on the grate upwards into the combustion zone, conditions are immediately created which favour clinker formation. The fuel bed should not be poked or disturbed in such a way as to create these conditions.

Clinker is more readily formed from smalls than from larger sizes of graded fuels, because smalls are massed together and the ash released is in closer physical contact. The size of the ash particles from small coal is also smaller, which again leads to easier fusion. The character of the ash is therefore as important as the quantity formed.

If the ash fusion temperature is so low that even with the precautions just mentioned clinker is formed, a remedy that has been found satisfactory is to keep the ash bed cool by admitting steam or very finely atomised jets of water under the grate. The cooling medium should be focussed upon the hottest zones where trouble is likely to originate (cf. Chapter VI).

(5) Excessive use of the poker, slice bar and hoe, when hand firing.

The comments made under (2) above will illustrate the importance of not disturbing the firebed too frequently and thus bringing the smaller pieces of coal on to the grate before they have been burnt.

(6) Careless stoking resulting in the fuel bed not being kept level.

This refers primarily to hand firing, but mechanical stokers should be watched to see that defects do not occur in the fuel bed.

(7) Firing coal on to bare spots on the grate.

Items (6) and (7) are special cases of the same cause, bare spots on the grate being the extreme condition arising from an uneven firebed. Obviously, if fresh coal is fed directly on to an uncovered grate the grate will serve as a screen through which the smaller pieces of coal will drop.

(8) Furnace volume too small for rate of combustion desired.

This difficulty should not arise with furnaces or boilers operated at the load for which they were designed. The difficulty arises principally when the furnace is pushed to give an excessively high output.

When attention to operating details fails to reduce the quantity of carbon in ashes to a negligible amount, consideration should be given to recovery for re-burning.

The following are methods of recovery :—

Hand Picking. The amount and size of the coke pieces frequently warrants hand picking.

Forking. Where supervision is good, large pieces of coke are rare, but the smaller material may contain a lot of carbon. The residue should be picked

over with forks having ¾-inch spaces between the prongs, the undersize being returned to the fire and the oversize being discarded. In larger installations this may be done mechanically. Some ashes contain their carbon in the form of large material and others of small material. The distance between the prongs of the fork should be selected in accordance with the characteristics of the material to be treated.

More Elaborate Recovery Methods. Some of the more elaborate methods are based upon the principles of coal washing already described in Chapter I. The ashes and clinker are broken up and the carbonaceous portion is floated off either in a clay-water suspension or in calcium chloride solution or in mechanically agitated water. A good deal can frequently be done by floating in water if the clinker is dry, since the highly porous particles of coke are generally sufficiently light to float in water for a short time until the water has entered the pores. Magnetic methods are also used on the principle that a good deal of the iron and iron oxides present in the clinker is magnetic, whereas the coke is not.

FORMATION OF CLINKER IN FUEL BEDS

The formation of clinker in fuel beds and the methods for the alleviation of this difficulty have been discussed at some length in Chapter VI. It should be noted that it is the character of the ash and not its total quantity that determines the difficulties arising from the nature of the clinker. When a low ash coal gives a bad clinker the trouble can sometimes be remedied by increasing the ash content either by adding shale or sand. It is, for example, quite customary to add broken firebrick to locomotive fires.

CHAPTER XII

PULVERISED FUEL

Equipment and application :—Pulverisation—Extraction fans—Distribution systems—Burners—Application to boilers—Cement—Metallurgical applications in the ferrous and non-ferrous industries—Choice of fuel—Furnace design.

Efficient use of P.F. in boilers :—Differences between operation on pulverised fuel and firing on grates—Bogey results—How to survey a pulverised fuel boiler installation.

INTRODUCTION

THE combustion of coal in the form of nuts, etc., differs from the combustion of gases in the extent of the surface of contact. A mixture of a combustible gas with air achieves molecular contact so that the surface exposed is the maximum and combustion can be so rapid as to be

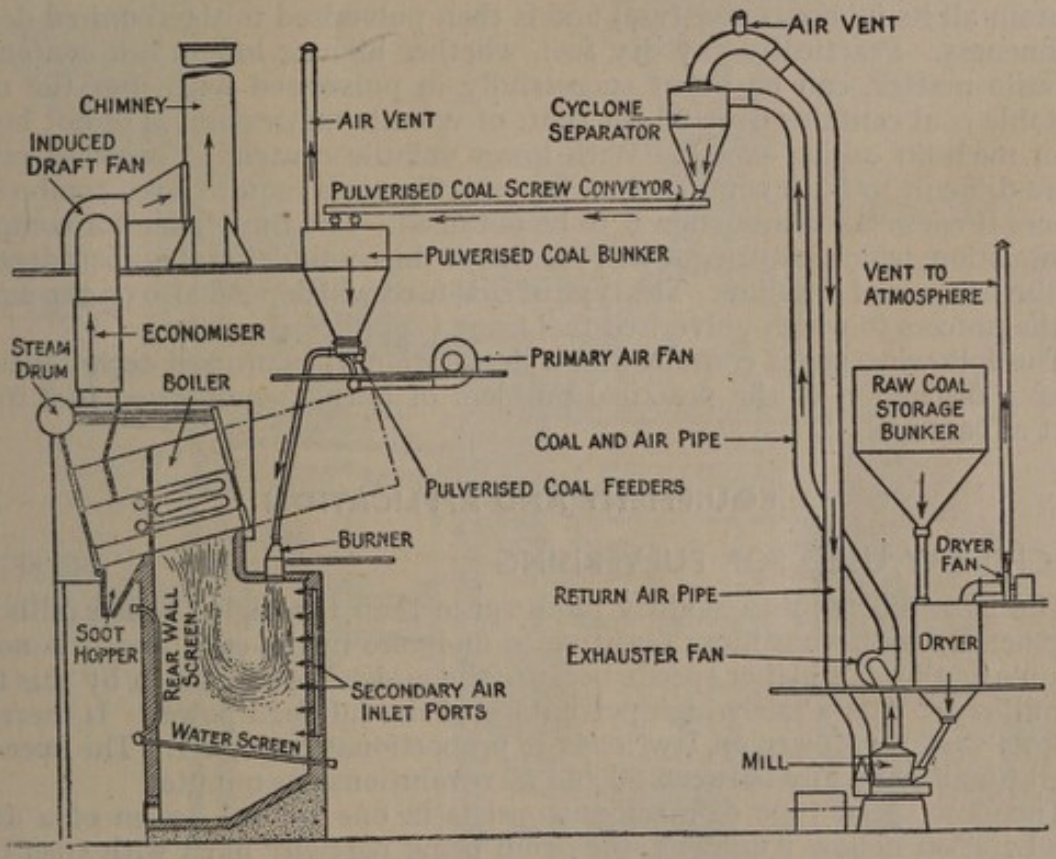

FIG. 89. Typical bin and feeder system.

(Reproduced by permission of the publishers (Edward Arnold) from "Fuel—Solid, Liquid and Gaseous," by J. S. S. Brame and J. G. King)

explosive. The surface of large coal and even of slack is limited by comparison so that combustion is slower. Liquid fuels are fired in a finely atomised form in order to secure the greatest possible surface of contact between fuel and air.

A similar method can be applied to coal (Fig. 89). By reducing the coal to a fine powder, which is then carried forward by an air blast, it is possible to obtain perfect combustion with absence of smoke when using only a small excess of air. High temperatures and high efficiencies result from this practice when correctly applied.

The earliest attempt to utilise pulverised coal was made in 1818. Various attempts were made between then and 1895, when the first really successful

application was made, this in the firing of cement kilns. Little progress was made following this till about 1918. From then on progress was rapid. The increase in the use in the 10 years following amounted to over twentyfold. By this time it had been demonstrated that all classes of solid fuel could be successfully pulverised and burned in that form.

Combustion of coal takes place in two steps, the first being the separation of the coal into gaseous hydrocarbons and coke, the second being the combustion of these fuels.

In the ordinary coal-fired furnace the volatile matter is driven off and burned as the coal is fed on to the grate, and the coke left behind then combines with the necessary air. In using pulverised coal the fuel, mixed with the correct amount of primary air for ignition, is injected into the furnace in a finely divided state. Under this condition the distillation of the volatile matter takes place very rapidly, and owing to the small size of the solid particles of coke for all practical purposes the result is one stage combustion.

The coal must be freed from most of its surface moisture (though it may still contain all its inherent moisture) and is then pulverised to the required degree of fineness. Practically any dry fuel, whether high or low in ash content or volatile matter, can be burnt successfully in pulverised form, but the most suitable coal contains over 20 per cent. of volatile matter and is of not higher than medium caking power. With lower volatile content it becomes rather more difficult to burn completely : low-volatile coals require large combustion spaces if complete combustion is to be obtained. The time taken for complete combustion, which in turn governs the size of the combustion chamber, depends on the fineness of grinding. The type of coal used will depend also on the nature of the process to which pulverised fuel firing is applied (see later).

The following pages contain first a description of plant and technique and later a discussion of the practical problem of burning pulverised fuel to the best advantage.

EQUIPMENT AND APPLICATION

MACHINERY USED FOR PULVERISING

The ball mill requires slightly more space than the higher speed mills. It is generally somewhat more sensitive to moisture in the coal, and it is not so flexible as those of higher speeds because the no-load power taken by this type of mill represents a fairly high percentage of the full-load power. It therefore follows that the power on low loads is proportionately higher. The speed of operation is generally between 20 and 30 revolutions per minute.

The slow speed type of machine consists in one typical design of a drum revolving on hollow trunnions, the drum being partially filled with steel balls of various sizes. The drum is revolved at the correct speed to lift the balls almost to the top centre, from which they drop in a cascade on to the (pre-crushed) coal to be pulverised. The coal is fed through one of the hollow trunnions and delivered from the hollow trunnion at the opposite end.

The power consumption of this type of mill, together with its exhausting fan, when grinding to the following specification :—

	Per cent.
Residue on 200 B.S. sieve	15
Residue on 100 B.S. sieve	3–5
Residue on 30 B.S. sieve	·05–·1

and working at its full rated load, is in the neighbourhood of 24 kWh per ton of coal pulverised, and varies according to the size of the machine, the type of coal being pulverised and the moisture content of the coal.

The medium-speed machines, i.e. those operating between 800 and 1,600 revolutions per minute, of the hammer type, and also of the combined hammer and moving peg type, owing to their design and speed, are more capable of dealing with coals of a higher moisture content. A drying system is often incorporated as an integral part of the machine. The necessary heat is either drawn

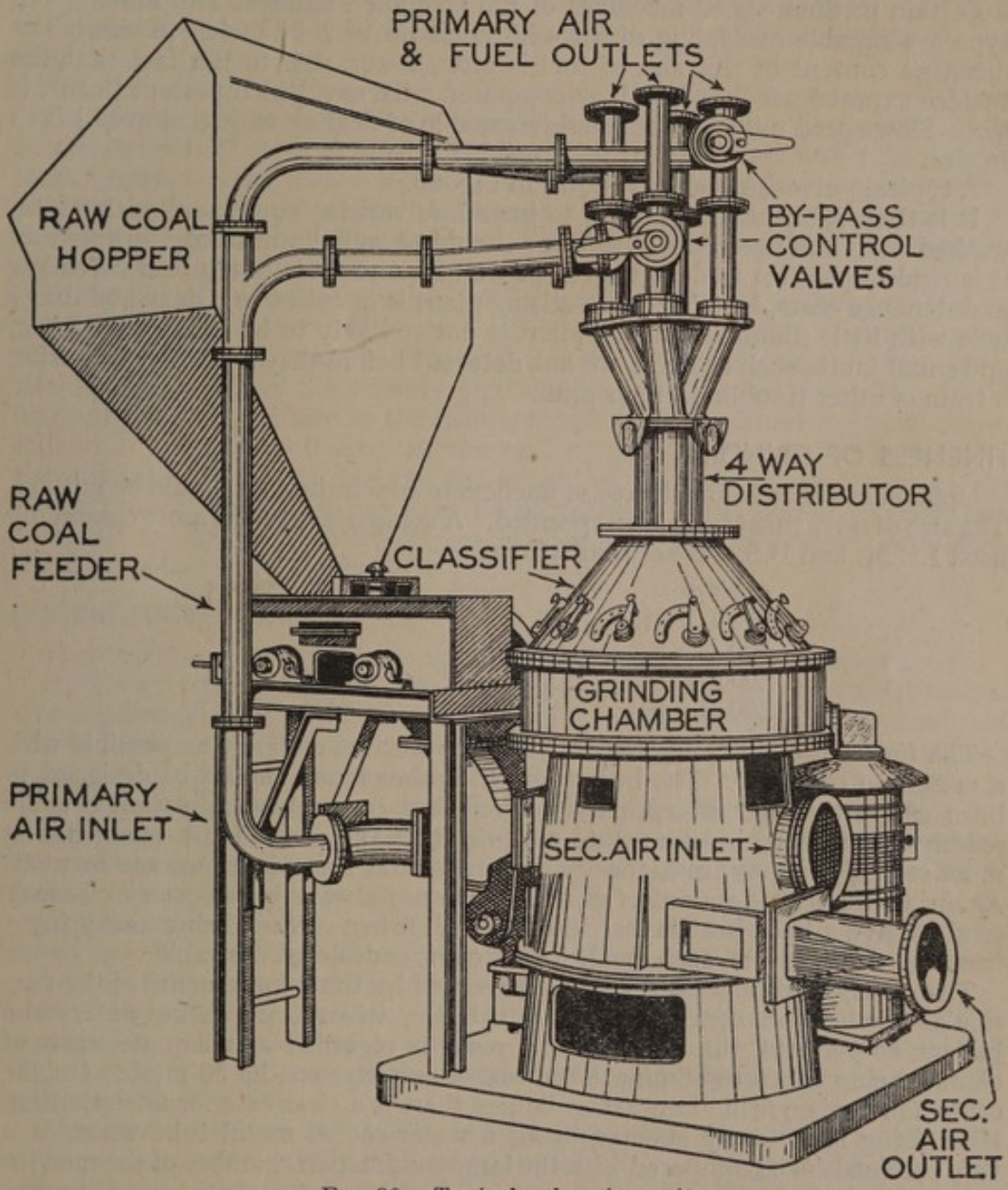

FIG. 90. Typical pulversing unit.

from an air heater or from hot waste gases. A self-contained unit of this type is capable in many instances of dealing with moistures up to 12–14 per cent.

There is a much greater difference between their full-load and no-load power requirements than in the slow-speed machines, and therefore they have a somewhat more flexible power curve for grinding over a wider range. The power consumption taken on full load is not very dissimilar to the ball mill, however, and under equal conditions would again be in the neighbourhood of 24 kWh per ton of coal pulverised.

The foregoing remarks are equally true of the high-speed mill, but it should be borne in mind that the very high speed mill tends towards a higher maintenance cost. All types of machines have an economic speed as regards maintenance. These high-speed mills sometimes run up to speeds of nearly 3,000 revolutions per minute.

Certain medium-speed machines of the combined hammer and moving peg type are capable of dealing direct with coals up to 2–2½ inches in size. The moisture content of this size of fuel is often lower, due to the fact that the surface exposed per lb. of coal, as compared with say ¼ inch peas or beans, is less. Slow-speed mills are less readily capable of dealing with coal over ½-inch in size.

A typical pulverising unit is shown in Fig. 90.

It is thoroughly sound practice to install pulverising equipment with white-washed walls and floors, and the house should be well illuminated. In this way it is readily possible to detect dirt. A well-kept plant is likely to involve low maintenance costs, but if the pulverising plant is installed in a dark and dingy hole with little illumination, the plant is not so likely to be well looked after, and small faults such as leaks are not detected before they multiply, and bring a train of other troubles in their path.

FINENESS OF GRINDING

Opinions vary as to the degree of fineness to which the coal should be ground. The size of sieve must be clearly specified. A comparison of the more commonly used I.M.M. and B.S. meshes is as follows :—

Sieve	Size of aperture
100 I.M.M.	·005 inch
100 B.S.S.	·006 ,,
200 I.M.M.	·0025 ,,
200 B.S.S.	·003 ,,

The function of the pulveriser is to grind the fuel as finely as possible with a minimum of power. The burner, on the other hand, should be designed to burn efficiently the coarsest possible coal, but complete combustion is more readily obtained with a fine state of division of the coal. Ultra-fine grinding must result, however, in higher power costs, and a marked increase in wear. Again, the highest standard of grinding may not always be necessary. It may tend to give a flash heat rather similar to oil firing. When firing heavy ingot furnaces, for example, such conditions are not considered desirable.

The question of fineness should be governed by the requirements of the particular application concerned. For instance, when firing large water-tube boilers, 65 per cent. through 200 B.S. mesh is regarded as adequate, care, of course, being taken to eliminate the coarse particles on the 30 mesh. On the other hand, when firing Lancashire boilers there is a clear case for fine grinding. Here firing is virtually effected down a water-cooled metal tube which is a very confined space compared with the large combustion chamber of the modern water-tube boiler.

Similarly, on very small drop stamp furnaces, of which many are being fired with pulverised coal, there is again a call for finer grinding than that required for large ingot heating or continuous reheating furnaces. Generally speaking, more emphasis should be laid on *uniformity of grinding* than fineness of grinding. If one could be sure of obtaining 100 per cent. through the 100 B.S. mesh, there would be fewer difficulties, since this would automatically ensure freedom from large particles, and there would still be a sufficient number of fines on the 200 mesh.

It will be evident from the above discussion that hard and fast rules of

practice have not yet been laid down, but for general furnace work the specification previously given under " Machinery used for pulverising " will be found suitable.

EXTRACTION FANS

In most types of pulveriser, extraction or suction fans are used for removing the pulverised coal and delivering it either direct to the furnaces or to the storage bins.

The pulverised coal fan, termed the primary air fan, when used on direct firing systems is generally of the simple paddle-wheel type, the scroll of the fan being made of sections of either chilled iron or reasonably heavy mild steel, these sections being easily removed for replacement when wear has taken place from the abrasive action of the coal. For the same reason the vanes of the rotor are generally made of either mild steel or chilled iron, and also easily removed for renewal purposes.

In the bin or storage system the coal is usually fed by means of screw feeders placed underneath the storage bin. This device consists of a simple screw rotated inside a metal casing. The coal thus fed is not delivered to the suction side of the fan, but to the delivery pipe leading from the fan to the burner. If necessary the air pressure in the delivery pipe can be reduced to below atmospheric at the point of entry of the coal by providing a correctly designed Venturi tube in the fan ducting and admitting the coal at the throat of the Venturi tube. Thus this fan may be of ordinary construction without any necessity for special precautions against abrasion required for primary fans which have to handle the coal.

DISTRIBUTION SYSTEMS

There are three main systems applying to pulverised coal, as follows :—

(1) *Unit System.* One pulveriser is applied direct to one or two furnaces. The application does away with all bins, and is suitable for medium sized and isolated furnaces. The size of pulveriser with this method must be selected carefully, otherwise the air for sweeping the mill, i.e. for the mechanical process of grinding, will not always coincide with the quantity of air necessary for combustion. This is often the simplest application, but it has its limitations. It would be impossible for example to cart coal to each of a large number of furnaces cramped together in a confined space.

Typical examples of an installation of this type are a single pulveriser firing direct to either one Lancashire boiler or one water-tube boiler ; and one small pulveriser firing a continuous reheating furnace or a forge furnace.

(2) *Bin and Feeder System.* This is usually the most complicated type of installation. It consists of a suitable pulveriser, of the slow-, medium- or high-speed type. A slow-speed pulveriser will generally have a separate coal drying plant (Fig. 89).

From the pulveriser the coal is delivered by means of a fan to a bin above which is a cyclone. This cyclone separates the air carrying the coal from pulveriser to cyclone, and deposits the coal in the bin. At the base of this bin are a number of feeders generally of the screw or rotary drum type, each distributing the pulverised coal to a boiler or furnace.

From the feeder the coal drops into an air stream supplied by a primary air fan, which delivers it through ducts to the burner.

This system is very suitable for distributing to a number of points in various directions, and has the advantage of centralising all the coal handling at one point. The number of feeders that can be put on one bin is limited. The more feeders on the one bin, the larger it must be, and the more sensitive the installation becomes to the influence of certain disturbing factors. Amongst these is

the tendency for moist coal to hang up in the bin, for the reason that its mass is increased whilst the rate of flow of coal through the bin is decreased. While large storage bins have their advantages, there is much to be said for keeping the bin size as small as is reasonably possible.

A typical example of such an installation is a central plant firing a battery of boilers or a battery of furnaces ; the furnaces may range in size from large continuous billet or ingot heating furnaces, or forge furnaces, to small stamping furnaces.

(3) *Ring Main System.* The coal preparation plant, inclusive of the pulveriser, is similar to that used in the bin and feeder system, in that the coal is pulverised and delivered by fan to the cyclone, which in its turn precipitates the coal into the pulverised fuel bin below it.

At the base of this bin are a number of feeders, each feeder supplying coal to a ring main which travels round the shop in which are the furnaces or boilers to be fired, and returns any surplus coal back to the cyclone for re-distribution. The mains may vary in length from 200 feet to 1,000 feet.

This system is particularly suitable for firing batteries of medium-sized furnaces with fuel consumptions each of, say, 30 to 350 lb. per hour. Its capital cost is considerably less than that of the bin and feeder system, and some 15 furnaces can be fed from one feeder point. By having four feeders at the base of a bin, each serving one ring main, it is possible to deal with a total

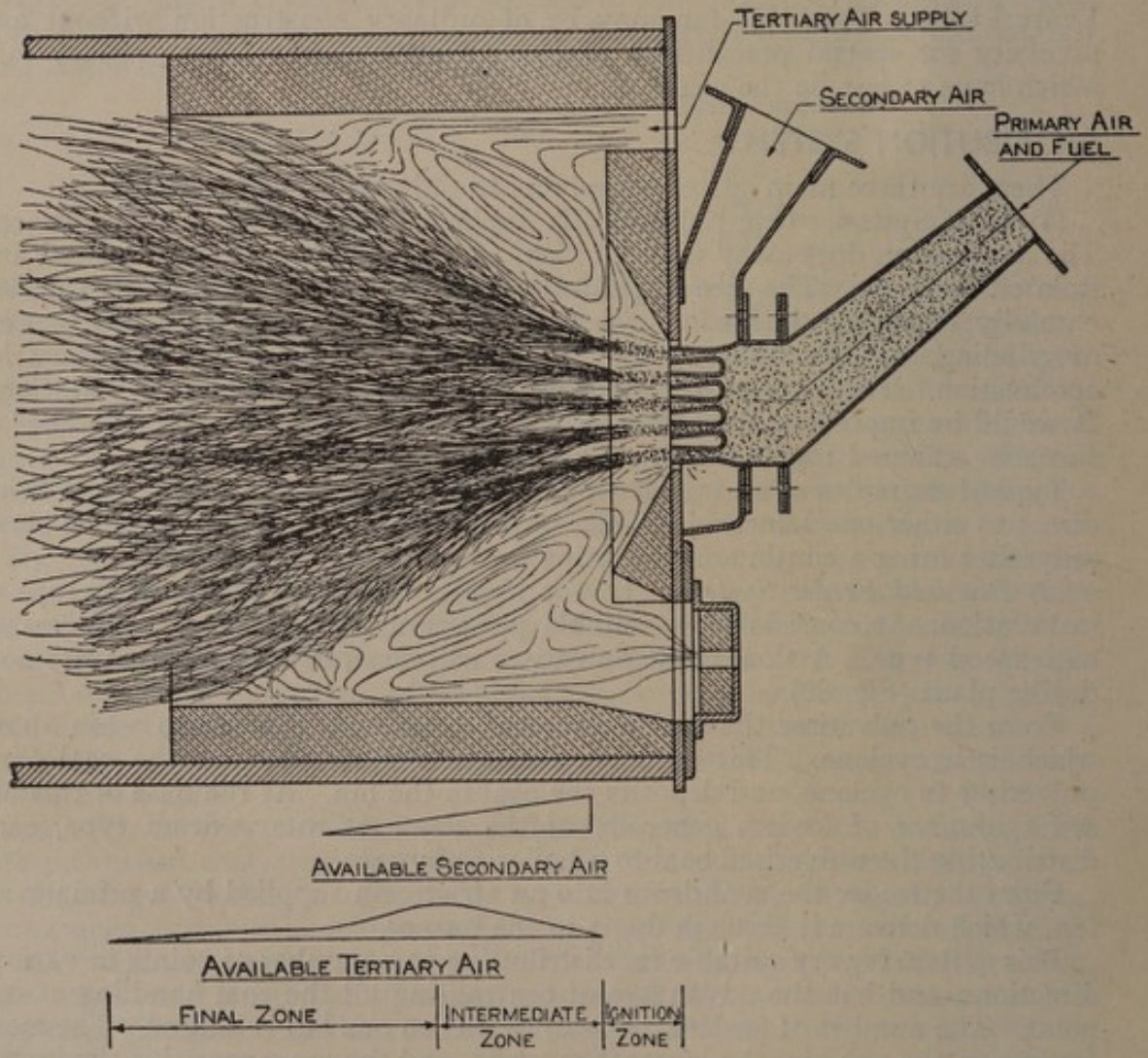

FIG. 91. The Fuel Research Board's grid burner.

of 60 furnaces. Obviously, to put down a bin with 60 feeders would be out of the question, firstly on capital cost, and secondly because of the size of bin which would be entailed. The only other method to adopt under these conditions is to have a bin and feeder system which in its turn is required to feed coal to secondary bins, each of which carries its own battery of feeders. The method is, however, less satisfactory than that involving the use of the ring main.

BURNERS

Burners for large water-tube boilers with combustion chamber capacity perhaps of the order of a 20-foot cube, should obviously have wide dispersive qualities. They are often of the semi-rotary type where the coal stream is given a rotation in one direction and the secondary air in the other, the idea being to obtain an intimate mixture which is widely dispersed throughout the chamber.

However, exactly the reverse may be required, for example, in a long narrow bar furnace for heating drop stampings. An intimate mixture of the air and coal is again required, but wide angles of dispersion which would merely fling the coal particles on to the brickwork are detrimental.

These two examples are given to illustrate how varied may be the requirements of different burners for different purposes.

The velocity of the primary air stream and coal leaving the burner generally varies between 30 and 60 feet per second according to the particular application concerned. The secondary air velocities are of the same order, but vary according to the loading of the burner. Generally, the primary air stream is constant and as the quantity of coal is increased or decreased the secondary air is altered as necessary. The velocity of the secondary air thus varies in order to obtain correct combustion as more or less coal is introduced into the burner.

The Fuel Research Board's Grid Burner is shown in Fig. 91.

GENERAL

At one time it was widely believed that pulverised coal firing was simply a means of burning dirty coal which was unusable by other methods. Fortunately this idea has given way to a realisation that pulverised coal is a highly scientific method of utilising to the highest extent the heating value of a given coal.

With this object in view, benefit results from the use of the cleanest and driest coal which is economically practicable. Where coal having a high ash content must be used much can be done to obtain the best possible results by pulverisation to a maximum degree of fineness.

The lower the moisture content the better. With preheated air most installations can handle coals containing as much as 10 per cent. of moisture, but such a moisture content will reduce the steaming load obtainable, and is best avoided where possible.

Practically all types of boiler, from the power station water-tube type down to the small vertical heating boiler, have been fired successfully with pulverised coal.

In water-tube boilers it is found practicable to obtain an overall efficiency up to 80 to 85 per cent.; and the Lancashire boiler can be made to give from 70 to 80 per cent. when fired with a good quality of pulverised coal, this figure including the added efficiency of superheaters, economisers and air heaters.

There are many types of furnace operation in this country to which pulverised fuel firing has now been applied. The flexible nature of the method of firing, and, where batch heating is involved, the fact that the source of heat is stopped immediately the operation is completed are conducive to economy.

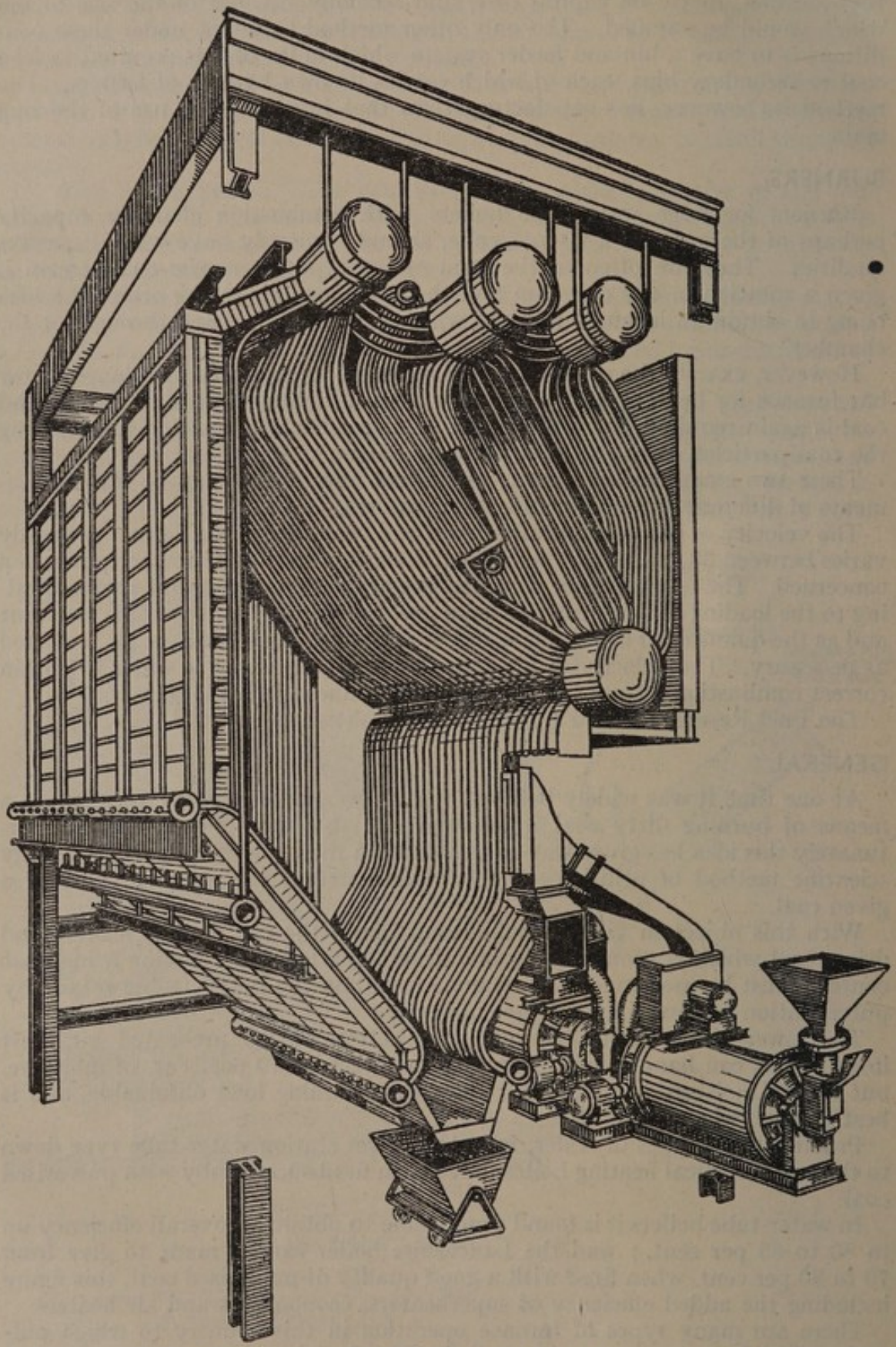

FIG. 92. Powdered fuel firing applied to a water-tube boiler.

For the control of fuel-air ratio the fineness of the coal should be checked periodically by taking small samples of say 100 grams and passing these through a 200 B.S. mesh sieve. By shaking and tapping this screen all the coal of the correct size will pass through the screen, and the balance which will be held on the screen should then be carefully weighed, thus determining the percentage of the coal which is reduced to the correct size. The volume or weight of air which is being delivered by the fans is checked by means of comparatively simple instruments (Chapter X). The volume of air delivered by the various fans may be controlled either by dampers placed in the suction or delivery ducts or by variable speed motors.

As with other methods of firing solid fuel a CO_2 recorder will serve to verify the combustion. A steam flow meter is also of great help as the plant operator is able, with a little experience, to learn the control settings at which his fans should operate in conjunction with varying rates of steam flow.

The applications for pulverised coal firing fall into three main categories—boilers, cement and metallurgical.

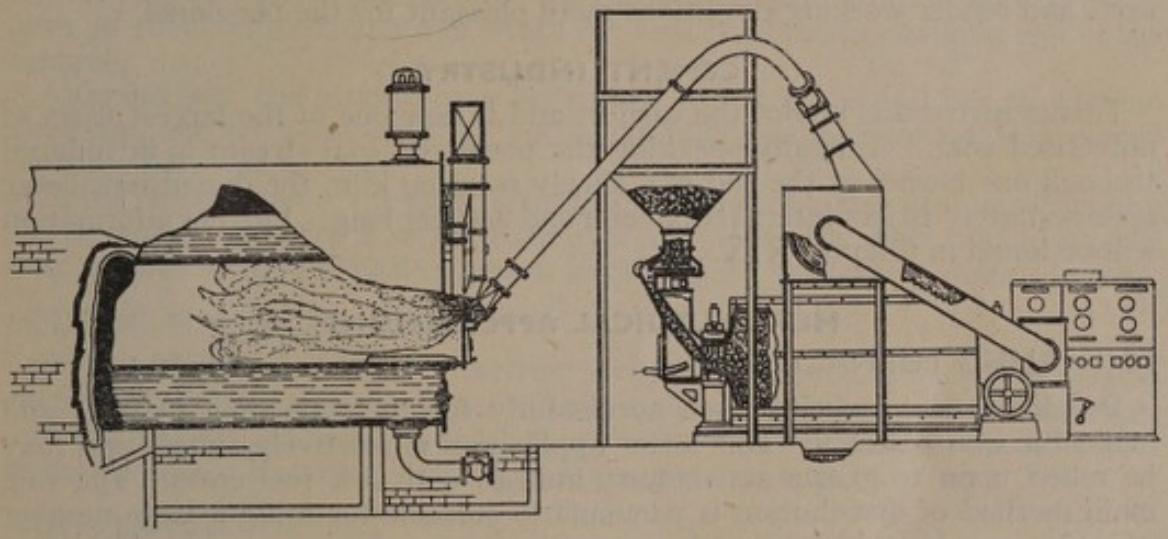

FIG. 93. Powdered fuel firing applied to a Lancashire boiler.

BOILERS

The application of pulverised coal to boiler firing has long been established practice. The main applications are to water-tube boilers (Fig. 92), but of late years a number of Lancashire boilers have been fired with success (Fig. 93). The application to economic and small vertical boilers down to a rating of 1,000 lb. per hour steam evaporation is by no means unknown.

The application of powdered coal solely to a single small vertical boiler of 1,000 lb. per hour steaming capacity, might not be generally economical, unless the boiler were only one point to be fired in a large installation, then the position would be entirely different. Nevertheless local circumstances may alter the position as powdered coal is used successfully on small heating boilers.

COMBUSTION CHAMBERS

Combustion chambers may be of several types depending on the duty required.

(1) They may be constructed of solid brickwork lined with refractory bricks.
(2) They may be of similar construction, arranged with an air space between the inner refractory bricks and the outer common brick wall.
(3) They may be of the suspended wall type, with the hollow space again between the refractory and the outer wall.

(4) The furnaces may be completely or partially equipped with what are known as water walls, these consisting of boiler tubes connected to the main boiler on their upper ends, and to headers outside the combustion chamber on their lower ends.

It is always advantageous to use preheated air both for primary and secondary air supply. This air may be heated by air-heaters placed in the flues beyond the combustion chamber or by means of the hollow walls just mentioned.

Users of pulverised fuel should investigate their plants with a view to discovering any sources of waste heat which might be utilised as a means of heating air. Economy and ease of operation could be improved in this manner.

In new installations where new or reconstructed combustion chambers are called for, it is recommended that wherever practicable the walls be of hollow construction. Air can then be drawn through these walls by the primary air fan or by means of a small subsidiary fan, and whilst this will give valuable preheated air for combustion it will also serve to prolong the life of the brickwork and render working conditions more pleasant for the personnel.

CEMENT INDUSTRY

This industry was one of the earliest and is now one of the largest users of pulverised coal. Generally speaking the powdered coal stream is introduced through one burner at the end of a slowly rotating kiln, the dimensions being approximately 10 feet internal diameter and 240 feet long. Further information will be found in Chapter XIX.

METALLURGICAL APPLICATIONS

ANNEALING FURNACES

Powdered coal is being used successfully for the annealing of steel and malleable iron castings. This latter application is relatively simple and may be relied upon to give a satisfactory anneal, with low fuel costs. The ring main method of distribution is particularly suitable for firing a large number of such ovens in any given works.

SMALL FORGE FURNACES AND DROP STAMP FURNACES

Many hundreds of these furnaces are fired to-day by powdered coal. The size of furnace varies from the small door type of only 18 by 18 inches, to the larger type suitable for cut lengths of billets up to 18 inches square section, having hearths of 12 by 6 feet.

HEAVY FORGE FURNACES

Blooms are heated by pulverised coal for serving both hammers and presses. The furnaces range from multiple small units dealing with 5-ton ingots, up to 50 and 100-ton ingots.

There are two main methods of applying powdered coal to forge furnaces; one is to cross fire into a side combustion chamber, allowing the gases to follow in their flow the contour of the roof, or alternatively the more direct method of applying the burners in the back wall of the combustion chamber and firing over the top of the ingot, care being taken that the direct flame from the burners does not impinge on the stock, particularly with alloy steels.

Care must also be taken to avoid the deposition of particles of fine ash on the metal. If this happens, the ash is forced into the metal under the hammer, and gives a faulty forging. This difficulty can be overcome to a large extent by installing, when space permits, a pre-combustion chamber before the furnace proper. The effect may also be minimised by ensuring consistently fine pul-

verisation of the fuel so that the ash particles are small enough to be carried over by the velocity of the gases.

CONTINUOUS FURNACES

Powdered coal has been applied to many sizes and types of continuous furnaces in the steel industry, with outputs varying from ½ ton to 15 tons per hour. For ordinary mild steels the application is simple, two or more burners generally being placed in the back or combustion wall of the furnace and directed in a parallel line with the lie of the stock.

The actual design of furnace for specific purposes such as the production of solid drawn tubes, or other types of piercings must receive careful consideration, for in both of these applications the importance of the steel being really well soaked cannot be over-estimated. As an alternative to an increased rate of heat input at the discharging end, boosting burners can be placed half-way up the furnace.

In attaining uniformity of temperature much can be done by the design of the furnace as regards the slope of the furnace hearth and roof both of which have an effect on the manner in which the heat reaches the charging end of the furnace.

Alternatively, the furnace may be worked with a horizontal hearth with no slope and the temperature distribution attained by means of suitably placed multiple burners.

In the larger types of continuous furnaces for ingot heating dealing with alloy steels, modern practice is to fire both over and under the ingot and to have a soaking zone at the end of the furnace for producing uniformity of temperature.

MELTING FURNACES

Powdered coal is used for melting iron for the production of malleable castings, and also in roll melting. The furnaces are of the reverberatory type, and the burners are generally placed in the back or combustion chamber wall of the furnace. Rotary or oscillating types are also applied.

NON-FERROUS METALS

In the non-ferrous metal industry the main application is copper melting and refining. One of the earliest applications of pulverised fuel firing on a substantial scale was in the roasting of copper ores.

The use of pulverised coal has the following general features :—

(1) Flexibility in the rate of combustion.
(2) Close control of furnace atmosphere can be secured as a result of its having certain features in common with gaseous and liquid fuels.
(3) A flame of high calorific intensity.
(4) Special problems arise associated with the elimination of ash and grit carried in the flame and products of combustion.

CHOICE OF FUEL

Generally speaking the moisture and ash content should be kept as low as possible, but here again the actual ash and moisture contents safely usable are governed entirely by the application. Thus, at collieries it is often quite sound practice to use de-dusted fines containing up to 30 per cent. of ash for the pulverised fuel firing of water-tube boilers. In this way the colliery can often find all its power requirements with a fuel which would otherwise be wasted. On the other hand to burn such a coal in say an ingot heating furnace would be impracticable due to clogging of flues and furnace and combustion chambers.

Most metallurgical applications call for a fuel of under 10 per cent. ash, whilst obviously the moisture should be kept as low as possible, though with

certain types of medium-speed mills moisture contents in excess of 10 per cent. can be dealt with.

On central plants the moisture should be kept very low, and should never exceed 6 to 8 per cent., or trouble can be anticipated in the pulverised coal bin itself.

Fusion Point of Ash. This is an important point ; a governing factor is whether or not it is required to work with a dry bottom or dry furnace as for example in an annealing furnace, or alternatively whether the temperatures of the process are so high that a liquid bottom or slag tap operation is inevitable. With a slag tap a coal with a low fusion point of ash, which will facilitate the tapping of the furnace, is preferable.

Conversely, a high fusion point ash is required in the fuel with a dry furnace.

Volatile Content. The V.M. may vary between 20 and 40 per cent., but should not be lower than 20 per cent. Certain boiler applications can use volatile contents down to 5–10 per cent., but special arrangements have then to be made as regards secondary air distribution and, generally speaking, anthracite is not desirable for furnace work.

FURNACE DESIGN

This is a specialised matter depending upon many considerations. The following are fundamental points of general interest.

Burner Position. The burner must be so positioned that the coal does not directly impinge on any refractory surface, otherwise there will be a very high rate of brickwork wear. More often than not, if there is a high rate of wear in a powdered coal furnace it is not due so much to excessive temperature as to the mechanical erosion of semi-burnt particles impinging on the walls. Here they continue to burn and combine chemically with the brickwork, causing rapid fluxing and erosion. A reducing action of the carbonaceous matter in the presence of ferruginous impurities in the ash can result in the formation of fluid magmas rich in ferrous silicate.

Flue Design. Flue design should always receive special attention. With pulverised coal the presence of ash is inevitable. Adequate provision for dealing with the ash is therefore imperative.

If the application calls for slagging in the furnace, an appreciable proportion will be dealt with in a liquid form and tapped out. If on the other hand the process is operated at temperatures below the range at which fluid slag is formed provision must be made for the removal of the ash from downtakes and flues.

Therefore the flue ports should, as far as possible, be large and few in number, and access should be provided for cleaning or rodding. It is often sound practice where room permits, to provide a small settling chamber between the furnace and the main flue. This is often possible in ingot heating furnaces and in some stamp furnaces.

The advantage is twofold ; firstly this chamber acts as a settling chamber for the dust, and secondly, as such it acts as an extension chamber and lowers the temperature, and here again, with careful design, it is possible to ensure that the temperature drops from say, 1,100° C. or 1,200° C. to 800° C. The ash would then have dropped below nodulising temperature into the dry state where it can be effectively dealt with. There should be easy access to these chambers.

Another expedient to assist in ash removal is, assuming adequate draught to be available, to admit a small amount of cold air by means of an airbreak damper which will chill the particles so that they can be dealt with easily in a dry state.

Flues should be kept above ground for ease of access where possible, but

where due to site conditions they have to be below ground they should be of adequate proportions. Another sound point is to have access covers at such obvious blockage points as right angle bends.

The flues of continuous furnaces are generally a source of little trouble, as the exit gases should be at a low temperature, and therefore the question of nodulisation in the flue does not arise ; the only problem is that of providing reasonable access.

Air Recuperation. The use of regenerators is not entirely impracticable, provided adequate means exist for the removal of accretions of dust and clinker.

A metal recuperator is recommended and if installed, the following points should be given close attention :—

(1) *Choice of Material.* An adequate grade of heat resisting steel or iron should be selected ; amongst available materials are those capable of operating at temperatures up to 1,000° C. Alternatively, lower grade materials to withstand, say 600°–700° C., can be used, but the temperature of the incoming gases must be controlled accordingly.

(2) *Entry of Flue Gases to Recuperator.* It is sound to arrange for the temperature of the gases entering the recuperator chamber to be at least 80°–100° C. below the supposedly safe temperature of the metal selected. The gases can be adjusted to the required temperature level by by-pass flues or air infiltration. If air infiltration is allowed it is necessary to safeguard against the possibility of stratification of hot and cold streams with resultant distortion of the recuperator.

At a point just prior to the entry of the recuperator chamber a recording pyrometer with an electric connection can be provided, so that if the temperature rises above the selected point a warning horn can be sounded, or a red light shown, to attract the furnace attendant's attention.

(3) *Ash Accumulation.* The spacing of the metal elements should be fairly liberal, and the ash can be readily removed by means of dust blowers, following the normal practice in water-tube boilers, since the ash in the recuperator, due to the limit on maximum temperature of entry of the hot gases and the cooling effect of the tubes, will be in a dry dust form.

Another precaution, although it is not absolutely necessary provided the incoming temperature of the gases never reaches a dangerous limit, is so to arrange a fan that if hot recuperated secondary air is not being used on the furnace, a proportion is being blown to atmosphere by means of a bleeding-off pipe. This ensures that air will always pass through the recuperator.

STORAGE OF COAL IN BINS

Under certain atmospheric conditions, particularly when there is considerable humidity and the coal is wet, there is a risk of the coal in storage bins heating up, and if this is allowed to continue indefinitely, spontaneous combustion will eventually begin.

As a rule there is no risk of explosion, but merely the heating up of the bin until the coal begins to glow. If this should occur, it can be dealt with by suitable CO_2 equipment.

The operation of the CO_2 supply can be made manually on the indication of pyrometers placed near the top as well as near the bottom of the bin, or alternatively the use of automatic fire prevention equipment controls the flooding of carbon dioxide into the bin.

THE EFFICIENT USE OF PULVERISED FUEL IN BOILERS

The use of coal in the pulverised form as a fuel for boilers introduces many problems which are peculiar to this system of firing and, if the highest possible fuel economy is to be realised, these problems must be properly understood.

In this section attention is directed to the more important factors influencing the attainment of efficient operation to stimulate rational investigation by means of which the engineer in charge can find out for himself of what his plant is really capable and in what ways it is falling short of what is attainable.

In approaching the subject in this way from the point of view of an existing installation the first questions which must be answered are the following :—

(1) What is the highest efficiency which it should be possible to obtain from the plant under service conditions ? This will be referred to throughout as the bogey efficiency, by analogy with golf.

(2) In what respects is the operation of the plant falling short of this bogey efficiency and how can the inefficiencies be corrected ? Some of the inefficiencies will almost certainly be due to causes in no way associated with the use of pulverised coal and only those directly or indirectly due to this method of firing will be here examined in detail.

The best procedure to adopt in investigating the efficiency of working of a boiler plant is first to examine a comprehensive list of all factors on which the efficiency depends and to prepare a bogey energy balance sheet for the installation. In connection with each item should be considered any data available on design conditions, the present achieved operating conditions and the bogey efficiency conditions attainable consistent with limitations existing due to design or local conditions. Such a list is given later, but before considering in detail the problems which arise it will be well for the engineer in charge of the plant to satisfy himself that he is really clear as to the fundamentals of the steam-raising process and what effects will result from changes in the many variable conditions, for example the relation between CO_2 in the flue gases and the overall thermal efficiency of his installation.

Whatever may be the class of fuel, the type of boiler and its operating steam pressure and temperature, or the purpose for which the steam is required, the basic principle is to convert economically the greatest possible amount of the energy in the fuel to energy in the steam leaving the boiler installation.

It is clearly impossible to realise a 100 per cent. conversion as some of the energy in the coal will, for example, be required to pump water into the boiler, some to drive air and gas fans and some to elevate and crush the coal. Losses must also occur from such factors as radiation from the boiler casing, the evaporation of moisture in the coal and heat carried to atmosphere in the chimney gases.

The above limitations can be set out in the form of an energy balance sheet : a typical sheet for a pulverised coal-fired boiler installation would be as in Table 62. This table gives bogey figures, as well as performance figures which might be obtained from a reasonably well operated installation, using a slack coal containing 15 per cent. ash and 3·5 per cent. moisture.

Table 62 does not take into account the energy required for coal crushing, screening and pulverising, for feed-water heating and pumping, and for such equipment as air and gas fans, etc. As the total energy for these purposes will not be inconsiderable, being of the order of 2 to 4 per cent. of the energy in the steam produced, every effort must be directed to obtaining the highest possible efficiency of operation of the auxiliary equipment.

Taking now the various items in the balance sheet (Table 62) consideration must be first given to the factors involved in each of these items and their relation to the magnitude of the loss, to the assessment of the " bogey " loss and to determining what steps can be taken to reduce the " actual performance loss." Brief mention only is made of these points where they are not of particular significance in pulverised coal firing, but whenever they are influenced largely by this method of firing sufficient detail is given to direct attention to the desirability and best method of investigation.

TABLE 62

Item	Percentage of energy in coal	
	Bogey	Actual performance
	Per cent.	Per cent.
1. Loss due to moisture in coal	0·3	0·3
2. Loss due to moisture in air supplied to boiler ..	0·1	0·1
3. Loss due to moisture formed in burning hydrogen in coal	3·6	3·6
4. Loss due to incomplete combustion of carbon in coal	0·1	0·2
5. Loss in dry gases discharged to atmosphere ..	3·4	4·6
6. Loss due to unburnt carbon in ash	1·4	3·2
7. Loss due to radiation	0·4	0·8
8. Loss due to heat losses in starting up and shutting down boilers	0·1	0·2
9. Loss due to boiler water blow-down	0·3	0·5
10. Loss due to steam for soot blowing	0·3	0·5
11. Unaccounted for losses	0·5	0·8
12. Energy in steam generated	89·5	85·0
TOTAL	100·0	100·0

(1) *Loss due to Moisture in Coal.* All the moisture which is present with the coal as it is fed to the boilers will be converted to steam and this steam will be finally discharged to atmosphere at the temperature of the chimney gases with a consequent loss of heat energy. The loss of energy due to this cause is relatively small for normal moisture coals and the only ways in which the loss can be reduced are either by reduction of the moisture in the coal or a lowering of the temperature of the gases leaving the boiler system. The effect possible by the latter is extremely small and can be neglected.

Reduction of the moisture loss by reducing the moisture content of the coal can be brought about :—

(*a*) By obtaining a coal with a lower moisture content.
(*b*) By removing some of the moisture in a predrying plant in which heat energy which would otherwise be lost is used for the drying operation.

Unless the coal in use has a very high moisture content (*b*) above need not be considered. Added moisture, due to rain on unsheeted waggons or unprotected receiving hoppers, taken over a long period will have only a small effect on the fuel efficiency : it may, however, be an important factor in connection with the pulverising process : this aspect is dealt with later.

(2) and (3) *Losses due to Moisture in Air supplied to Boiler and to Moisture formed in Burning Hydrogen in Coal.* For all practical purposes both of these losses can be considered as fixed losses. The first, being of very small magnitude, is unimportant ; the second, which is quite considerable, is, however, quite unavoidable so far as concerns the latent heat of the resulting steam and most of its sensible heat.

(4) *Loss due to Incomplete Combustion of Carbon in Coal.* Unless operation is seriously at fault this loss will be very small with a well-designed installation. Abnormal conditions such as a sudden change in steam demand may lead to excessive coal feed to a burner or to incorrect adjustment of coal and air rates resulting in incomplete combustion of the coal, but, provided these conditions are infrequent, the effects on thermal efficiency will be very small.

Incomplete combustion of the carbon in the coal will be accompanied by a discharge of black smoke from the chimney, and boiler operators should be

encouraged to observe the condition of chimney gas at regular intervals if the position of the boiler operating floor permits of their doing this without interference with the necessary attention to the control of the boilers.

(5) *Loss in Dry Gases discharged to Atmosphere.* The magnitude of this loss, whilst, of course, being dependent primarily on the design of the installation in respect of such items of equipment as economisers and air preheaters, is, for any particular boiler, determined almost wholly by the quantity of air supplied to the furnace. This loss would be at a minimum value if it were possible to limit the air supply to the quantity theoretically necessary for complete combustion of the coal, but in practice about 30 per cent. excess air is necessary. The design of the boiler in respect of such items as type of burner, size, shape and degree of cooling of combustion chamber will determine the amount of excess air which must be supplied to ensure complete combustion of the coal, but as all these factors will be influenced by the fineness of the pulverised coal it is important that the optimum fineness should be determined.

Practical tests should be carried out to ascertain the degree of fineness of the pulverised coal necessary to ensure a minimum loss in the dry gases discharged to atmosphere. As, however, the loss due to unburnt carbon in ash (see (6) below) is also dependent very largely on the fineness of the pulverised coal the optimum operating condition must be determined from an examination of the following variables :—

(*a*) Fineness of pulverised coal.
(*b*) Energy consumed by pulverising equipment, air and gas fans, etc.
(*c*) Loss in dry gases discharged to atmosphere.
(*d*) Loss due to unburnt carbon in ash.

The nature of the necessary tests and the form of the relationship between the above variables is dealt with under (6).

(6) *Loss due to Unburnt Carbon in Ash.* In pulverised coal-fired installations this is the most important of the controllable losses and if the installation is not correctly operated may reach serious proportions.

When coal is pulverised the resultant powder is not merely a mixture of coal particles and ash particles, but of coal particles, ash particles and particles containing ash and coal intimately combined. When this powder is burnt in a boiler furnace the coal particles and a part only of the coal in the particles containing ash and coal are converted into gases ; the remaining coal in the particles containing ash and coal is either deposited with these particles in the combustion chamber or is carried through the boiler system with them, entrained in the gases.

The portion that is deposited in the combustion chamber will vary in quantity as between one type of boiler and another, but seldom will be more than 15 to 25 per cent. of the whole ; the coal in this fraction being subjected to furnace conditions for an appreciable length of time will, to a large extent, be burnt, but the remaining 75 to 85 per cent. of the particles will pass through the system unburnt due to the cooling effect of the boiler heat-absorbing surfaces.

The amount of coal in the ash particles which pass through the boiler system and which causes almost all the loss under consideration will depend on a number of factors, such as :—

(*a*) The physical nature of the coal.
(*b*) The ash content of the coal.
(*c*) The fineness of the coal after pulverising.
(*d*) The amount of excess air supplied to the combustion chamber.
(*e*) Type of burner and combustion chamber.
(*f*) The load on the boiler.

Except inasmuch as it may be possible to employ a coal containing less ash the only factors which demand consideration are (*c*) and (*d*), and an investigation of how to reduce the losses due to dry gases discharged to atmosphere and to unburnt carbon in ash should follow the lines set out below.

A series of tests, each preferably of not less than 24 hours' duration, should be made covering a fairly wide range of pulverised coal fineness and of excess air supplied to the furnace. Two tests at least should be made for each condition and the tests must be so arranged and conducted as to eliminate as far as possible any effects due to such factors as variation in type of coal, boiler load and cleanliness of the boiler system.

It is outside the present scope to discuss all the problems associated with such tests, but it cannot be emphasised too strongly that the greatest care must be taken to ensure that the results are not influenced by factors other than those under examination. As an example of the steps which should be taken to make certain of these conditions, the following is a procedure which is probably the best to ensure that the state of cleanliness of the boiler shall not affect the conclusions to be drawn from the tests.

Assuming that it has been decided to carry out first a series of tests to relate the unburnt carbon content of the ash with the fineness of the pulverised coal, the first set of tests should be started with a boiler which has just been cleaned and should be worked up through a range of fineness from the finest to the coarsest grading to be tested. On completion of this set of tests the boiler should be cleaned and a second set of tests made, working down from the coarsest to the finest gradings.

As an example of the results which might be expected from such tests results are here given obtained from an installation using a coal having the following analysis :—

Ash	15·0 per cent.
Moisture	3·5 ,,
Carbon	68·0 ,,
Hydrogen	4·5 ,,
Sulphur	3·0 ,,
Oxygen and nitrogen	6·0 ,,

TABLE 63. RELATION BETWEEN COAL FINENESS AND CARBON IN ASH

Pulverised coal fineness : Percentage > 100 I.M.M. mesh	1·6	3·1	5·7	9·8	13·1	19·3
Average carbon in ash	8·8	10·1	12·1	17·4	20·7	27·5

The results in Table 63 were for the excess air rate to the boiler found from previous tests to be the most economical. From these and other data obtained during the tests the results in Table 64 were obtained ; an analysis of these

TABLE 64

Item	Units					
Fineness of pulverising	Percentage > 100 I.M.M. mesh	2	5	8	11	14
Carbon in flue dust	Percentage	9·1	11·7	14·9	18·3	21·9
Boiler thermal efficiency	Percentage	86·75	86·30	85·72	84·92	83·98
Mill output	Tons/hour/mill	8·1	9·2	10·0	10·6	11·2
Mill power consumption	kWh/ton of coal	30·1	27·7	25·6	24·0	23·1

results showed that the economic fineness of pulverised coal for the installation was 4 per cent. > 100 I.M.M. mesh.

It will be noted that the power consumption of the mill is very considerable ; this was due to the use of a coal with a poor grindability and the employment of ball mills operating on the storage system. The power consumption of the mill air fans is included.

Losses due to (7) *Radiation*, (8) *Starting Up and Shutting Down Boilers, and* (9) *Boiler Water Blow-down.* The use of pulverised coal as a fuel has no special effect on the magnitude of these losses which, in any event, being small in properly maintained installations, are not potential sources of any considerable fuel savings. It is, nevertheless, important to ensure that the lagging on the system is both adequate and kept in good condition, that unnecessary boiler shut downs are avoided and that the losses due to boiler water blow-down are minimised by the adoption, if possible, of a continuous blow-down system with heat recovery from flash-off steam.

(10) *Loss due to Steam for Soot Blowing.* With pulverised coal firing the quantity of ash carried forward in the boiler system is always very considerable except where very low-ash coals are used, and in many installations may amount to as much as 0·75 to 1·0 ton for every 100,000 lb. of steam generated. If this ash is allowed to accumulate on the boiler, economiser and air preheater surfaces the heat absorption of the system will be impaired. Removal of these deposits by soot blowers is, therefore, necessary, but it must not be forgotten that soot blowers, whether they operate with steam or compressed air, absorb energy, and that it is important that their positioning, as well as the sequence and frequency of their operation, should be the best possible.

It is difficult to carry out tests on soot blowing, but if the soot blowers are always operated to a strict routine with a timed period of operation of each soot blower the effect of precise changes in the routine can be observed.

It is probably true to say that in the majority of installations the soot blowers are used excessively with a resultant lowering of the overall efficiency.

(11) *Unaccounted for Losses.* It may be considered strange that in Table 62 the bogey value of these losses is not given as zero. The reason for this is that under this heading are included items which are not losses in the strict sense of the word. This item in the energy balance covers really the errors which exist in the other items in the balance sheet due to inaccuracies in metering steam, recording CO_2 in gases, measuring gas temperatures, and so on.

Any great variations in the value of this item in the routine plant balance sheets must be taken as an indication that instruments are not functioning properly or are incorrectly read, or that there has been faulty operation of, for example, soot blowers. The energy balance loss in the case of soot blowers is usually based, not on metered steam quantities, but on quantities calculated for a defined routine operation of the blowers.

Before leaving the subject of the energy balance sheet for the installation it will be well to recapitulate the issues which are of importance in a pulverised fuel fired installation.

The energy losses which are particularly affected by the use of the pulverised coal system of firing as compared with the grate system are :—

(*a*) Loss in dry gases discharged to atmosphere.
(*b*) Loss due to unburnt carbon in ash.

The first of these losses should be lower with pulverised coal than with grate firing since with pulverised fuel it is possible to use a smaller quantity of excess air and by the use of higher air preheat, amongst other factors, to attain a lower exit gas temperature and, therefore, a smaller loss due to dry flue gases.

With pulverised fuel firing the loss due to unburnt carbon in ash will be almost invariably greater than with grate-firing and unless the matter has been thoroughly investigated this loss will, in the vast majority of plants, be one which can be appreciably reduced.

The point which must, therefore, be stressed is that the precise magnitude of these losses under existing operating conditions should first be determined, and, from data obtained from carefully conducted and scientifically planned tests, the optimum practical operating conditions should be ascertained.

In Table 65 are set out typical balance sheets for pulverised fuel and for chain-grate fired boilers, for the type of coal referred to earlier in this chapter. The higher loss figures under columns " A " and " C " are those which might be occurring in a well-designed plant indifferently operated ; the figures under columns " B " and " D " represent conditions closely approaching the bogey losses for the installations.

TABLE 65

Item	Percentage of energy in coal			
	Pulverised fuel		Chain-grate stoker	
	A	B	C	D
1. Loss due to moisture in coal	0·3	0·3	1·5	1·4
2. Loss due to moisture in air supplied to boiler ..	0·1	0·1	0·1	0·1
3. Loss due to moisture formed in burning hydrogen in coal	3·6	3·6	3·7	3·7
4. Loss due to incomplete combustion of carbon in coal	0·2	0·1	0·3	0·1
5. Loss in dry gases discharged to atmosphere ..	4·6	3·4	7·6	5·7
6. Loss due to unburnt carbon in ash	4·0	1·8	3·1	0·7
7. Loss due to radiation	0·8	0·5	0·8	0·5
8. Loss due to heat losses in starting up and shutting down boilers	0·2	0·1	0·2	0·1
9. Loss due to boiler water blow-down	0·5	0·3	0·5	0·3
10. Loss due to steam for soot blowing	0·5	0·3	0·5	0·3
11. Unaccounted for losses	0·8	0·5	0·8	0·5
Total losses	15·6	11·0	19·1	13·4
Gross overall thermal efficiency	84·4	89·0	80·9	86·6

Consideration has now been given to all the principal factors which are associated with the combustion process and it remains to deal with the general survey of the installation mentioned in the opening paragraphs of this section.

To present a really comprehensive survey of all the items of equipment in a boiler installation is outside the present scope. To indicate, however, the method of examination which is recommended a partial survey of a typical boiler plant is set out in Table 66. The points covered by this survey are by no means all which will have to be considered, but should be sufficient to indicate the method of approach to the problem.

If a survey on these lines is conscientiously carried out it is almost certain to draw attention to items about which the engineer in charge lacks complete information or in connection with which improved conditions are possible. There is very little doubt that in all installations some, if only a small, improvement in overall efficiency is possible and that in many installations very substantial savings can be effected.

Efficient operation of boiler installations is of vital importance and no time should be lost or effort spared in effecting all economically practicable improvements. The boiler installation is too often considered as of importance only

in so much as it shall continue to produce the quantity of steam required; much is lost by this attitude and if engineers would devote more time to a scientific investigation of the performance of their installations they would make a very substantial contribution to the national effort, and would, at the same time, find in their investigations much of real interest to them as engineers.

The best possible advice to the engineers in charge can be summed up in a few sentences :—

Know your installation in the greatest possible detail; its capabilities and its shortcomings.

Take a very close interest in the operators and maintenance personnel and make sure that they, too, understand thoroughly the portions of the installation for which they are responsible.

Study the performance of the installation and by properly conducted tests ascertain the optimum conditions of operation.

Make sure that the information on which the installation is controlled, for example, the fineness of the pulverised coal, the CO_2 in the flue gases, the unburnt carbon in the ash, is reliable and is properly used.

Provide the operating and maintenance staff with clear and precise instructions and make sure that they understand and carry them out.

TABLE 66. SURVEY OF PULVERISED COAL-FIRED BOILER INSTALLATION (Storage System)

Item	Particulars of equipment and other information	Matters to be considered
Raw coal	Coal consumption of plant, 2,600 tons/week. Coal from three sources in varying quantities. Average analysis of all coal for past twelve months. Per cent. Ash .. .. .. 15·0 Moisture .. .. 3·5 Carbon .. .. 68·0 Hydrogen .. .. 4·5 Sulphur .. .. 3·0 Oxygen and nitrogen 6·0 Moisture content at times as high as 5·5 per cent.	Can a lower ash coal be obtained in order to reduce unburnt carbon in ash losses, mill wear and power consumption, and rate of slag formation in furnace ? As choking and corrosion of air pre-heaters is being experienced could a lower sulphur content coal be obtained or could arrangements be made on the mills to purge out some of the pyrites in the coal ?
Raw coal handling	All coal is rail-borne direct to plant. 2 receiving hoppers at ground level (not roofed). 2 skip hoist elevators. 2 band conveyors.	Are waggons sheeted and if so, are sheets removed only immediately before discharging the waggons in rainy weather ? Are mill operators informed by off-loading operators of the type and condition of the coal which is being elevated and the bunkers to which it is being fed ? Are the loading valves correctly adjusted to ensure a full skip bucket and so keep operating time and, therefore, power consumption of elevators to a minimum ? Is there any spillage of coal from the skips, and if so, is the coal being collected and returned to the hoppers ? Is the no-load power consumption of the band conveyors satisfactory ? Are the conveyor idlers properly lubricated and are steps taken to ensure that the free motion of the return half of the band is not restricted by spillage coal ? Are conveyors stopped immediately the coal supply to them has ceased ?

Item	Particulars of equipment and other information	Matters to be considered
		When a batch of wet coal is received, is this coal distributed over a number of bunkers or otherwise mixed with drier coal to minimise effect on the mills ?
Pulverising equipment	4 ball mills with circulating air fans and cyclones. 5 tons/hour rated capacity with coal of type referred to under " Raw Coal " above and with grading of pulverised coal 8 per cent. > 100 I.M.M. mesh. Actual performance—4·6 tons/hour at 6 to 8 per cent. > 100 I.M.M. mesh. Coal fed to mills by variable speed rotary feeders. Mill systems supplied with cold air from boiler house and vented to boiler primary air fans. Atmospheric vents provided for starting up and abnormal conditions.	In view of performance of mills being below specification what information is available on the following points :— (*a*) What is the condition of the mill linings ? (*b*) What is the condition and weight of the ball charges ? Has the optimum ball charge been determined by plant tests ? What is the procedure for maintaining the ball charge ? What is the ball wear per ton of coal pulverised ? (*c*) Is the correct circulating air rate being maintained ? (*d*) Are the grading tests of the pulverised coal systematic and reliable ? What is the condition of the test sieves and are they checked against a standard sieve ? (*e*) Does the performance vary as between one mill and another and with one operator and another ? (*f*) What is the economic fineness of pulverised coal for the installation ? (*g*) Is coal being wasted by improper use of the mill system atmospheric vent or by continuous leakage through the vent valves ? (*h*) Is sufficient interest being taken by the engineers in charge in the performance of the pulverising equipment and has the significance of maintaining the correct final product fineness been properly explained to the operators ?
Firing equipment	4 spider feeders per boiler each feeding coal from the pulverised coal bunker to the primary air stream to the four burners. 4 turbulent type burners with refractory burner mouths set in refractory front wall of combustion chamber. Trouble experienced fairly frequently due to spider feeders tripping out due to choking with coal. Trouble experienced with slag formation on burner mouths. This slag usually on lower part of mouth.	Do the feeders maintain at all speeds a steady coal supply to the burners ? Do the feeders, possibly as a result of wrong adjustment or of wear, give rise to aggregation of the coal particles with consequent losses due to incomplete combustion ? Is this a cause of the slag formation on the burner mouths ? Has the correct ratio of primary to secondary air been determined by plant tests ? Are the burner air louvres correctly adjusted ? When refractory repairs are carried out is proper care taken to ensure symmetry and correct contouring of the burner mouths ? Are adequate instruments available to enable the operator to control the boiler efficiently ? Are the instruments properly maintained so that complete reliance can be placed on them ?

Item	Particulars of equipment and other information	Matters to be considered
		Is the CO_2 recorder sampling connection positioned so that a true sample of the flue gases is obtained or is a false reading registered due to stratification of the gases in the boiler flues ? Are all boiler instrument charts and record sheets examined by a competent engineer daily, and is any action, indicated by them as being necessary, taken immediately ?
Combustion chamber	Rating : 2·0-lb. coal/ft.^{3}/hr. at design load.	Are combustion chamber soot blowers operated at sufficiently frequent intervals to ensure that deposited ash particles will not sinter and form slag accumulations which cannot be removed by soot blowers ? Is the combustion chamber provided with adequate and suitably placed inspection openings and do operators keep a sufficiently close watch on combustion chamber conditions ? As the combustion chamber rating is high, is care taken to keep individual boiler loads as low as possible by equal distribution of the load ?
Boiler heating surface and flue gas system	Tri-drum boiler with single gas pass : radiant heat and convection superheaters ; steaming economiser and regenerative air preheater. Trouble experienced with slag formation on front row of boiler tubes. The rate of formation varies considerably as between one boiler and another. Considerable trouble experienced with No. 4 boiler air preheater with dust deposits and corrosion of plates.	Is the soot-blowing routine properly planned, efficiently carried out and recorded on the boiler record sheets ? Is care taken to ensure that in extremely cold weather the temperature of the air to the air preheater is not allowed to become so low that trouble results from flue dust adhesion to the heating surfaces at the cold end ? Is steam from superheater drains, steam traps, ash sluice channels, etc., allowed to enter the forced draught fan suction resulting in troubles in the air preheaters due to high moisture content of air ?
Boiler fittings and steam and water systems	Trouble has been experienced with safety valves not shutting off completely after blowing.	Are the safety valves absolutely steam-tight under normal working pressure conditions ? Are the water level gauges positioned at the right level and is the water level maintained at such a level as to ensure that carry-over of water with the steam is kept at a minimum quantity ? Is the boiler water blow-down procedure controlled in such a way that whilst maintaining correct boiler water conditions the heat losses are kept at a minimum value ? Is the steam flashed from the blow-down water used by delivering it into the low pressure mains for heating or other purposes ? Are steam trap lines effectively lagged so that the discharge of the trap is not due more to condensation in the trap lines than in the steam mains drained by the traps ?

CHAPTER XIII

STEAM BOILERS

Boiler design—Vertical boilers—Stationary locomotive boilers—Cornish boilers—Lancashire boilers—Externally-fired boilers—Economic boilers—Waste heat boilers—Water-tube boilers—Effect of prime mover on boiler design—Feed water temperature—Furnace design—Water circulation.

WITHIN the limits of modern knowledge, steam forms the most convenient and the most economical vehicle of heat for power and process purposes. Other substances, such as mercury and diphenyl, have been employed, but none of these is so universally available as the water from which steam is generated. Steam boilers therefore predominate in industry, and range in size from the elementary unit evaporating a few gallons per hour to the complicated power-station steam generator with an output of many thousand pounds per hour.

The purpose of a steam boiler, or more correctly a steam generator, is to produce steam under pressure from the raw materials fuel, water and air. The potential heat of the fuel is set free and this available heat then transmitted to, and stored by, water vapour in the form of sensible and latent heat.

The change and transference should be made with the minimum loss of heat, i.e. at maximum efficiency, it always being borne in mind that so far there have been financial, as well as practical, limitations to the degree of efficiency, but that in view of the urgent need to conserve fuel, the financial limitations may have to be somewhat relaxed.

Given equally efficient firing conditions, therefore, a good average thermal efficiency must be inherent in a boiler design by virtue of the disposition and form of its heating surface.

With some designs or types, the highest efficiencies can only be obtained by adding accessories to the boiler such as superheaters, air heaters, economisers, etc., but in other types such additions would lead to practical difficulties in the operation and excessive maintenance and repairs.

THE EMPIRICAL NATURE OF BOILER DESIGN

The development of steam boilers has so far proceeded along the lines of trial and error rather than by deliberate design based on theoretical consideration, since our knowledge has been insufficient to admit of any other course. To-day, however, accumulated knowledge enables the true boiler designer to design with somewhat more confidence and to forecast with reasonable accuracy the general conditions that may be expected to prevail within the boiler.

As the outcome of experience, boiler makers in general have now reduced the number of types to about six or seven, one or other of which have been found to be best suited to certain specific conditions. Thus the vertical cross-tube boiler is installed in factories requiring small amounts of steam, an improved type being employed when a greater output and higher economy is desired.

The stationary locomotive boiler is favoured by some trades because of its fairly rapid steam-raising qualities, and others prefer the larger Cornish or Lancashire types on account of their greater reserve and ability to satisfy fluctuating steam demands.

When space is limited and outputs of 1,000 to 30,000 lb. of steam per boiler per hour are required, the Economic boiler is widely used. In large capacity industrial and power-station installations where high working pressures are required, the water-tube boiler is necessarily adopted. Transport is one of the limiting factors in the manufacture of shell boilers.

EVAPORATIVE CAPACITY AND GRATE AREA

In all these widely varying types of steam boiler, the evaporative capacity is a function of the amount of fuel that can be burned in the firebox or furnace. It is also a function of the calorific value of the fuel used, since obviously a grate designed for a combustible such as wood fuel must be larger than one using a good quality coal, if the same heat input is desired.

In a vertical cross-tube boiler similar to that illustrated in Fig. 94, the diameter of the boiler is directly related to the area of grate required to burn a given amount of fuel. Boilers of this class are normally coal-fired under natural draught, and can burn from 14 to 20 lb. of coal per hour per square foot of grate area. Again, 1 lb. of coal burned in one of these boilers will

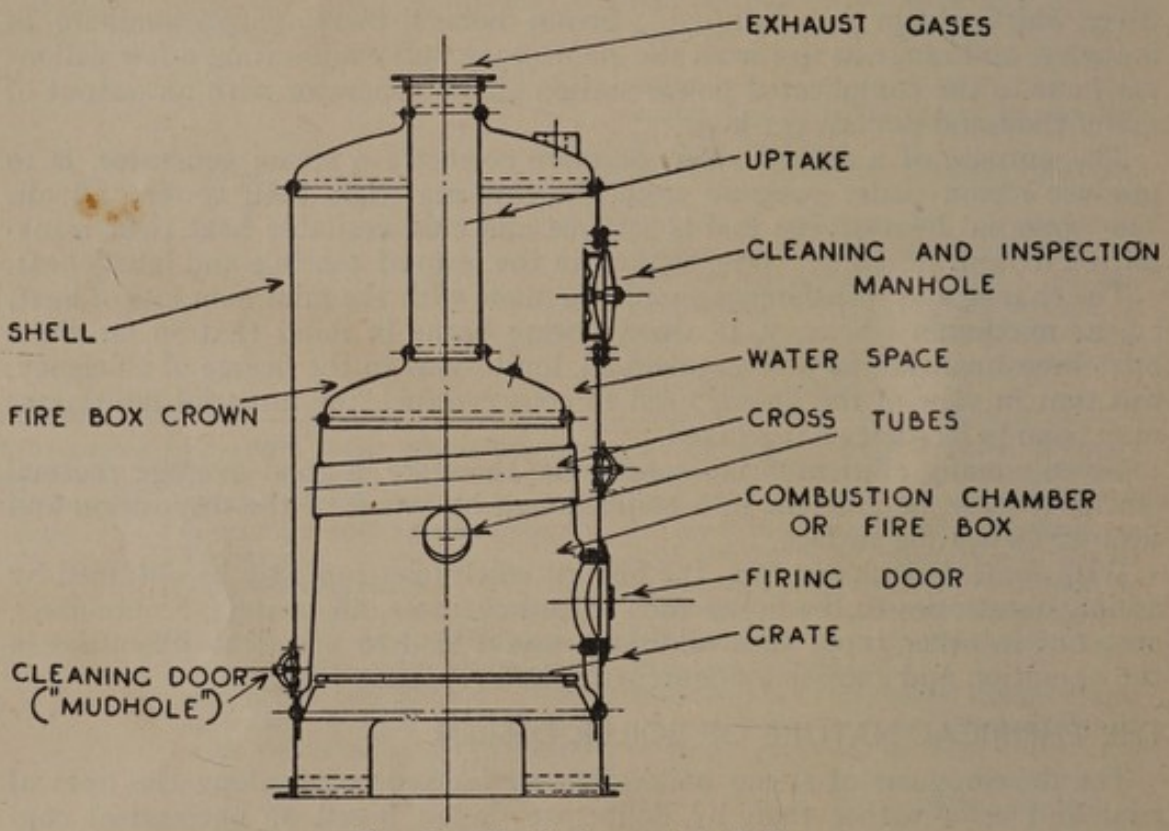

FIG. 94. Vertical cross-tube boiler.

evaporate from 5 to 8 lb. of water from and at 212° F., so that the computation of fire-grate area for a given evaporation is a comparatively simple matter.

VERTICAL CROSS-TUBE BOILER PROPORTIONS

As an example—and the method applies equally to all internally fired boilers, with suitable modifications—there is here considered a vertical cross-tube boiler required to produce about 1,700 lb. of steam per hour (from and at 212° F.). It is assumed that 18 lb. of coal are burned per square foot of grate area, and that each lb. of coal will evaporate 6 lb. of water.

The evaporation per square foot of grate area will be $6 \times 18 = 108$ lb., and the grate area required will be $1{,}700/108 = 15{\cdot}75$, say, 15·8 square feet. This gives a grate diameter of 54 inches, and, allowing, say, $\frac{1}{2}$-inch for the firebox plate thickness, the external diameter of the firebox is 55 inches. A space of at least $2\frac{1}{2}$ inches must be allowed between the firebox and shell for cleaning and circulation, so that the shell diameter must be 60 inches.

The number of cross-tubes fitted varies from one to six, being arbitrarily determined according to the boiler diameter. In the boiler under considera-

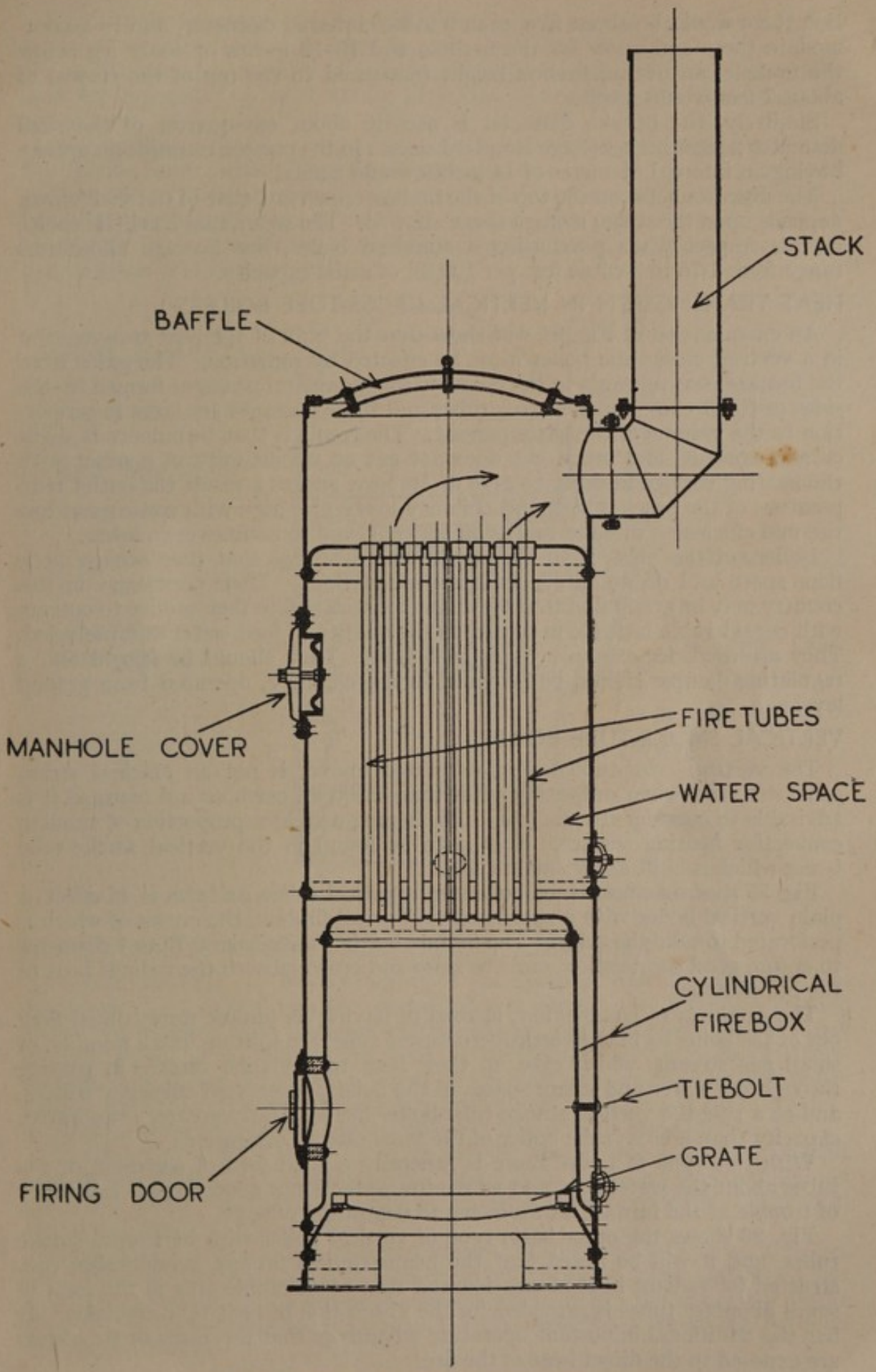

FIG. 95. Vertical fire-tube boiler with vertical tubes.

tion there would be about five, each 9 inches internal diameter, and to accommodate these and allow for the firehole and 10–12 inches of water leg below the firehole, an overall firebox height (measured to the top of the crown) of about 7 feet would result.

Similarly, the uptake diameter is usually about one-quarter of the shell diameter ; most makers have standard sizes. In the present example an uptake having an internal diameter of 14 inches would suffice.

The distance between the top of the firebox crown and that of the shell crown depends upon the steam storage space allowed. The steam user has little choice in this respect when purchasing a standard boiler, but average allowances range from 1·75 to 3 cubic feet per 100 lb. of rated capacity.

HEAT TRANSMISSION IN VERTICAL CROSS-TUBE BOILERS

An examination of Fig. 94 will show that the bulk of the heat transmission in a vertical cross-tube boiler must be effected by radiation. The gases from the furnace pass upwards to the chimney via segmental passages formed by the sides of the firebox and the cross-tubes and these passages are large in proportion to the volume of the gases passing. The result is that turbulence is slight or non-existent, and much gas does not get an opportunity of contact with the heating surface in order to give up its heat and as a result the outlet temperature of the gases is high and chimney losses also high with consequent low thermal efficiency, often as low as 50 per cent. and sometimes even lower.

Boilers of this class have the important advantage that they occupy little floor space and do not need elaborate foundations. Their popularity in this country may be attributed to these characteristics and to their ability to operate with considerable latitude in regard to the quality of feed water and fuel used. They are used, for example, as crane boilers. They should be lagged and a regulating damper should be provided in the chimney, operated from ground level.

VERTICAL SMOKE-TUBE BOILERS

The vertical cross-tube boiler, as stated above, is not an efficient steam generator, and when outputs greater than 1,500 lb. per hour are required it is advisable to consider the use of a boiler having a greater proportion of tubular convective heating surface. This is to be found in the vertical smoke-tube boiler which is built as two main types.

Fig. 95 shows a smoke-tube boiler with vertical tubes, and this is, in effect, a plain vertical boiler with a shortened cylindrical firebox, the crown of which is perforated to take the tubes. The firebox height varies from 0·75 to 1 diameter to ensure good combustion, and the sides and crown absorb the radiant heat of the burning fuel.

The products of combustion, instead of having an almost unrestricted flow out of the boiler as in the vertical cross-tube type, are split up into a number of small gas streams which give up their heat to the tube surface in passing through the water and steam spaces of the boiler. Increased efficiency results, and as a rule the vertical smoke-tube boiler has a slightly greater evaporative capacity than a cross-tube boiler of the same overall dimensions.

With this type of boiler there is generally a good deal of corrosion of the tubes about the water line, and where the water is not good there is a liability of trouble at the furnace crown owing to deposition of scale.

Fig. 96 shows the other basic type of vertical boiler with horizontal smoke tubes, and it will be noted that the hemispherical firebox is admirably constructed for radiant heat absorption and that considerable area in the form of small diameter tubes is provided for the absorption of heat by convection. It has the additional important operating advantage that no seams or tube ends are exposed to the direct heat of the fire.

Further, the refractory-lined chamber arranged in way of the back tube plate serves to complete the combustion process.

The efficiency is generally of a reasonably high order and the boiler can be used for capacities up to 6,000 lb. of steam per hour under easy steaming conditions.

Another design of this type has an entirely submerged combustion chamber, thus giving more radiant heat absorbing surface.

The ratio of heating surface to grate area in the form of smoke-tube boiler just described varies from 10 to 26 : 1, against the cross-tube boiler ratio of about 8 to 10 : 1, and higher combustion rates up to 25 lb. of coal per square foot of grate area are permissible.

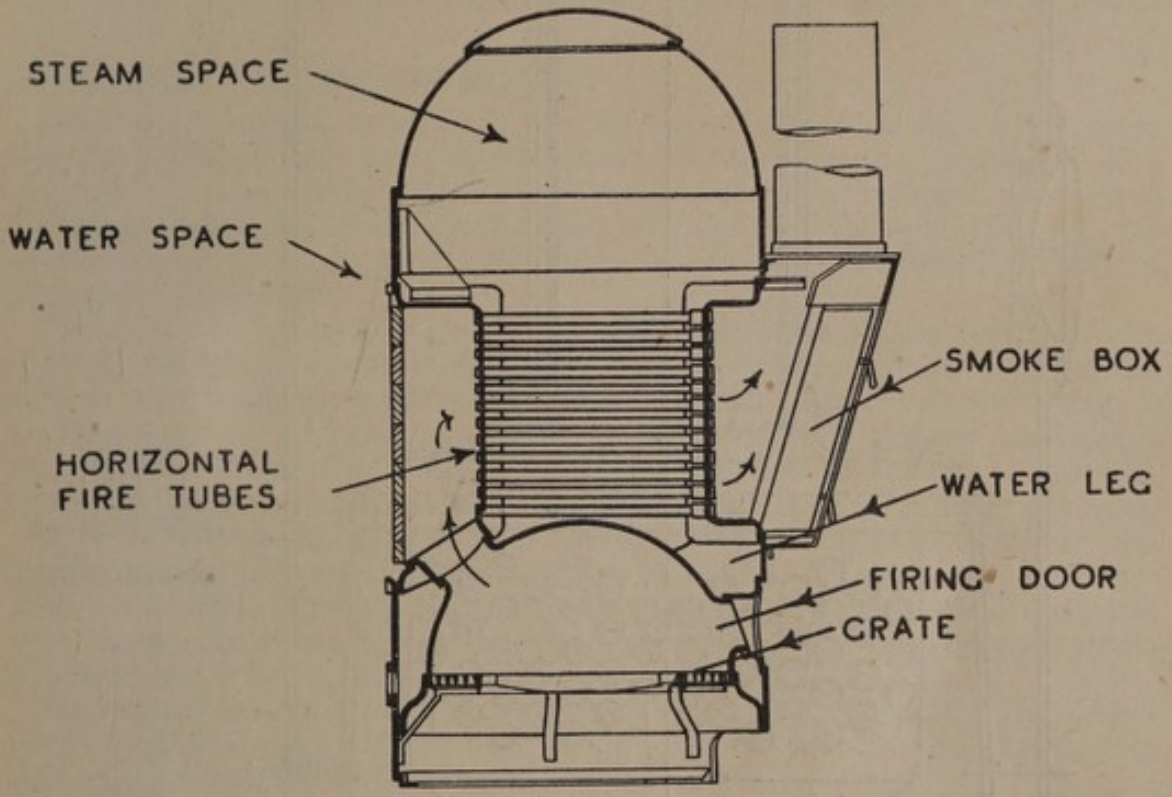

FIG. 96. Vertical fire-tube boiler with horizontal tubes.

HEAT TRANSMISSION IN SMOKE- AND WATER-TUBES

In Chapter VIII the adverse effect of gas films on heat transmission was discussed. In a smoke-tube the amount of heat given up by a hot gas flowing through it depends upon the number of molecules coming into contact with the metallic wall. If the tube is too short, or its diameter is too large, the heat-resisting film of gas molecules adjacent to the wall remains more or less undisturbed, and since its thermal conductivity is about one-fifth of that of a good-quality insulating brick, heat transmission is impaired in consequence.

On the other hand, if water-tubes are suitably disposed in a gas flow, instead of a corresponding nest of smoke-tubes, there is an increased tendency for the gases to " scrub " the external surfaces, so continuously removing the gas film and benefiting heat transmission.

Experiments indicate that a water-tube nest of this type may be up to 80 per cent. more efficient than the equivalent number of smoke-tubes, and, when this principle is applied to the ordinary vertical boiler, it has the important result that the water circulation is increased in the restricted water leg between firebox and shell.

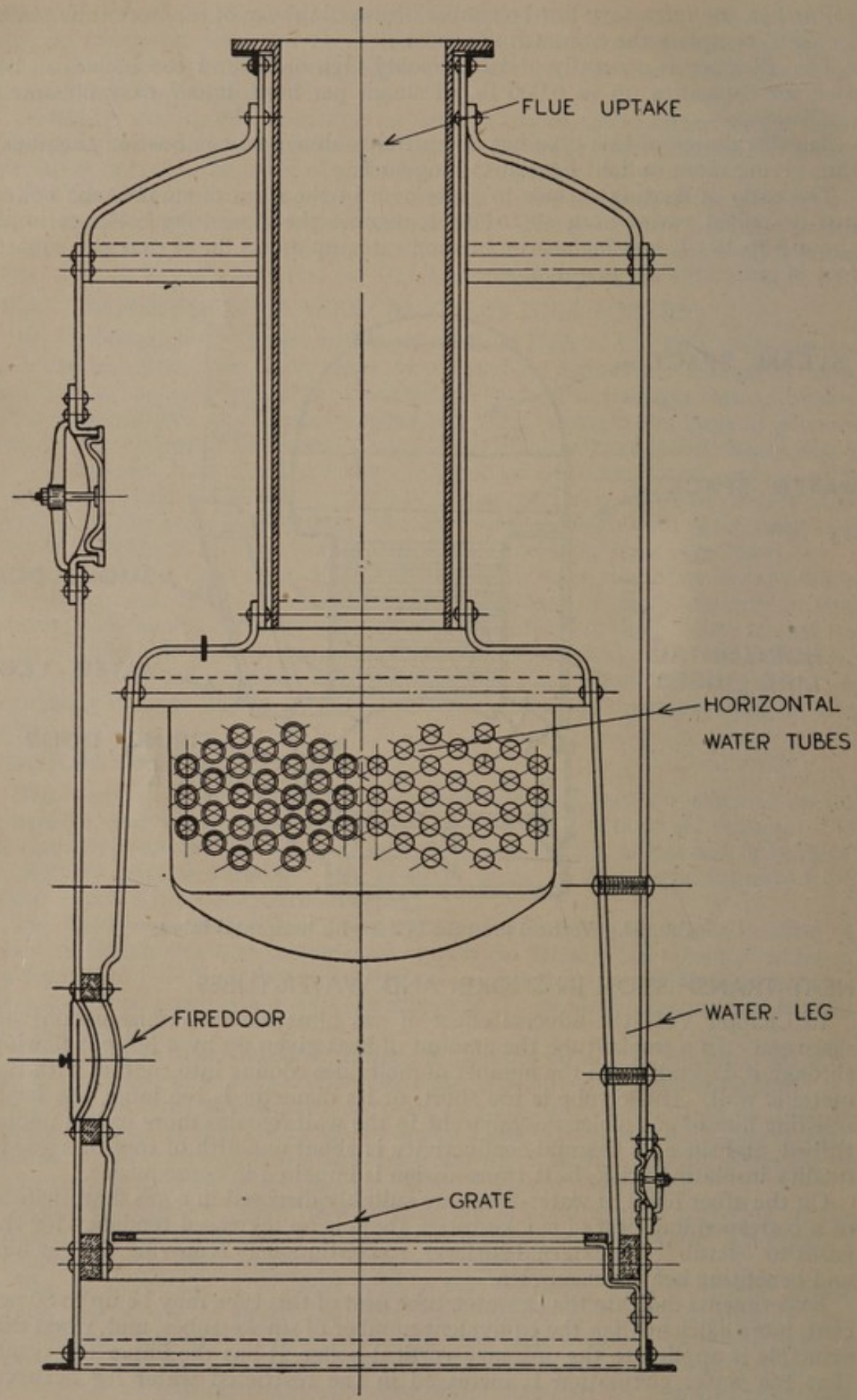

FIG. 97. Vertical water-tube boiler with horizontal tubes.

VERTICAL WATER-TUBE BOILERS

A basic type of vertical boiler with horizontal water-tubes is shown in Fig. 97, the cylindrical vertical firebox being modified to form tube plates for the water-tubes. The lower parts of the firebox and the lower tube rows are subjected to the radiant heat of the fire, while the staggered nest of tubes in the upper part of the firebox is effectively scrubbed during the passage of the gases to the conventional type of uptake.

The water-tubes are given a slight inclination to the horizontal, causing a positive and uni-directional flow of water across the boiler. Steam-raising, in consequence, is fairly rapid, and this type of boiler is obtainable for outputs up to 6,000 lb. per hour. A thermal efficiency of about 70 per cent. can be obtained under reasonably good operating conditions with a well-designed boiler of this type.

STATIONARY LOCOMOTIVE BOILERS

From the foregoing remarks it will be appreciated that one of the most difficult problems facing the vertical boiler designer is the production of a boiler at a commercially suitable price, while providing sufficient convective heating surface to absorb the residual furnace heat resulting from the high rate of combustion necessary in the firebox to keep down the overall dimensions (and cost) of the boiler.

This is a form of vicious circle, and some steam users have turned to the stationary locomotive type of boiler as a solution of the problem when requiring outputs between about 2,000 and 5,000 lb. per hour ; the locomotive boiler of less than 2,000 lb. per hour capacity is rather restricted in cleaning and inspection facilities internally if the barrel is much under 42 inches in diameter. Boilers of this type have been made for producing 9,000 lb. of steam per hour.

Like the vertical boiler, it does not require any elaborate foundations, and it is practically self-contained as shown in Fig. 98.

Its other advantages are that it can be designed to incorporate a comparatively large area of heating surface per unit volume, it is a fairly rapid steam generator and it will stand a limited amount of forcing.

The cubical firebox is water cooled at its sides and top, a construction which allows high heat-release figures per cubic foot, but has the disadvantage of poor water circulation in the surrounding vertical water " legs."

The horizontal dimensions of the firebox are determined from the weight of coal that it is possible to burn per square foot of grate area, but the height is a compromise between the requirements of adequate combustion space, the area required for the tube plate (which, of course, also affects the barrel diameter and cost), and the steam space required above the working water level if priming is to be avoided when steaming.

Combustion rates vary from 16 to 25 lb. of coal per square foot of grate area under natural draught, and from 6 to 9 lb. of water can be evaporated per lb. of coal burned. These rates are, of course, very much less than those obtained in railway practice, where the rate is sometimes as high as 100 lb. per square foot as a result of the induced draught effect obtained from the engine exhaust and the better circulation and movement of the water resulting from vibrations. Stationary locomotive boilers having an evaporative capacity of 4,000 lb. per hour, for example, would require a grate area of about 24 square feet, under natural draught conditions of operation.

The firebox height, as has been already mentioned, is limited in a boiler of this type because of its encroachment on the steam space, and also to a lesser extent, because of expansion arising in service from the use of high fireboxes. A cubical firebox is not the ideal form for withstanding high temperatures and pressures, and undue forcing may result in strained seams and tube-plate

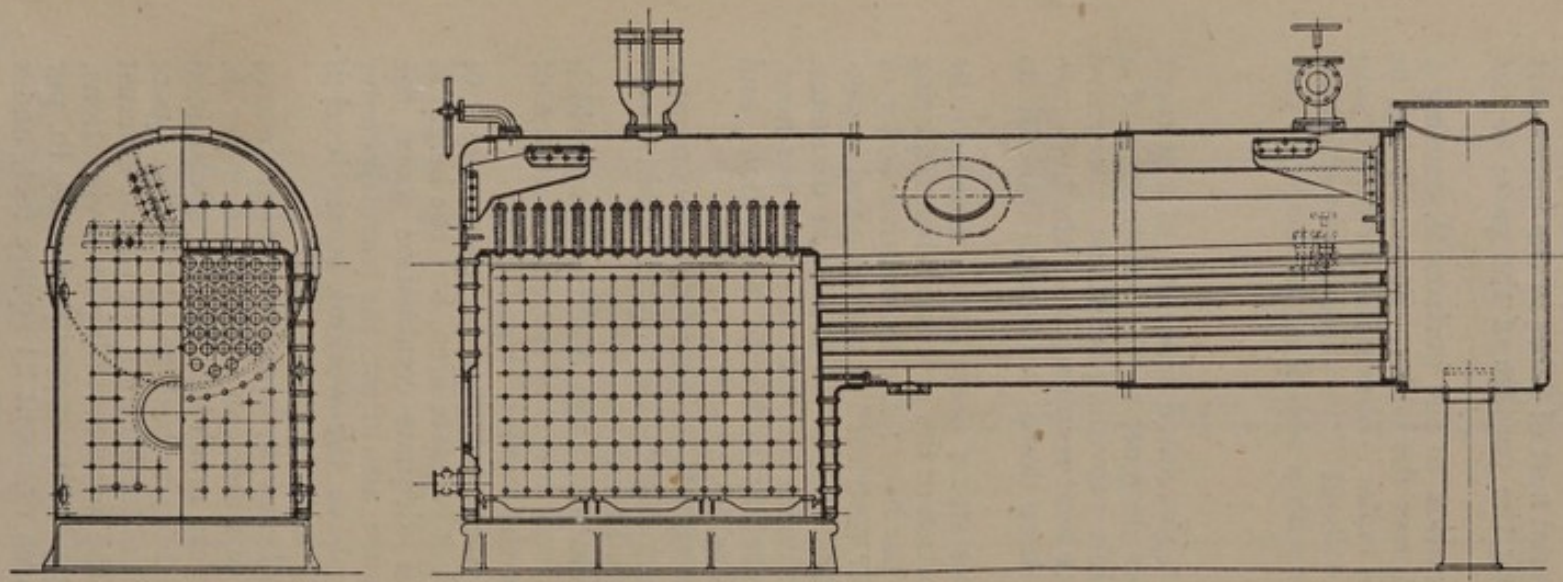

FIG. 98. Stationary locomotive boiler.

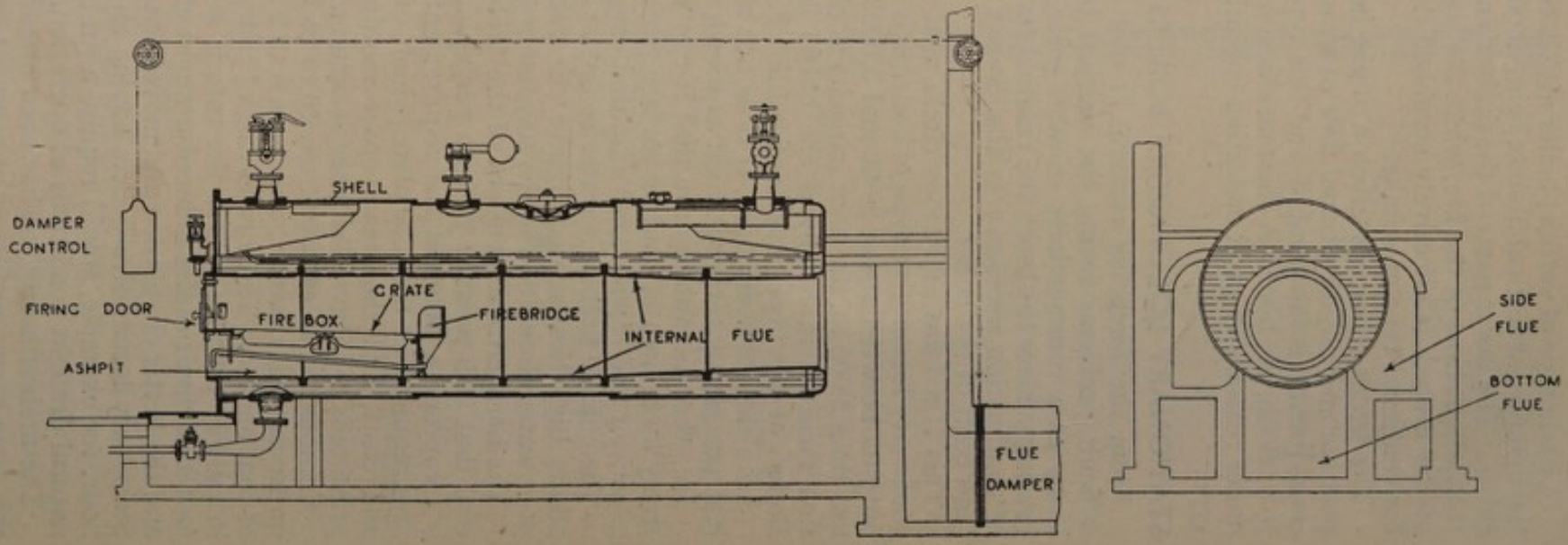

FIG. 99. Cornish boiler.

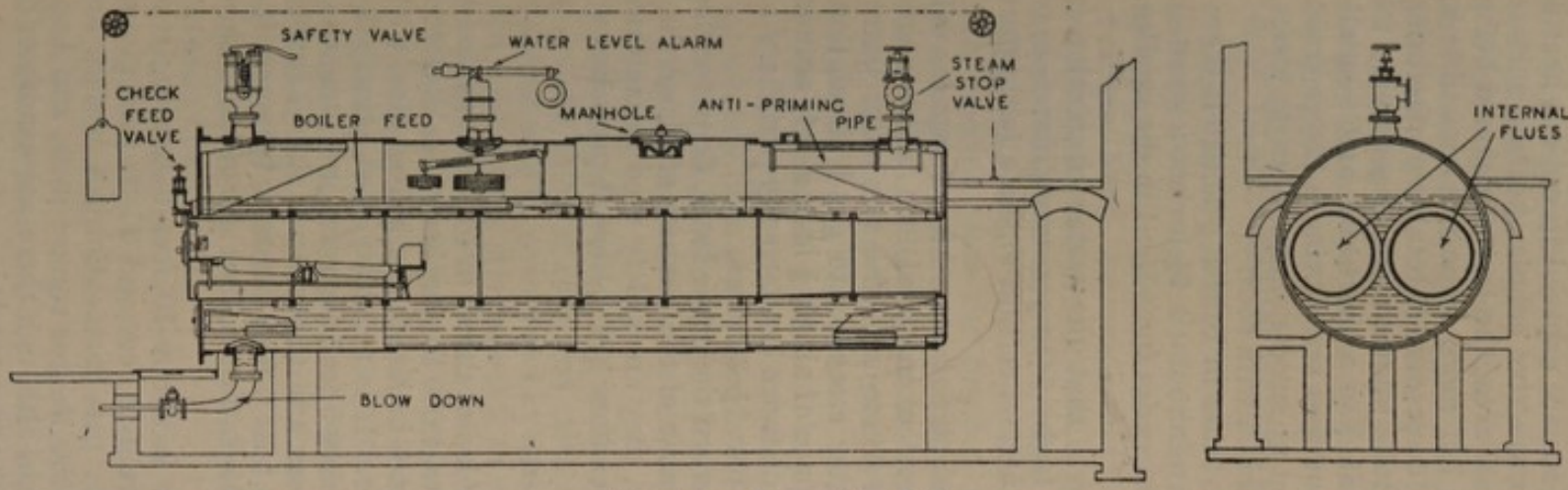

FIG. 100. Lancashire boiler.

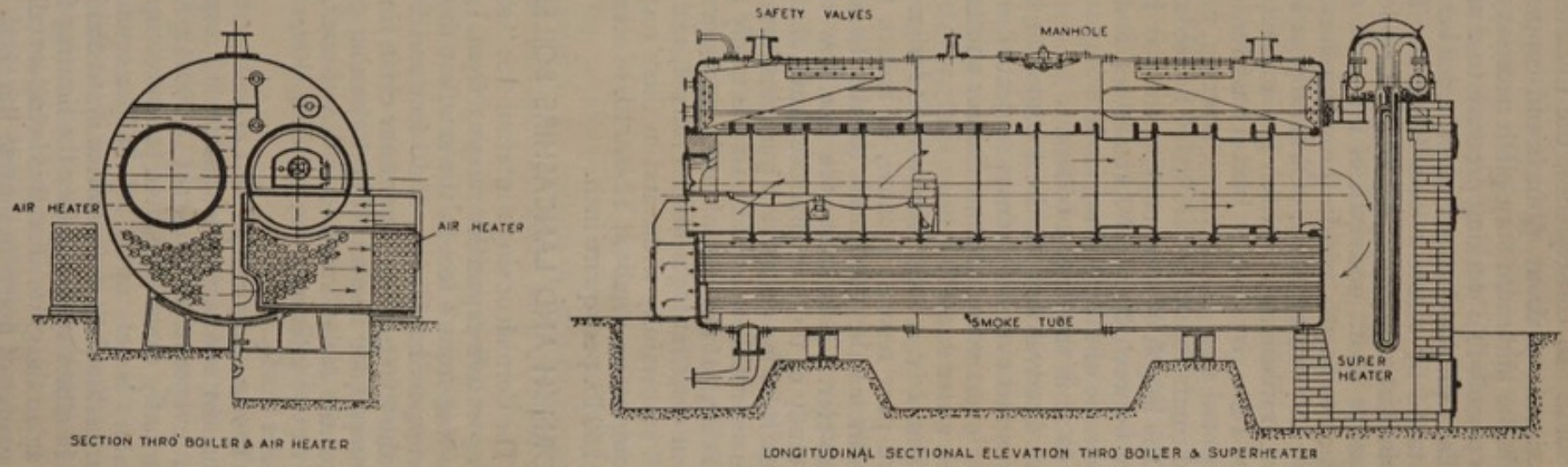

FIG. 101. Modified Lancashire boiler.

troubles. In consequence, the thickness of steel fireboxes is restricted to a maximum of about $\frac{9}{16}$ inch and copper fireboxes to about $\frac{5}{8}$ inch, and this, with the minimum stay pitch necessary for flexibility plus strength, restricts the permissible working pressure to a maximum of 300 lb. per square inch.

Thus the maximum area of boiler heating surface exposed to radiant heat is curtailed by constructional features. The convective tubular heating surface is similarly limited in regard to cross-sectional area, the last mentioned being equal to about one-sixth of the grate area, but more scope is permissible in regard to length, within the requirements of a reasonable total manufacturing cost and draught loss. Obviously, a long tube allows more opportunity for heat transmission between gas and tube wall than a short one, and a small-bore tube gives better heat transference than does one having a larger cross-sectional area. Long tubes, however, may set up uneconomical draught losses, and small-bore tubes are liable to become choked with soot.

The length of stationary locomotive boiler tubes in British practice is therefore between 30 and 50 diameters, and the minimum is $2\frac{1}{4}$ inches (external). The ratio of total heating surface to grate area varies from 15 : 1 in the smallest to 40 : 1 in the largest boilers.

A disadvantage of the locomotive boiler is that the water circulation is restricted over those areas subjected to the highest temperature difference, so that overheating at some time or another is almost certain unless a fairly pure feed water is used.

Similarly, a boiler of this type designed and manufactured for cheapness will give poor results, because, to reduce the cost of material, the boiler-maker assembles the maximum tubular heating surface into the minimum barrel diameter. This limits the free flow of water essential to good circulation. A study of the best practice shows that a space of about $\frac{3}{4}$ inch is left around each tube for circulation purposes, and the distance between the periphery of the outer tubes and the barrel is rarely less than $2\frac{1}{2}$–3 inches.

Boilers of the Cornish and Lancashire class, on the other hand, do not suffer from disadvantages of this nature, and, because of the amount of abuse they will successfully withstand in these respects, they are widely used in factories requiring up to about 12,000 lb. of steam per hour per boiler at pressures up to 250 lb. per square inch.

CORNISH AND LANCASHIRE BOILERS

The Cornish or single internal-flue " shell " boiler shown in Fig. 99 is made for evaporative capacities ranging from 1,000 to 4,000 lb. per hour, the diameter varying from 4 feet in the smallest to 6 feet 6 inches in the largest size, with corresponding lengths of 10 feet and 24 feet. It is probably used principally in the 2,000–4,000 lb. per hour class, since below 5 feet diameter it is somewhat cramped for inspection and cleaning purposes and the internal flue diameter becomes rather small for efficient operation. At 6 feet diameter it is possible to fit two internal flues, as in Fig. 100, and this is the smallest size of Lancashire boiler made.

The maximum size of Lancashire boiler is about 10 feet diameter; the smallest and largest lengths being respectively 18 feet and 30–32 feet, giving an evaporative range of 3,000–12,000 lb. per hour. Cornish boilers are normally constructed for pressures up to 160 lb. per square inch, and Lancashire boilers up to 200 lb. per square inch, although there are examples of Lancashire boilers in service up to 260 lb. per square inch.

In either type of boiler the limiting factor in regard to evaporative capacity is again the grate area, this being curtailed in the smallest size by the permissible diameter of flue fitted inside the shell, and in the larger sizes by the length of grate that can be properly worked by hand.

In a natural draught boiler this restricts the grate area to a maximum of about 50 square feet divided between two flues, and a combustion rate of 25 lb. of coal per square foot of grate area. Higher combustion rates are possible by using mechanical stokers and mechanical draught, but the convective heating surface of a Lancashire boiler is such that the addition of an economiser is necessary to obtain a good overall efficiency.

The horizontal shell class of boiler, however, has the important advantage that a reasonably maintained unit may have a useful working life of upwards of 40 years ; its surfaces are readily accessible for inspection and cleaning ; and its operation may be entrusted to a low grade of labour, although this is not advised from a fuel economy point of view. Good results on any boiler above the simplest vertical types require a skilled man as fireman.

HEAT TRANSMISSION IN CORNISH AND LANCASHIRE BOILERS

The ratio of heating surface to grate area is usually from 20 to 30 : 1, but some 50–60 per cent. of the evaporation of an internal-flue boiler is considered to be produced over the furnace portion, the bulk of this being effected by means of radiant heat. Assuming that the length of internal flue in which the radiant heat effect is most pronounced is approximately 15 feet, then it will be apparent that about 25 per cent. of the total heating surface provided is thus employed.

The remaining 75 per cent. of the heating surface is intended to absorb the heat rejected from the furnace, but this surface must be supplemented by superheater, economiser and/or air heater heating surface if thermal efficiencies of the order of 75 per cent. are desired.

The internal furnace is completely water cooled, allowing large heat-release figures per unit volume, but it also has the disadvantage that any irregularity in the combustion conditions may result in heavy smoke.

The importance of the firebed surface in furthering radiant heat transmission has already been mentioned in Chapter VIII, and since the transmission varies as the fourth power of the absolute temperature, it is worth repeating that a bright, hot firebed must be maintained at all times if the best results are to be obtained.

The use of mechanical stokers, mechanical draught and preheated air is therefore to be encouraged, although the application of these to an existing boiler should be considered in consultation with the makers of such equipment. The heat absorption capacity of an internal flue has limits, and, if these are exceeded, structural defects may develop.

The convective heating surface of a Cornish or Lancashire boiler consists in the main of curved surfaces arranged over external flues, the products of combustion being constrained to flow through the external flues during their passage to the chimney flue. A great deal of heat transmission by gas radiation effects occurs, especially in the first pass below the boiler (where temperatures up to about 1,000° F. occur), and the effects are doubtless assisted by the turbulence resulting from the change in direction of the gases when they leave the internal flues.

Apart from the nature of the surface provided, the design of the external flues does not lend itself to high efficiency. The flues are proportioned to give bodily access for cleaning, etc., as the primary object, and as a result, it is probable that in many boilers layers of gas have only a brief contact—if any—with the external shell on their way to the main flue.

Moreover, the brickwork walls of the external flues must be heated every time steam is raised, and they continue to absorb heat during the steaming period, in proportion to the radiation losses of the setting as a whole.

A defective setting can also cause considerable loss by allowing the gases to be short-circuited directly from the flue below the boiler, into one of the side

flues. A 30-feet long Lancashire boiler expands longitudinally about $\frac{3}{4}$ inch when heated, and the movement eventually causes spaces to form between the shell and the back end and main seating brickwork, through which the gases flow into the side flues and thence straight into the chimney without giving up heat. Similarly, leakage spaces develop at the front end of the boiler and allow air to be drawn into the main flow of gas, with detrimental results. Cracks in the setting walls have the same effect, the importance of which has already been emphasised in Chapters VI and XI.

MODIFIED TYPE OF LANCASHIRE BOILER

Defects of this nature have caused at least one British boiler manufacturer to develop an arrangement with a Lancashire type of boiler in which the inefficient external flues are dispensed with. In the modified design the internal flues are slightly smaller and are set rather higher in the shell than is customary, and in the space below them are arranged suitable nests of smoke-tubes, as indicated in Fig. 101. The tubes pass from endplate to endplate, so that the products of combustion discharged from the main internal flues flow through tubes in the water space of the boiler.

The internal flues are proportioned to allow of maximum radiant heat transmission, while the smoke-tube cross-sectional area is such that the gases flow through at a high speed, thus increasing the convective heat transmission.

On leaving the smoke-tubes at the front end of the boiler the gases are divided into two approximately equal streams, and are then caused to flow through tubular air heaters arranged at ground level, on each side of the boiler shell. Most of the remaining usable heat is thus abstracted.

Balanced, forced or induced draught, is employed in this boiler, and an efficiency of 83·3 per cent. is stated to have been obtained in tests. With the exception of the downtake lining (which has a patented air-excluding device) no brickwork setting is required.

EXTERNALLY-FIRED MULTITUBULAR BOILERS

In externally-fired boilers of the horizontal shell class the internal flues are omitted and the fire-grate is arranged externally below the lower part of the shell, as illustrated in Fig. 102.

The surface normally used for convective heating in the Lancashire boiler

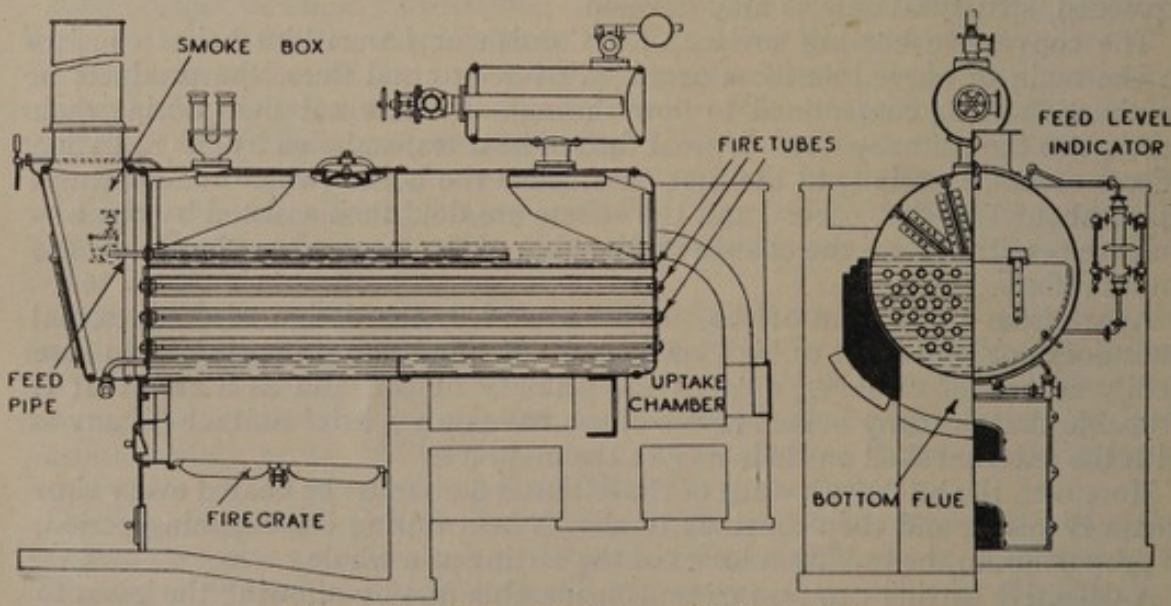

FIG. 102. Externally-fired multitubular boiler.

is in this form used principally for radiant heat transmission. The advantages of this are that the furnace portion can be designed to burn various kinds of waste fuel that could not be readily handled on the restricted area of an internal-flue boiler, and the water circulation along the bottom of the shell is greatly improved.

Usually the externally-fired boiler is mounted upon a brickwork setting, and the products of combustion flow along the bottom of the shell into an uptake chamber, one side of which consists of the back endplate of the boiler.

The back and front endplates are perforated to take smoke-tubes arranged longitudinally along the shell, and the gases flow through the tubes. The discharge from the tubes may be led into a chimney uptake, or guided back along the sides of the shell by means of suitable external flues.

The great disadvantage of the externally-fired boiler is that it requires reasonably good feed water, and, if this condition is not met, overheating of the shell plates along the bottom is almost inevitable when deposits accumulate.

Externally-fired boilers of this class are constructed for evaporative capacities between about 1,000 and 9,000 lb. of steam per hour at 160 lb. per square inch. Ratios of heating surface to grate area in the brick-set type vary from 30 to 70 : 1 ; the corresponding ratios of tubular heating surface to shell surface being from 3·5 to 11·5 : 1.

THE ECONOMIC BOILER

The externally-fired multitubular boiler has never been employed in Great Britain to the extent that it has in the U.S.A. ; British factory engineers apparently prefer to use other forms of horizontal shell boilers.

The disadvantages of the Lancashire boiler in regard to space occupied and its low efficiency necessitating the provision of an economiser led to the introduction of the Economic boiler and many engineers turned to this internally-fired type of boiler as a solution of their problem.

This class of boiler, for a given output, occupies only about one-half the space of a corresponding Lancashire boiler without economiser, and can be constructed as a self-contained unit, without a brickwork setting. A Lancashire boiler capable of evaporating 10,000 lb. per hour, for instance, would be 9 feet in diameter by 30 feet long, whereas an Economic boiler having a similar output would be 9 feet in diameter by only 15 feet 6 inches long.

In making comparisons of space occupied it should be borne in mind that with an Economic boiler space must be provided for cleaning, and, if necessary, replacing the tubes, and this makes the approximate overall length required twice the length of the tube, plus the combustion chamber, and plus again any space required behind the combustion chamber for access purposes. On the other hand the Lancashire boiler requires the overall length of the setting and such flues as come inside the boiler-house, plus some 8 or 9 feet for firing space.

The shorter shell of the Economic boiler also reduces the initial and maintenance costs of brickwork settings, where these are deemed necessary. Very few brick-set Economic boilers are, however, now installed as it has been found that no improvement in either efficiency or evaporation is obtained and maintenance costs are saved. One advantage of brick side flues is that they act as grit arresters.

With these boilers, under good conditions of operation, an average of up to 75 per cent. efficiency can be expected and with experienced stoking and suitable draught to burn the coal, even higher efficiency is obtainable without the use of economisers, air-heaters or superheaters.

The Economic boiler consists essentially of one or more furnace tubes, arranged in a cylindrical shell (see Fig. 103), the gases from the furnaces being discharged into an external combustion chamber. The combustion chamber

is refractory lined to assist the completion of the process. The gases then flow to the front end of the boiler through smoke-tubes passing from endplate to endplate. Usually one, but often two, and sometimes three " passes " of tubes are employed to achieve the maximum convective effect, but it will be appreciated that the multi-pass construction results in greater draught losses, and generally necessitates the application of induced draught.

As in all internally-fired boilers, the evaporative capacity depends on the heating surface and the grate area and the draught conditions to burn the fuel required. The larger sizes of Economic boiler, however, are provided with mechanical stokers working in conjunction with mechanical draught, and up to 40 lb. of coal per square foot of grate area can then be burned, giving an evaporation of about 10 lb. of steam per lb. of coal.

This class of boiler is constructed for evaporative capacities ranging from 750 to 30,000 lb. of steam per hour, and for maximum pressures of 250–300 lb. per square inch according to the size of the boiler. The boilers vary in

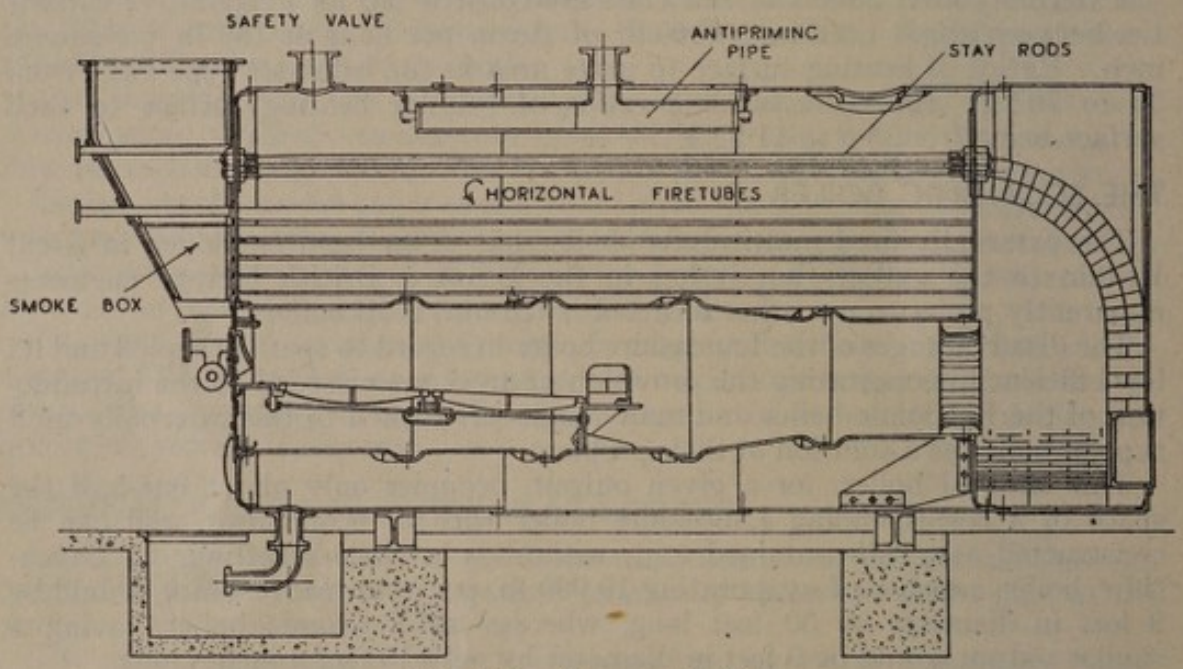

FIG. 103. Economic boiler.

size from 4 feet 6 inches diameter by 7 feet 6 inches long in the smallest, to 13 feet diameter by 20 feet long in the largest, the combustion chamber adding another 2–4 feet to the overall length.

In the conventional design of Economic boiler a single furnace tube is fitted for evaporations up to about 4,000 lb. per hour, and outputs up to 25,000 lb. per hour are obtainable with two furnace tubes. For capacities higher than this it may be advisable to consider the adoption of a water-tube boiler, but if local conditions justify the retention of a shell type, there are Economic boilers capable of generating up to 30,000 lb. of steam per hour with three or four furnaces.

As in other internal-flue boilers, a major portion of the total evaporation is effected over the furnace heating surface, which, in the Economic class, amounts to about 13 per cent. of the total heating surface.

ECONOMIC BOILER DESIGN

In the later designs of Economic boilers the combustion chamber is made as an integral part of the boiler and is brick lined and lagged with the boiler and thus the radiation losses and brickwork troubles are eliminated. The refractory

lining in the chamber tends to promote more thorough combustion, the process being assisted by the turbulence resulting from the flow of gases around the chamber. Combustion will be completed in the internal flue if the quantity of air admitted to the furnace is correct for the amount of fuel being fed on the grate, and the quantity of fuel supplied is correct for the area of grate. It is desirable that combustion should be completed before the gases reach the chamber. The depth of the chamber should be adequate for the free flow of the gases and their change of direction, this depth depending on the class of fuel used and the size of the boiler for the required evaporation.

The amount of tube area provided for the flow of gases from the combustion chamber is a function of the tube length per diameter ratio adopted, of the ratio between the heating surface and grate area and of the total cross-sectional area required to keep the draught loss within economical limits.

The amount of tube area for the flow of gases from the chamber should be such as to give the correct velocity required for heat transfer. The length of the tube can then be made such that the temperature of the gases is reduced by the desired amount in passing from the combustion chamber to the exit of the boiler.

Generally speaking, a length of 15–16 feet is about the practicable maximum for the ordinary type of smoke-tube. A tube diameter of $2\frac{1}{2}$–3 inches is the most suitable to allow for proper tube spacing and to give freedom from rapid choking with soot, bearing in mind that small tubes give a greater heating surface per unit of boiler volume occupied than large ones.

The greatest length per diameter ratio is therefore about 70 : 1. This may be reduced to half this figure in the small sizes of boiler.

The ratio between total heating surface and grate area depends upon the designer's experience and practice, and varies from about 20 : 1 to just over 60 : 1.

The designer's task has been to fit the desired area of heating surface into the smallest (and therefore least expensive) size of shell, while allowing at least 6 inches of clear space around the furnace or furnaces and between the smoke-tubes and shell, as well as about 1 inch to $1\frac{1}{4}$ inch around each tube periphery. At the same time, he must also provide a suitable ratio of steam space to water space—usually $\frac{1}{2}$ to $\frac{1}{3}$—and ensure that there is sufficient steam disengaging area at the normal working water level in the boiler, or priming difficulties may arise under working conditions.

In addition, the empirically determined ratio of total tube cross-sectional area to grate area must be maintained within the limits of one-tenth to one-fifth common in British practice. This task, of course, has in the past necessitated a considerable number of experiments in laying out specimen tube plate arrangements, etc., and these have now been incorporated in the tables of standard sizes offered by all reputable boiler makers.

MODIFIED ECONOMIC BOILER TYPES

Within recent years a few manufacturers, however, have modified their standard designs with a view to eliminating a great disadvantage of all internally-fired horizontal shell boilers, i.e. poor circulation along the bottom, in addition to improving the heat transmission generally.

Many of these modifications are not really new, being adaptations of arrangements previously used. They consist principally of placing tubular heating surface in the lower part of the shell, coupled with increased gas velocities on the fire side. Mention can here be made of the two- and three-pass boilers, the latter having a specially designed combustion chamber. The better disposition and use of the available heating surface have undoubtedly resulted in enhanced performance and increased efficiency, which under working conditions is often

of the order of 80–82 per cent. The introduction of a further bank of smoke-tubes necessitates the use of an induced draught fan which, however, enables complete regulation of the draught to be maintained with a lower height of chimney with slightly increased operational costs for the fan power.

One or two makers have produced improved forms of smoke-tube having a sinuous form for which greatly increased rates of heat transmission over a given length are claimed in comparison with the normal plain tube.

One maker has also adopted the completely immersed combustion chamber characteristic of the Scotch marine boiler, thus eliminating the external combustion chamber losses, but he modifies the design by making the boiler of the two-pass type.

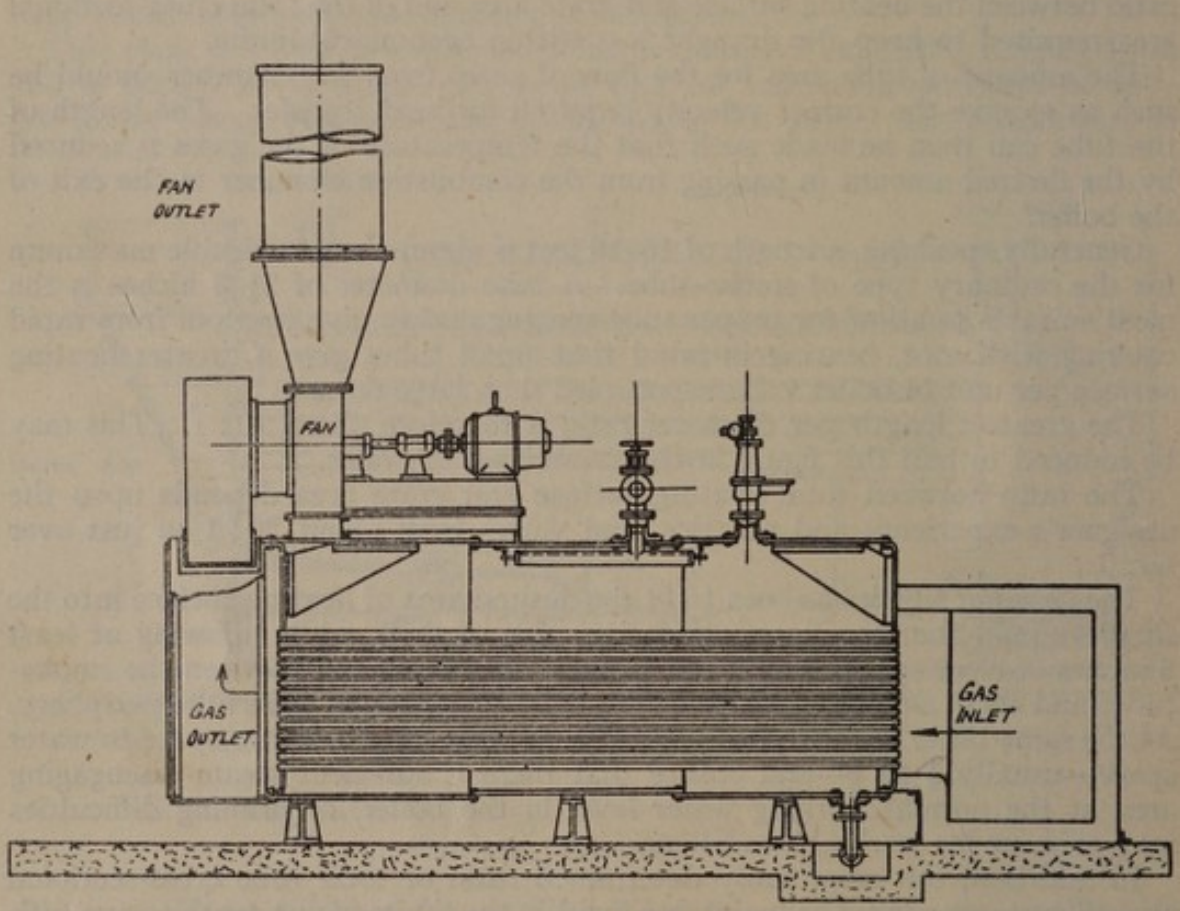

FIG. 104. Waste heat boiler.

WASTE HEAT BOILERS

Before leaving the subject of shell boilers, mention must be made of the waste heat type. In many manufacturing operations volumes of furnace gases at temperatures of the order of 1,000° C. and less are surplus to the main manufacturing process, and can be utilised for the generation of steam. Boilers of the water-tube class were at first employed for the purpose, but it was found that more satisfactory results could be obtained from a cylindrical horizontal smoke-tube boiler.

In some industries steam raised from waste gases is required at high pressure, that is to say, beyond the limit for which the shell boiler is suitable and the water-tube boiler is still installed to meet these conditions ; this also applies where the gas quantities are very large.

The gases to be used are led in special ducts to a refractory-lined inlet chamber and then flow through smoke-tubes expanded into the front and back tube-plates and situated within the water space of the boiler. The cooled

gases are collected in an outlet chamber before being drawn off to the chimney. A diagrammatic view of a waste heat boiler provided with an induced draught fan is shown in Fig. 104.

Since the temperature and the hourly volume of the gases available are capable of almost infinite variation over the field of industry, the standardisation of such boilers is somewhat difficult. Each boiler must therefore be designed with particular conditions in mind, but there are a few fundamental factors that are common to all tubular waste heat boilers.

The heat transmission is accomplished chiefly by convection, so that the insulating effect of gas films on the tube walls must be overcome by increased gas velocity. A fairly high gas pressure drop across the boiler is necessary to achieve this object, and an induced draught fan is thus an almost essential auxiliary. The sudden increase in velocity as the gas enters a tube mouth results in considerable additional turbulence extending for some distance into the tube, the extent of the effect depending upon the magnitude of gas pressure drop considered to be permissible in any given circumstances.

The ratio of tube length to diameter is also of importance as the gas must be allowed sufficient time to impart its heat to the water, and ratios up to 150 : 1 have been applied. In modern boilers the ratio may be in the neighbourhood of 120 : 1, but when sinuous tubes are fitted it may be reduced to as low as 66 : 1, with consequent savings in boiler size and constructional cost.

Again, since evaporation takes place on the wetted surfaces of the tubes, space must be allowed for the free release of steam bubbles and the unretarded circulation of water. The tubes are in consequence fairly widely pitched, horizontally and vertically, and are provided with circulation lanes extending the full depth of the shell. Ample area is also required between the peripheries of the outer tubes and the shell.

Within the last decade or so boilers of the thimble-tube and forced-circulation coil types have been used for waste heat utilisation, but as these applications are in the main in connection with internal-combustion engines, they will not be discussed here. Waste heat boilers are discussed in detail in Chapter XX.

SHELL AND WATER-TUBE BOILERS

The shell boiler, because of its constructional characteristics, is normally limited to a maximum working pressure of about 250 lb. per square inch, although a few have been constructed for 300 lb. per square inch ; any higher pressure imposes the use of abnormally thick shell plates, when maximum evaporative capacity is also required. Similarly, shell boilers are generally restricted in land practice to an output of about 20,000 lb. per hour in a single boiler, although in marine practice the Scotch type of multitubular boiler is designed for outputs in excess of that figure.

The water-tube boiler can be constructed for capacities varying from a few hundred to more than a million lb. of steam per hour, and for all steam pressures up to the critical value of 3,200 lb. per square inch.

For the smaller steam generating plants in making the choice between shell and water-tube boilers for an installation, consideration would need to be given to efficiency, relative first and operational costs, working pressure required, load fluctuations and local conditions. For large installations, however, where high pressures and large outputs are required, the selection must inevitably fall on the water-tube boilers.

Before leaving shell boilers, mention should be made of the considerable advance which has been made in recent years in the design and operation of furnaces and stokers for horizontal shell boilers. These have undoubtedly contributed to the high efficiencies reached, enabling coals of a very wide

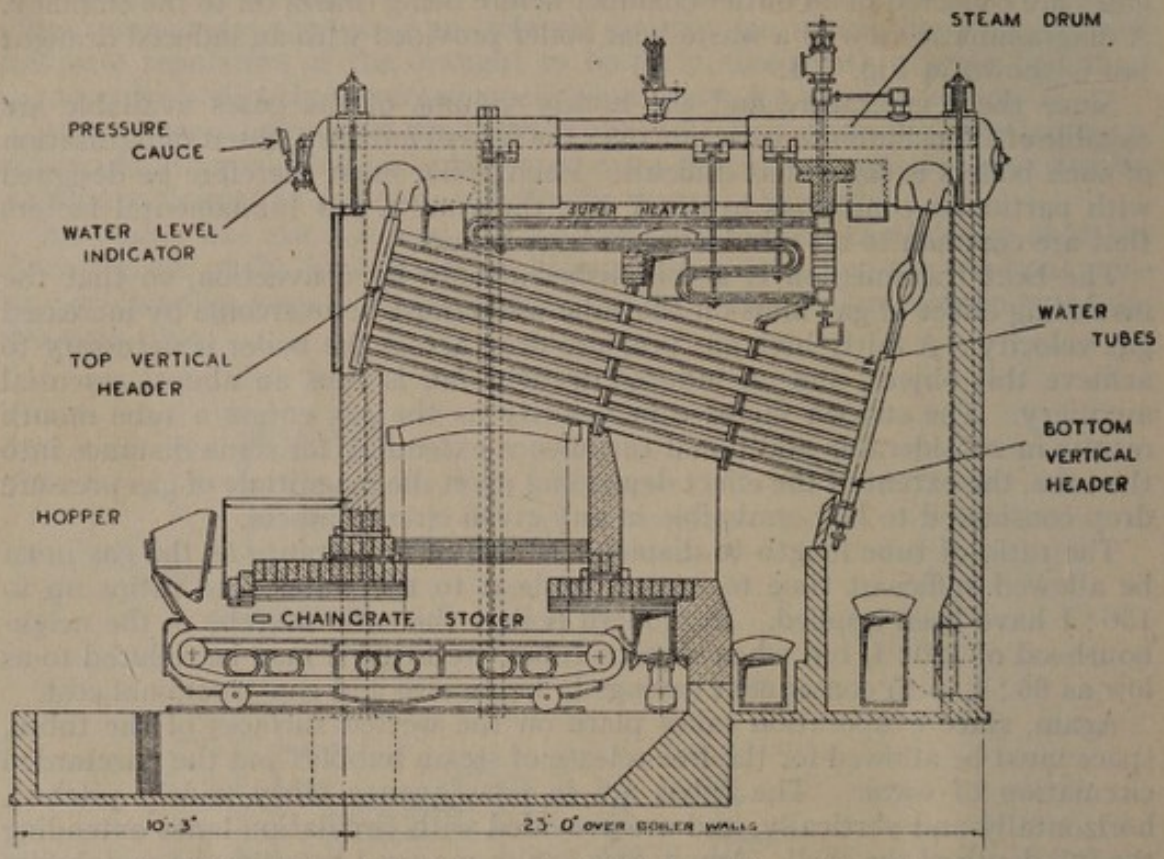

FIG. 105. Water-tube boiler—straight tubes with sectional headers.

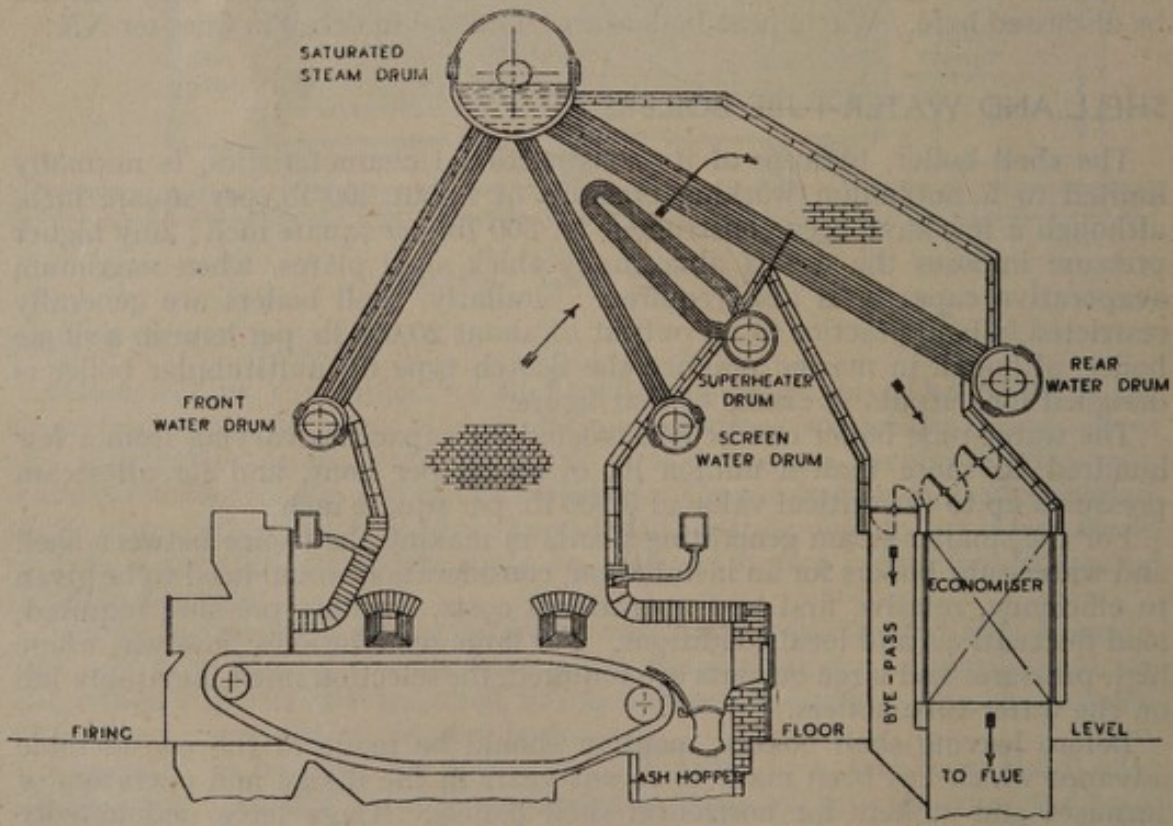

FIG. 106. Water-tube boiler—tubes expanded into drums.

range, both as to quality and grade, to be used in them. Mechanical stokers have also greatly simplified the problem of firing large boilers.

Shell boilers are very easily adapted for oil firing, and very many of them are installed for heating institutions, large buildings and workshops, the boilers being used for generating hot water at high or low pressure or steam ; some are fired by gas or electricity.

WATER-TUBE BOILER TYPES

There are three principal types of water-tube boiler used in this country at the present time ; the straight-tube, the bent-tube and what might be termed the forced circulation boiler.

The first is exemplified in Figs. 105 and 106 ; the former being a sectional

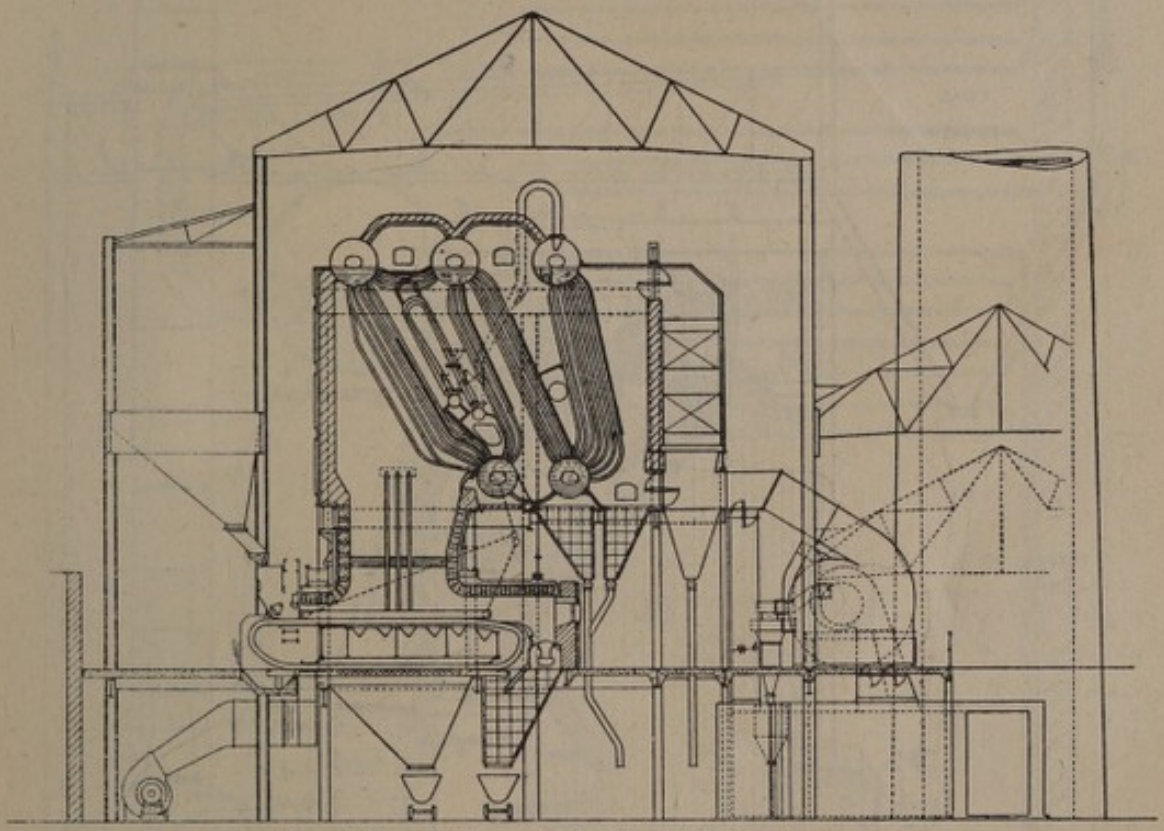

FIG. 107. Water-tube boiler with five drums.

header straight-tube boiler and the latter a vertical straight-tube boiler with tubes expanded into suitable drums.

The second type, the vertical bent-tube boiler, is shown in the five- and three-drum arrangements in Figs. 107 and 108 ; the number of drums may be reduced to two or increased to four or more, according to the purpose for which the boiler is intended.

The third type is to be found in boilers such as that shown in Fig. 109, the heating surface of which consists principally of water tubes suitably connected to form a continuous length.

EFFECT OF PRIME MOVER ON BOILER DESIGN

The development of the water-tube boiler from the comparatively small type of unit in use at the beginning of the present century to the modern high-pressure, high-capacity steam generator, has resulted largely from the demands made by steam-turbine designers for an increased heat content per lb. of steam flowing through the turbine stop valve.

The thermal efficiency of a steam turbine depends upon the difference between the initial and the final temperature of the steam supplied to and exhausted from the machine, and since the lower temperature is at its economic

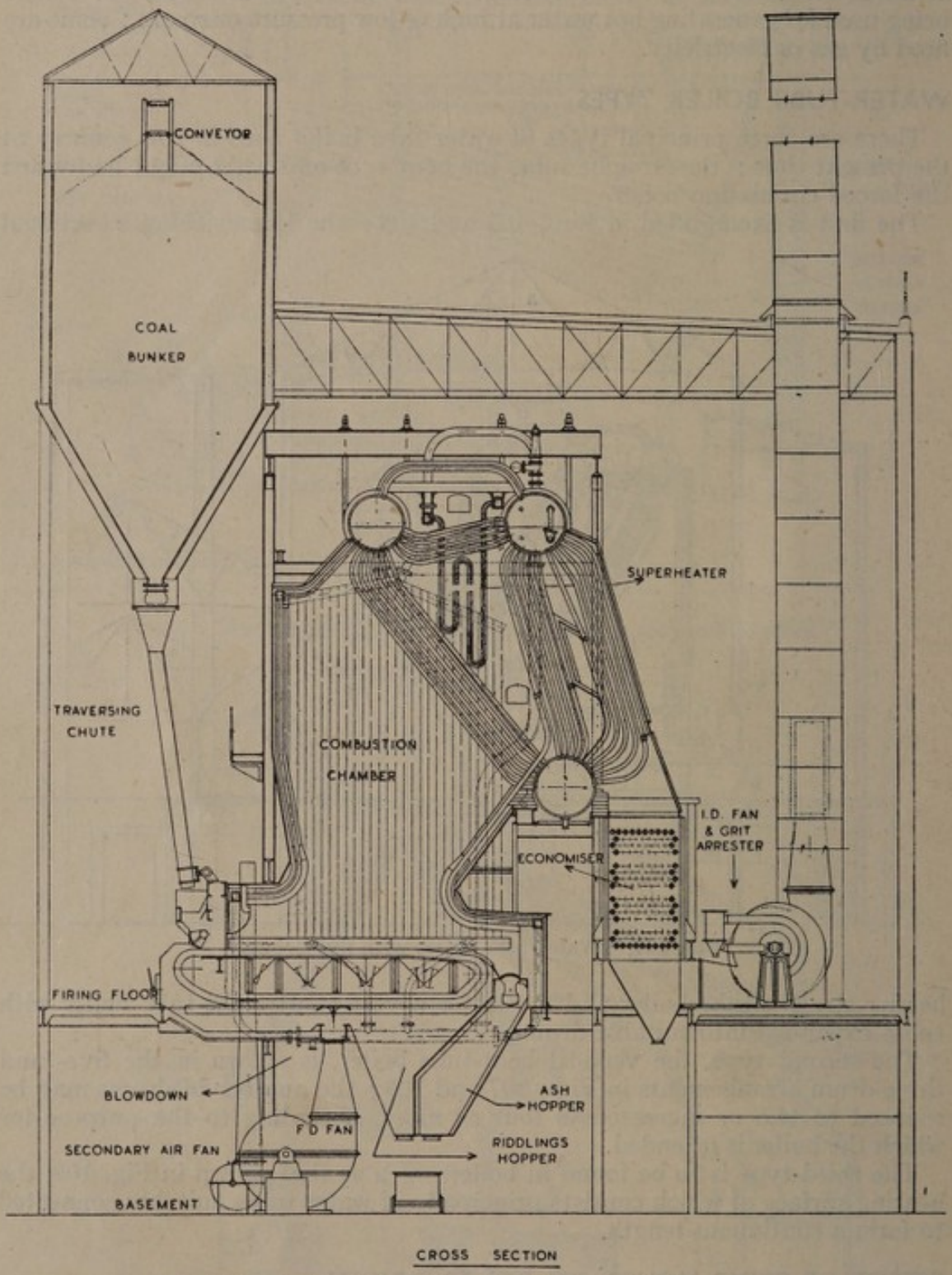

FIG. 108. Water-tube boiler—bent tubes with three drums.

limit, improvement has had to be obtained on the boiler side. An increase in steam pressure raises the initial temperature, but the usefulness of higher pressures is limited by the amount of condensation permissible at the turbine exhaust, the moisture content in good practice being about 12½ per cent.

The stage in a given expansion range at which condensation begins can be retarded by the imposition of superheat, or by re-heating the steam, so that pressures up to the critical value and steam temperatures up to 1,000° F. have been applied in service. It will be appreciated, however, that unless the steam is re-heated between the high-pressure and low-pressure stages of the turbine, a definite relationship of initial steam pressure to initial total steam temperature

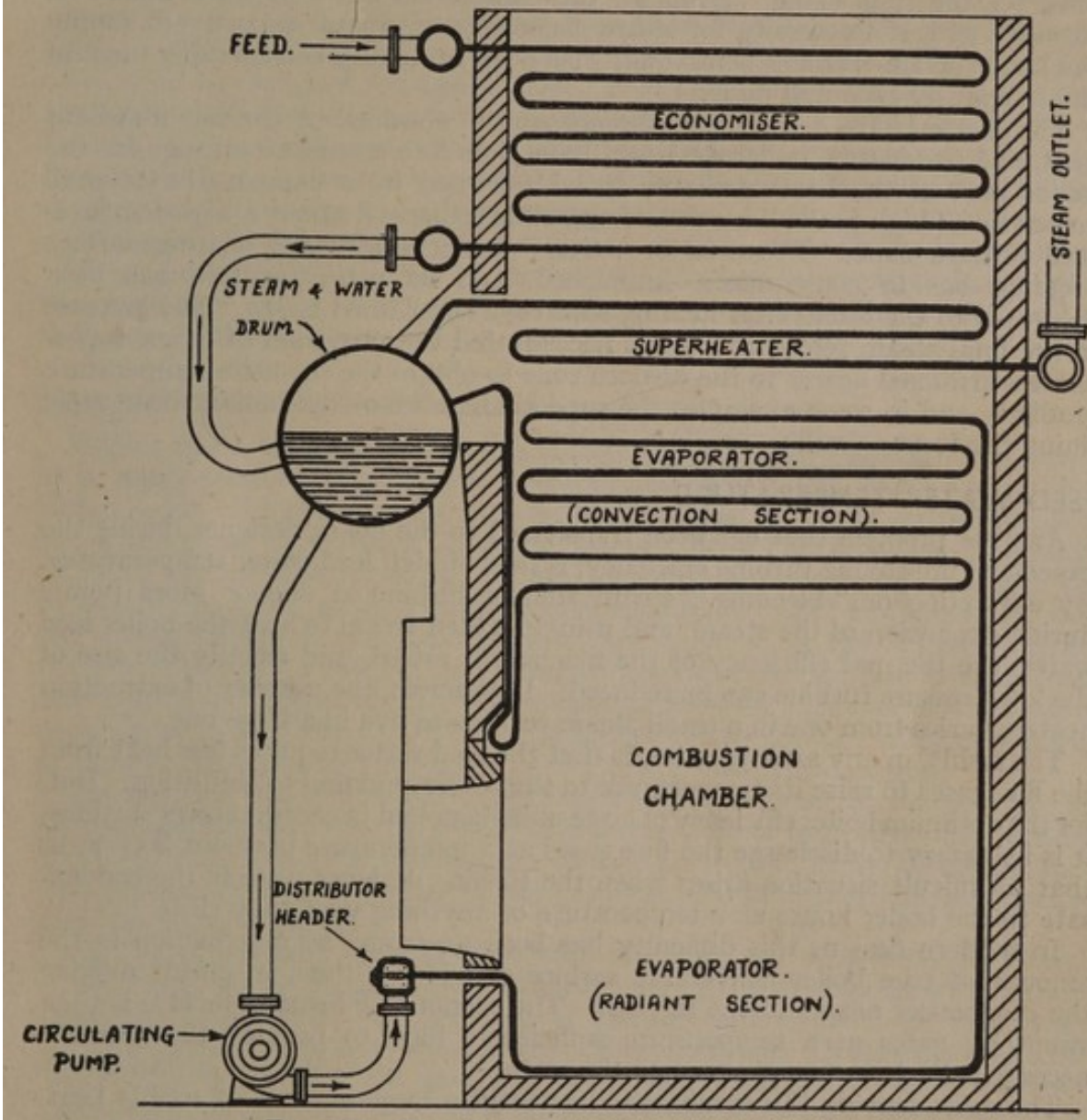

FIG. 109. Water-tube boiler—forced circulation.

must exist if excessive condensation at the turbine exhaust is to be avoided. Sir Leonard Pearce recently stated that for normal turbine conditions the steam temperature and pressure relationship is approximately as set out below.

Steam temperature °F.	Steam pressure lb. per sq. in.
750	580
800	750
850	950
900	1,200
950	1,500
1,000	1,900

To meet turbine requirements, steam pressures and temperatures have been gradually increased until at the present time boilers working at 650 lb. per square inch and 800° F., are relatively common in British power-station and industrial practice. Higher pressures and temperatures than those cited are also in use, but the properties of the commercially available superheater tube materials render it advisable to halt at the 825°–850° F. steam-temperature limit for the time being. Even so, the cumulative effect of these increases, together with the necessity for steam generators of greater capacity to supply the larger steam turbines being built, has been to modify considerably most of the previously accepted designs.

The greatest rate of heat transference can be obtained by the use of radiant heat, and water-tube boiler designers have therefore tended to arrange for the major part of the heat available to be absorbed by a comparatively small amount of " high-grade " heat-receiving surface massed about a high-temperature furnace zone. The area of " true " boiler convective heating surface required has in consequence diminished, and its main function has been delegated to the economiser heating surface, as explained below. The increase in the final steam temperature has necessitated the provision of larger superheaters arranged nearer to the furnace zone to obtain the requisite temperature gradient, and in some examples the superheaters are of the radiant-heat type, lining the furnace walls.

FEED WATER TEMPERATURE

Another problem that has been transferred to the boiler designer during the process of improving turbine efficiency, is that of high feed water temperatures. By extracting or " bleeding " steam from a turbine at one or more points during expansion of the steam, and using the bled steam to heat the boiler feed water, the thermal efficiency of the machine is raised, and usually the size of the low-pressure turbine can be reduced. In practice, the number of extraction heaters varies from one in a small steam turbine to five in a large one.

The result, in any arrangement, is that the feed water requires less heat from the flue gases to raise its temperature to that corresponding to ebullition. But, for the optimum boiler efficiency of large units installed in central power stations, it is necessary to discharge the flue gases at a temperature of about 300° F. so that a difficult situation arises when the turbine designer returns the condensate to the boiler house at a temperature of anything up to 400° F.

In modern designs this difficulty has been overcome by a reduction in the amount of true boiler convective surface provided; thus, in effect, moving the economiser nearer to the furnace. The economiser heating surface is then swept by gases at a temperature sufficiently high to provide the gradient necessary for heat transference to the water.

The gases leaving the economiser have still a large amount of usable heat, and this may be absorbed by an air heater installed in series with the economiser. The use of an air heater in these conditions, however, is limited by the method of burning the fuel. Mechanical stokers do not operate satisfactorily when the air temperature exceeds about 350° F., and the upper limit for primary air for pulverised fuel or oil burning is in the region of 500° F.

Thus the adoption of regenerative feed water heating may at times force the boiler designer not only to re-arrange the disposition and purpose of the boiler heating surface, but also to adopt another method of burning the fuel.

FURNACE DESIGN

The increase in the evaporative capacity of individual boilers occasioned by the use of larger turbo-generators has reacted on the dimensions of water-tube boilers, particularly in the furnace. The earlier boilers were constructed

with their heating surface arranged at only a short distance from the fire, and any attempt to raise the heat input was limited by smoke and similar troubles. In Chapter VIII it was shown that the rate of radiant heat transmission in a furnace is directly related to the fourth power of the absolute flame temperature, and that the percentage of the available radiant heat that is absorbed by the boiler heating surface decreases with the weight of air supplied to the combustion chamber per lb. of fuel.

Thus the aim of the designer is to provide as high a flame temperature as possible by restricting to as near the theoretical minimum as possible the amount of air supplied for combustion. It was found that higher flame temperatures and better combustion resulted from an increase in the height of the furnace, but as the heating surface was progressively withdrawn and furnace temperatures rose, trouble was experienced due to the relatively rapid failure of the refractory lining at the higher temperatures.

This aspect of the problem was partly obscured for a time by the construction of larger furnaces, by the use of air-cooled walls and by modifications of the furnace flame, but it re-appeared as furnaces were again forced to the limit of their heat-release capacity.

ASH AND SLAGGING DIFFICULTIES

Higher furnace temperatures also introduced ash and slagging difficulties, as in many plants the ash in the coal was heated to a temperature above its fusion point. In consequence, the molten ash was deposited on the furnace walls, and either fluxed with the refractory or " froze " on it setting up a form of rapid surface disintegration. Thus, although a furnace lining might be capable of standing up to a furnace temperature of 3,000° F. (1,650° C.), the use of coal having an ash fusion point of, say, 2,000° F. (1,100° C.) would limit the heat input to the rate corresponding to the lower temperature in order that heavy brickwork maintenance costs might be avoided. Similar troubles were experienced in the first rows of boiler tubes due to slag deposits causing choking of the gas passages and a reduction in the rate of heat transmission.

The furnace temperature can be diminished by increasing the amount of excess combustion air, or by providing a larger area of heat-absorbing surface in the furnace. The first method can be applied to practically all boilers, but it gives a lower CO_2 percentage in the products of combustion and a reduction in combustion efficiency.

The second method must be used with discretion, as with low-volatile and similar fuels undue cooling of the furnace may render it difficult to maintain ignition. Water-cooled furnace walls of the bare-tube or block-covered types have therefore been widely adopted in suitable conditions. Existing water-tube boilers will in consequence show improved performance when provided with water walls, but the effect of these on furnace efficiency and the utilisation of the available heat input must not be overlooked.

At the present stage of water-tube boiler development the position is that the necessary conditions of turbulence, temperature and time must be provided in the furnace to obtain the best all-round results, but the methods of meeting these conditions vary according to the class of coal burned and the type of boiler employed.

WATER CIRCULATION

The addition of water walls and the increases in steam pressure and feed water temperature described above have introduced new factors in regard to water circulation throughout the boiler. A free natural circulation of water presupposes a reasonable difference in temperature between one part of the

circulatory system and another and boiler engineers have had to take care of such physical characteristics when designing the various sections of a boiler.

Special consideration needs to be given to the separation of the steam from the water in regions subjected to high rates of heat transmission. Designers have therefore given considerable attention to the provision of adequate downcomer (or feed) area and the correct distribution of water to the various heating surfaces. These design problems have been solved to such an extent that natural circulation boilers in commercial undertakings have been operating satisfactorily for extended periods at pressures up to 2,500 lb. per square inch.

With forced circulation, as in the type of boiler illustrated diagrammatically in Fig. 109, there is a controlled circulation of water under all conditions of operation, and the heating surface can be arranged to obtain the maximum heat absorption. Moreover, the diameter of the water tubes can be made smaller and the design itself can be conveniently adapted to restricted space conditions at site. There may also be some advantage in reduced weight of the boiler.

A number of these boilers are in service in this country, but some engineers consider that they introduce complications in the form of continuously-running circulation pumps and difficulties in the removal of scale from the interior of the water-tubes.

Although much of the foregoing discussion applies to water-tube boilers at high evaporative capacity, the experience gained in extensive development work has been used in the design of the relatively smaller boilers installed in numerous factories.

Advantage has been taken of the increased knowledge of heat transmission to produce water-tube boilers capable of working efficiently within the Economic boiler range of evaporation. Some very compact steam generators have thus been made available for industrial service. In all installations the feed water supply must be reasonably pure.

Again, existing water-tube boilers have been modified to bring them into conformity with modern ideas on furnace design, and stoker and pulverised fuel burner manufacturers have collaborated with boiler designers to produce efficient fuel-burning appliances suitable for the various heat inputs required.

Progress has been facilitated by the greater use of boiler-house instruments capable of continuously checking and recording combustion efficiency and which give immediate warning of any variation from normal performance.

However, in order to secure the full advantages from the installation of more efficient boiler units and also from provision of a full range of boiler house instruments, it is essential that skilled operating engineers be placed in charge of the plant.

CHAPTER XIV

STEAM BOILER AUXILIARY PLANT

Boiler mountings and fittings—Soot removal—Superheaters—Economisers—Feed water heaters—Air preheaters—Auxiliary power and steam utilisation—Boiler feed pumps—Pumping hot water—Maintenance of feed pumps—Draught production by fans.

THE performance of a steam boiler, as has been indicated in Chapter XIII, depends upon the correct disposition of its heating surface, and for the highest efficiencies it is necessary to superimpose on a sound design additional heat-absorbing appliances, such as superheaters, economisers and air preheaters. Certain mountings, fittings and instruments must be attached to the boiler installation. Similarly, the selection of feed pumps, fans and other auxiliaries has an important effect on the efficiency of the boiler plant as a unit.

BOILER MOUNTINGS AND FITTINGS

Attachments to boilers are divided fairly rigidly into two classes : mountings and fittings. Instruments are a separate item and seldom actually attached to the boiler fabric.

Under the heading of mountings are comprised safety valves, high and low water alarms, main stop valves, feed and check valves, circulators, blow-down valves, water gauge frames and valves, pressure gauges, anti-priming pipes and scum valves.

Under fittings would be included damper frames and damper gear, injectors, soot blowers, furnaces and stokers, superheaters or preheaters, feed regulators and thermostatic or pressurestat controllers.

Generally speaking the term " mounting " implies that the equipment is mounted on a pad or stool riveted on to the fabric of the boiler, as distinct from a " fitting " which may or may not be attached to the boiler, but for which there is neither stool nor pad.

In both categories there are some mountings and fittings which are essential to operation and others which may not be essential, but which in themselves, in one form or another, fulfil a perhaps necessary service to the plant to which they are attached.

Safety Valves. Safety valves are essential on all steam boilers, and are generally of three main types : dead weight, lever or steel yard and spring loaded, each type being made in a variety of patterns with or without means for locking or adjustment. Whichever type is selected the design should be such that unauthorised interference with the setting of the valve is impossible.

Every boiler must be fitted with two independent safety valves each capable alone of discharging the maximum amount of steam that the boiler can produce ; so that if one valve sticks for any reason there is still one left functioning and the danger from abnormal pressure is avoided.

Unless it is imperative to do so, discharge through a safety valve should never be piped away ; it is far better that the safety valve should only blow on very rare occasions. Every time the valve blows, fuel is wasted at a prodigious rate.

High and Low Water Alarms. A very common valve of this type comprises a balanced float gear which is usually built into the lever type of safety valve. The combination thus provides a safety valve as demanded by law, together with high and low water protection. The sound made by the escape of steam is different from that made by water, so that the operator can detect whether it is the high or low water alarm which is functioning or whether it is the safety valve which is blowing off.

Boiler Feed Valves. The boiler feed or " clack " valve probably does more work in continuous movement than any other piece of equipment on the boiler, particularly if the feed pump is of the reciprocating type. The correct position for the feed valve pad is about 3 inches below the water level, and it should be set as close to the boiler shell as possible so that there will be no necessity for a bend in the internal feed pipe to carry the feed into the internal shell in a downward direction. Internal feed pipes in shell boilers should not be extended beyond the middle of Lancashire or Cornish boilers, and in the Economic boiler the pipes should never reach within 6 feet of the rear tube, especially if the feed is relatively cool.

Circulators. There are two major types of circulator, (1) thermodynamic, depending on convection currents, and (2) the static which makes use of energy otherwise dissipated in priming. All shell boilers suffer from faulty circulation owing to the fact that heat is always applied at a point above the lowest water level, so that circulators are added to overcome this difficulty in some measure.

Static circulators are characterised by a hood or reaction plate which is always placed in the position where the maximum priming occurs. Circulators require that the water level should be rigidly maintained, otherwise, if over-feeding takes place, violent priming in the boiler might result.

Blow-down Valves. Blow-down valves are an important fitting to any boiler and should be essentially of first class structure, inasmuch as the valve is often in a position where it is not continuously under the eye of the operator ; leakage through this valve may result in a heavy waste of fuel.

SOOT REMOVAL EQUIPMENT

A factor having considerable influence on the efficiency of a steam boiler during operation is the thickness of deposits which form on the fire side of its heating surfaces. When boilers are worked at comparatively low rates the deposits may amount to no more than a film of relatively easily removed soot, but in larger water-tube boilers in which furnace temperatures are high, the deposits may assume the properties of a hard cement-like scale. This operating problem has given great trouble within recent years and is not yet fully solved. These notes are concerned more with the method of removing the deposits than with the mechanism of their formation.

Obviously, one way of removing these deposits is to shut down the boiler and scrape and brush the heating surfaces. This is not always convenient, particularly during emergency conditions, and resort is therefore made to the various types of blowing jets that have been developed for this purpose over the past twenty-five years.

In the vertical boiler with water-tubes across the firebox, a nozzle can be arranged below or between the tube nests to play on the spaces between the tubes and blow the accumulated soot up the chimney.

In Lancashire and similar boilers, steam jets can be arranged in the external flues, the soot and dust thus removed being deposited in a special dust-collecting chamber. Such jets work in a comparatively low-temperature atmosphere, but in Economic boilers, where the blowers project into the combustion chamber, to clean the smoke-tubes blowers of the withdrawable or retractable type are necessary if a reasonable working life is to be obtained from them. The blowers are then designed so that a single handwheel causes the jets to move into the combustion chamber and deliver steam at the same time, while oscillating over a pre-determined arc to cover all the tubes.

Blowers of this type can be adapted for use in connection with the furnace and superheater tubes of water-tube boilers, and for other positions where the nozzle can be arranged some 2 feet 6 inches to 5 feet from the area of impact. Where situated in regions of high gas temperature, the blower parts must be

made of suitable materials and must be designed to retract well out of the influence of the hot gases when not in use. A fairly high steam pressure—200–500 lb. per square inch—is required for the successful operation of this blower.

In regions of lower temperature and when nests of tubes have to be cleaned, a form of multi-nozzle blower may be installed. The nozzles or jets are arranged to blow into the spaces between the tubes, but care must be taken to prevent impingement of the steam-blast on to the thin water-tube walls. These multi-nozzle blowers may be fixed a few inches or up to 2 or 3 feet from the tubes to be cleaned, and the steam pressure required varies from about 100–350 lb. per square inch, according to the distance from the tube nest.

Automatic methods of soot blower operation are now available in which the sequence and duration of action of soot blowers arranged throughout the boiler structure can be determined by push-button remote control, and in large modern water-tube units the soot-blowing equipment is considered simultaneously with the general design.

HEAT LANCING

In circumstances in which deposits build up on the fire side of water-tube boiler surfaces quicker than they can be removed by the normal design of soot blower, it may be necessary to use a portable steam " gun," operated by hand.

Recently water sprays have been used working under a pressure of 50 to 100 lb. per square inch. In this system the water is sprayed on to the boiler tubes, and it is stated that after only a few minutes the deposit is loosened. The water is then shut off for an interval until the deposits heat up, and a second spraying usually causes them to break away. The action is somewhat similar to the heating and rapid cooling of copper evaporator coils adopted by the marine engineer to remove scale from the coils without having to open up the evaporator.

Compressed air has not been used a great deal in this country for soot blowing purposes, but it is significant that a recent (1942) American high-pressure power station has adopted compressed-air soot blowing at a pressure of 250 lb. per square inch. The designers of this station consider that the compressed-air system is just as economical as steam blowers, and is more adaptable for combining with a water system, should water jets become necessary.

SUPERHEATERS

The use of superheated steam has been considered in detail in Chapter VII, where its value for power generation was indicated.

Most superheaters are installed within the boiler setting and receive their heat from the same flue gases and same furnace as the boiler. These are termed " integral " superheaters, and they are generally of the convection type, absorbing heat from the gases of combustion which sweep over them. In some instances externally or separately fired superheaters have to be provided, but in general steam-raising practice these are comparatively rare and for the present purpose can be disregarded.

Gas Temperatures and Velocities. For effective heat transmission in a superheater the temperature difference between the gas and steam sides of the elements must be about 300° F. (149° C.). In the convection type of superheater the resistances to heat flow are the gas film, the metal wall of the tube and the steam film. The most important of these is the gas film, the effect of the other two becoming relatively insignificant as the rate increases. The final steam temperature is a function of the gas temperature and of the mass velocity of the gases (i.e. the weight of gases in lb. per second per square foot of gas passage area), a small increase in the mass velocity having a propor-

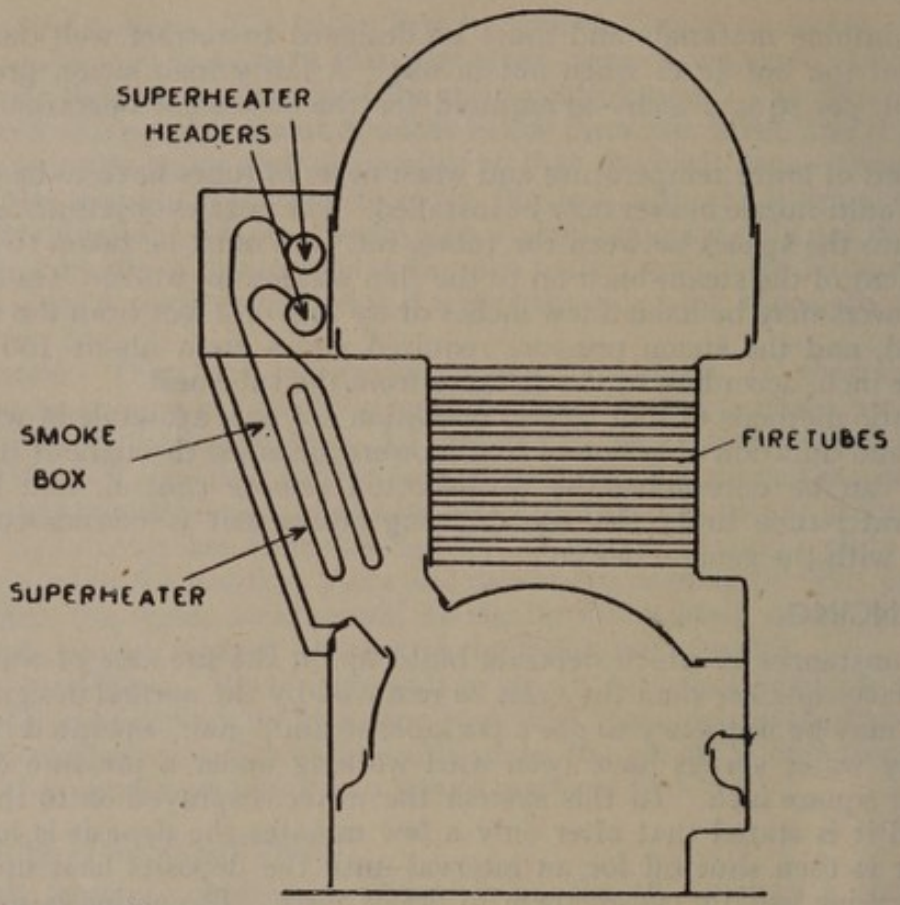

FIG. 110. Superheater applied to a vertical boiler.

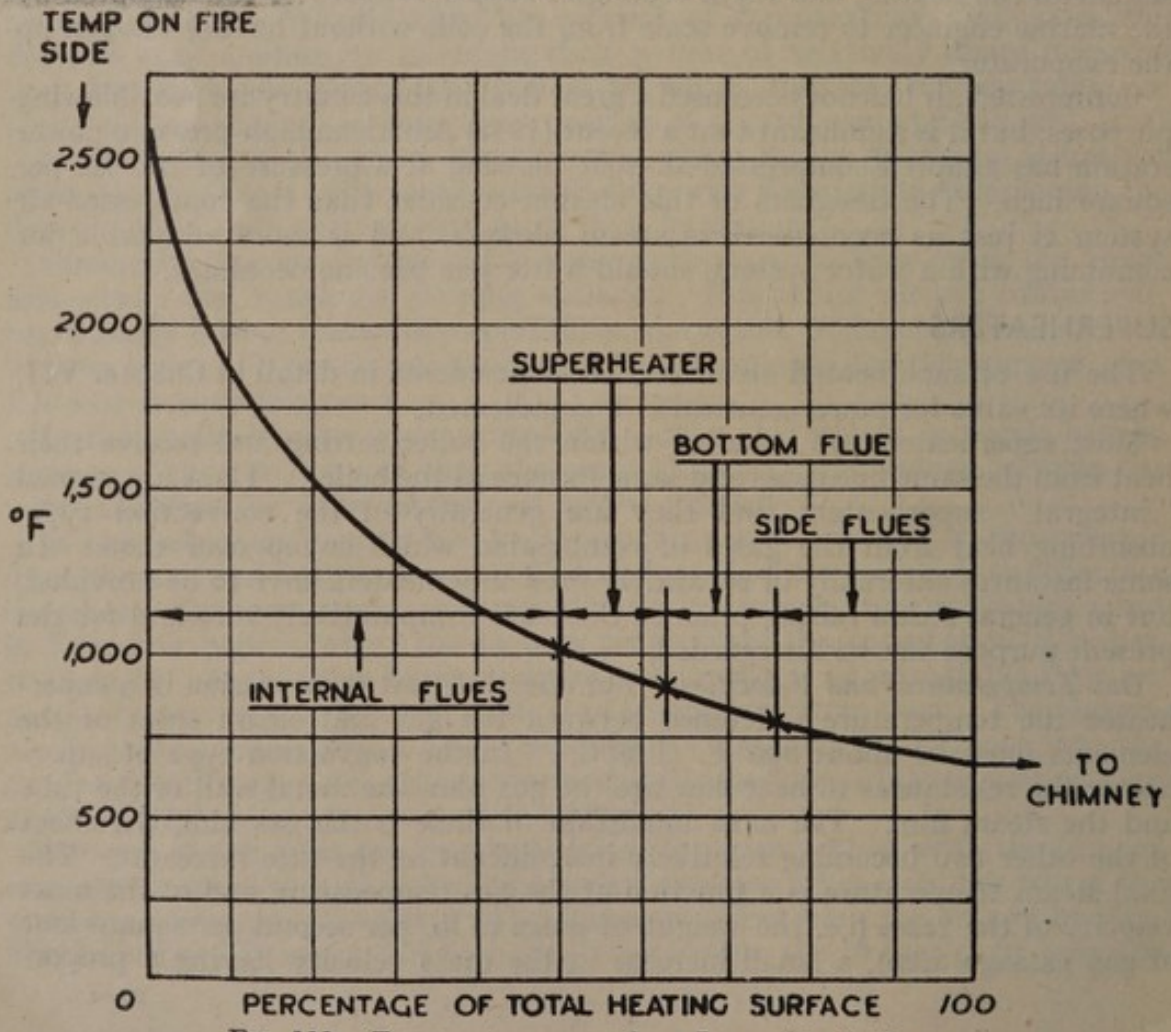

FIG. 111. Temperature curve for a Lancashire boiler.

tionately greater effect on heat transmission than a small increase in gas temperature.

On the steam side, the primary requirement is that all the superheater tubes should receive a uniform flow of steam, and the steam velocity must be high enough to prevent overheating. There is a critical velocity at which heat transmission is at a maximum, according to the tube diameter and length, but too high a velocity may result in an unduly large pressure drop across the superheater. A pressure drop of from 2 to 5 per cent. is usually considered to be permissible, although superheater designers frequently assert that this is unnecessarily low in view of subsequent steam pipe-line losses.

Superheater Types. The construction, size and position of a superheater relative to the boiler surface which it amplifies depend upon the steam tem-

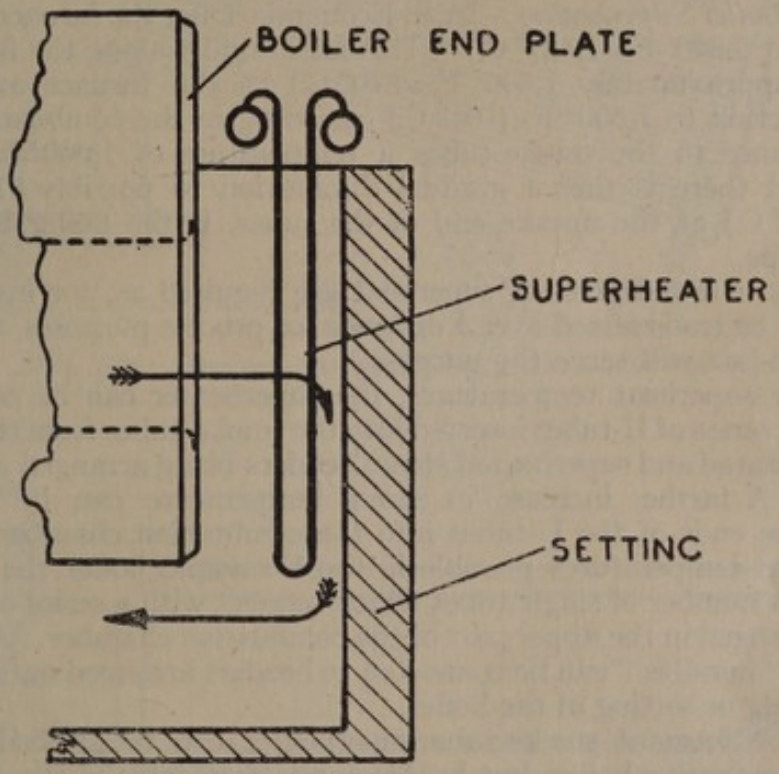

FIG. 112. Superheater arranged for a Lancashire boiler.

perature desired, and also, to a certain extent, upon the type of boiler to which the superheater is applied. In a vertical cross-tube boiler, for instance, the space available for a superheater is extremely limited, and the only practical position is in the uptake. The diameter of the uptake restricts the area of superheater surface that it is possible to provide, so that when fitted in this type of boiler it forms, in effect, an extension of the steam pipe placed in the path of the products of combustion. In a vertical boiler having vertical smoke-tubes, the superheater can be arranged over the top tube-plate in the form of a continuous coil, and the provision of a greater area of superheater surface is possible.

Provision for still greater area can be made when a boiler of the type shown in Fig. 110 is under consideration, and in this example the superheater may be constructed of a number of " hairpin " bends located in the combustion chamber space. This design results in much higher superheat temperatures than those mentioned above.

In a Lancashire boiler the temperature on the fire side of the boiler falls progressively from the furnace to the main chimney flue opening. An approximate temperature curve for a 9 feet 6 inches diameter by 30 feet long Lancashire

boiler is shown in Fig. 111, and it will be seen that the greatest temperature drop occurs in the internal flues.

When the maximum amount of superheat possible with a Lancashire boiler is required, it is necessary to arrange the superheater elements inside the internal flues, where they are subject to gas temperatures of the order of 1,300° F. (704° C.), as well as to a certain amount of radiant heat.

For more normal requirements up to about 250° F. (120° C.) of superheat, the superheater can be arranged at the mouths of the internal flues, as shown in Fig. 112, the gas temperature in this region being about 1,000° F. (538° C.). From the curve it will be appreciated that the remainder of the available temperature drop is of comparatively little use for the higher range of superheat temperatures, since on passing through the bottom external flue the temperature falls to about 750° F. (400° C.).

Economic Boiler Superheaters. In an Economic boiler the furnace temperature may be about 2,600° F. (1,427° C.). This falls rapidly along the furnace length until it is approximately 1,600° F. (870° C.) at the furnace exit mouth, a further reduction to 1,300° F. (704° C.) occurring in the combustion chamber. At the entrance to the smoke-tubes a temperature of 1,000° F. (538° C.) is common, and there is then a gradual diminution to possibly 575° or 550° F. (302° or 288° C.) at the uptake end of the tubes, if the boiler is of the self-contained type.

When only a few degrees of superheat are required as, for instance, where steam has to be transmitted over a distance for process purposes, a superheater in the smoke-box will serve the purpose.

For higher superheat temperatures, the superheater can be constructed in the form of a series of U-tubes inserted into the smoke-tubes from the smoke-box end, the saturated and superheated steam headers being arranged in the smoke-box space. A further increase in steam temperature can be obtained by extending the ends of the U-tubes into the combustion chamber, but for the highest steam temperatures possible in the Economic boiler the U-tubes are replaced by a number of single tubes which connect with a series of superheater elements arranged in the upper part of the combustion chamber. Alternatively, the element " bundles " can be connected to headers arranged outside the back-end and casing or setting of the boiler.

The U-tube form of smoke-tube superheater can be applied also to the stationary locomotive boiler, but in this application some of the upper rows of smoke-tubes are replaced by tubes of larger bore capable of receiving the elements, the header or headers being placed in the smoke-box. When only a moderate increase in steam temperature is desired, this space can be utilised to house superheater surface consisting of continuous tubing of suitable size.

Water-tube Boiler Superheaters. In water-tube boilers, steam temperatures up to 1,000° F. (538° C.) have been maintained in service, but in normal installations the steam temperature is restricted to about 850° F. (454° C.) to allow of the maximum use of ordinary commercial steel tubes. The degree of superheat obtainable depends upon the position of the superheater in the gas flow and upon the area of heat-absorbing surface interposed between the superheater and furnace.

There are four principal types of superheater employed with water-tube boilers: the interpass, the interdeck, the intertube and the radiant heat superheater. The interpass design is illustrated in Fig. 113, and receives most of its heat by convection. It is generally used in connection with a refractory-lined furnace when a moderate steam temperature is required, and obviously the superheat obtainable can be varied in the design stages by suitably moving the first-pass baffle to expose more or less water-tube area to the flow of gases.

A defect of the convection superheater just illustrated is that the steam

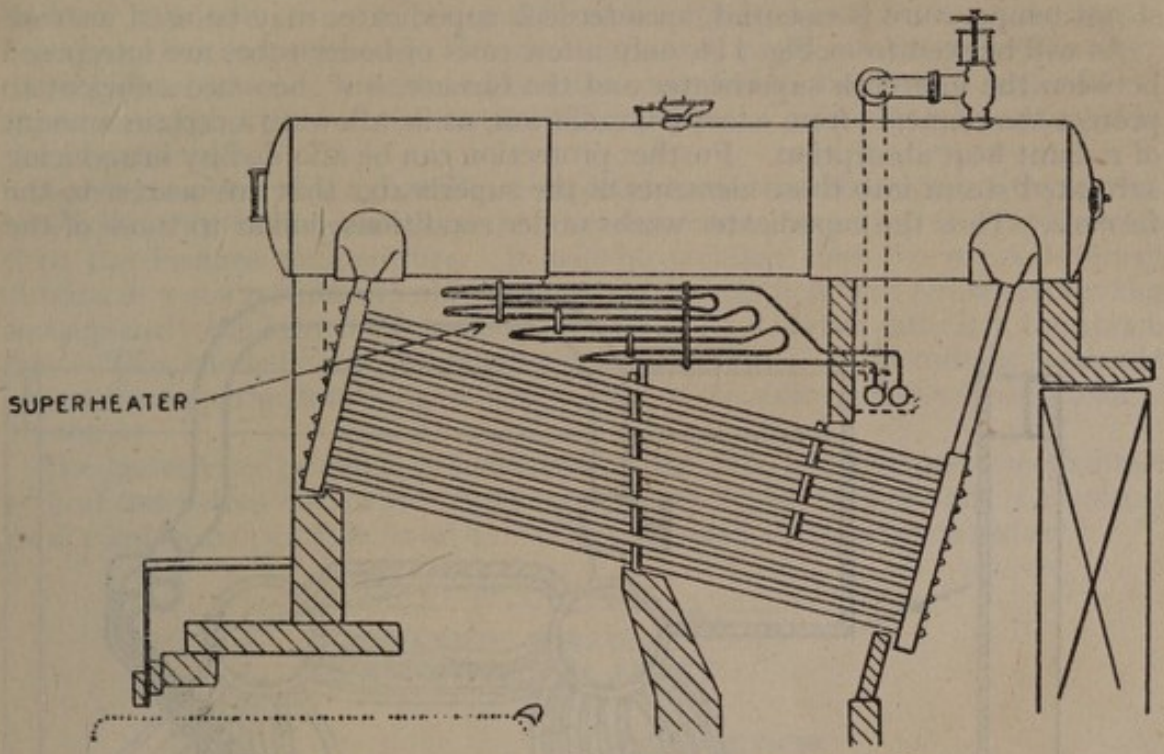

FIG. 113. Interpass superheater in a water-tube boiler.

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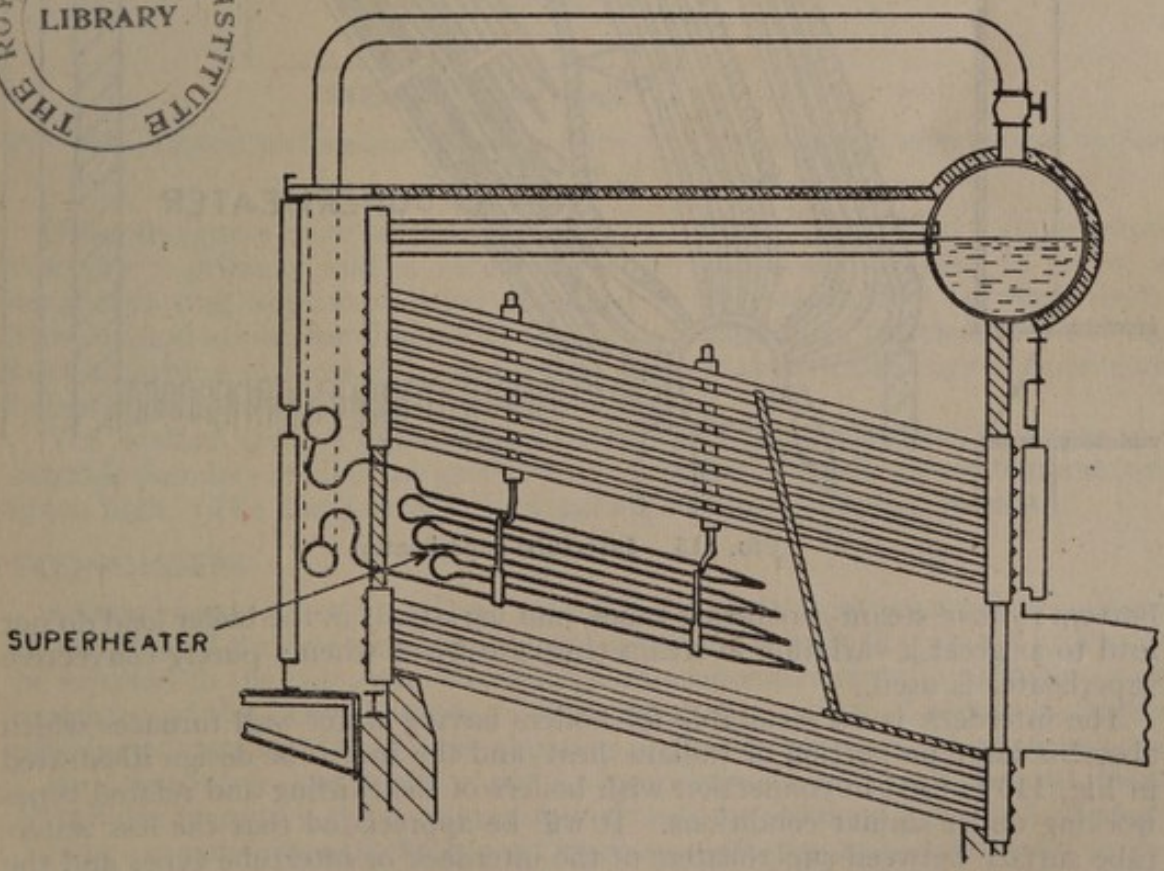

FIG. 114. Interdeck superheater.

temperature rises with an increase in boiler load, so that when a fairly steady steam temperature is essential, an interdeck superheater may be used instead.

As will be seen from Fig. 114, only a few rows of boiler tubes are interposed between the interdeck superheater and the furnace, but these are sufficient to protect the elements from excessive radiation, while allowing a certain amount of radiant heat absorption. Further protection can be afforded by introducing saturated steam into those elements of the superheater that are nearest to the furnace. Thus the superheater works under conditions similar to those of the

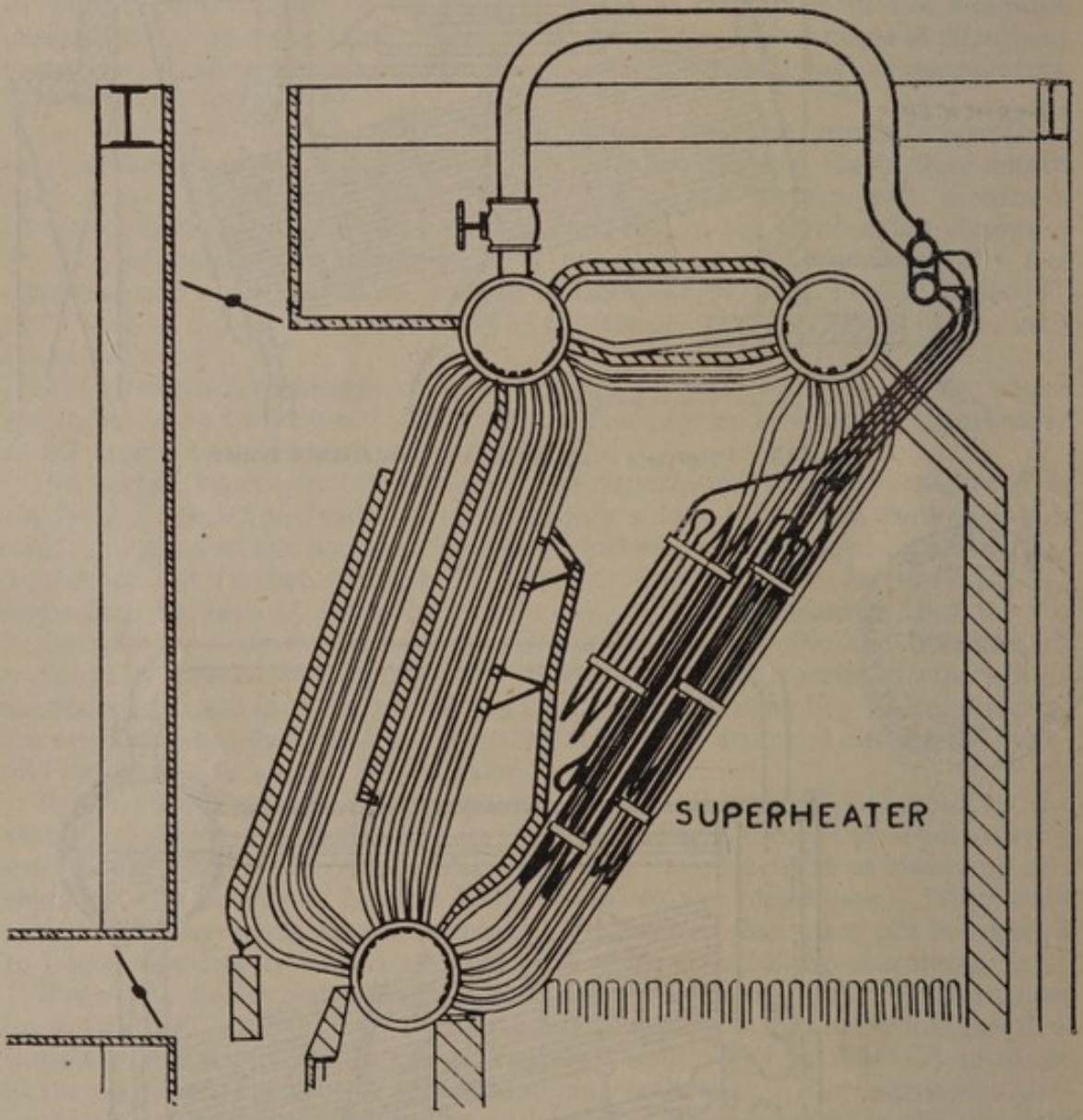

FIG. 115. Intertube superheater.

bottom rows of steam-generating tubes, and variations in the boiler load do not lead to as great a variation in steam temperature as when a purely convective superheater is used.

The interdeck is very suitable for boilers having water-wall furnaces which absorb a large proportion of radiant heat, and the intertube design illustrated in Fig. 115 is used in connection with boilers of the Stirling and related types working under similar conditions. It will be appreciated that the less water-tube surface between superheaters of the interdeck or intertube types and the furnace, the less will be the area of superheater surface required, and the higher

will be the final steam temperature. The more severe the fire side conditions, however, the better must be the quality of the material used for the elements.

Radiant heat Superheaters. These remarks are particularly applicable when the water-tube screen is removed and the superheater becomes a radiant heat type. In applications of this nature the superheater tubes may be arranged in the furnace walls.

A characteristic of a radiant heat superheater is that the superheat temperature falls as the boiler load increases, the steam flow increasing at a greater rate than the furnace temperature. It will be recalled that the opposite effect obtains in a convection superheater, due to the gas flow and temperature, and consequently the heat transmission, increasing at a higher rate than the steam flow. This has led some designers to use a combination of radiant heat and convection superheaters in series with a view to obtaining a flatter characteristic curve.

The underlying principle is indicated in Fig. 116, which shows approximate typical individual curves for these superheater types, and the curve resulting from combined operation in series. The combined curve is much flatter.

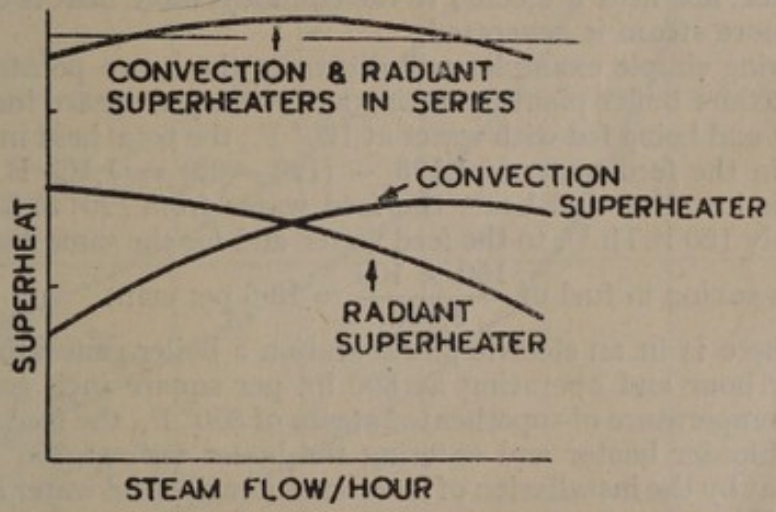

FIG. 116. Typical performance curves for convection and radiation superheaters, and for a combination of the two.

Other designers have preferred to adopt a form of " controlled " superheater whereby a primary and a secondary superheater are coupled in series, a desuperheating appliance being arranged in the steam flow between them. This method allows for changes in the rate of steam flow, the cleanliness of the heat-absorbing surfaces, etc., and results in a constant final steam temperature independent of all other operational variables.

Yet another method of superheat control is to provide by-pass passages and suitable dampers so that the gases can be diverted when the steam temperature is too high. The dampers may be manually or automatically operated.

ECONOMISERS

An economiser, at one time, was regarded as a heat-recovery appliance placed in the path of the flue gases to abstract some of the heat that would otherwise be rejected to the chimney. Nowadays an economiser may be regarded as an extension of the boiler heating surface added as a means of obtaining higher efficiency at less cost than if boiler heating surface were provided for the purpose.

An economiser, therefore, must be proportioned with the general plant conditions in mind if the maximum benefit is to be obtained from the minimum capital expenditure, and the benefit increases with the cost of fuel used in the boiler. It is estimated that about 1 per cent. of fuel costs can be saved for every

10° F. increase in the temperature of the boiler feed water, so that where very inefficient boiler plant is being considered, savings up to 15 and 20 per cent. can be effected.

When more heat is available than can be used in increasing the sensible heat of the feed water, it may be necessary to employ a "steaming" economiser or reject the surplus heat to the chimney, if additional air heating cannot be adopted. In the average economiser, however, the feed water is not heated higher than to within 50° to 30° F. of the temperature corresponding to the boiling point of the boiler water, thus preventing any possibility of steam generation and subsequent water-hammer risks. A useful, if unconventional use of the economiser is to provide hot water for the factory for central heating, or for offices.

Economics of Economisers. The economic aspect of the installation can be regarded from two points of view. It can be considered that for a fixed evaporation the economiser provides a portion of the total heat added to the feed water to convert it into steam, and, since this heat is obtained for nothing, it effects an appreciable saving in fuel. Alternatively, for a fixed weight of fuel burnt per hour, less heat is ejected to the chimney, more heat is converted into steam and more steam is generated.

The following simple examples will illustrate these two points of view.

In a Lancashire boiler plant operating at 120 lb. per square inch gauge with no superheat and being fed with water at 120° F., the total heat in 1 lb. of steam above that in the feed water is 1,193 − (120 − 32) = 1,105 B.Th.U. If an economiser is installed and heats the feed water from 120° to 270° F. it adds approximately 150 B.Th.U. to the feed water and for the same hourly evaporation effects a saving in fuel of $\frac{150 \times 100}{1{,}105} = 13{\cdot}6$ per cent.

Suppose there is in an electric power station a boiler generating 100,000 lb. of steam per hour and operating at 600 lb. per square inch gauge pressure, with a final temperature of superheated steam of 850° F., the feed water leaving the turbine bleeder heater and entering the boiler unit at 300° F. Suppose, moreover, that by the installation of an economiser the feed water is heated from 300° to 450° F.

	B.Th.U./lb.	
Total heat of steam at 600 lb. per square inch gauge superheated to 850° F.		1,435
	B.Th.U./lb.	
Heat in feed water at 450° F.	430	
,, ,, ,, 300° F.	270	270
Heat added in economiser	160	
Heat in steam at 600 lb. and 850° F. with feed water at 300° F.		1,165

Without an economiser all this heat must be added in the boiler and superheater. With an economiser, 160 B.Th.U. of this heat are added in the economiser, leaving 1,165 − 160, or 1,005 B.Th.U. to be added in the boiler and superheater.

Therefore the evaporation will be 100,000 × 1,165/1,005 = 115,900 lb. per hour.

It is thus seen that for the same size coal-handling plant, mechanical stoker, combustion chamber, boiler, superheater, forced and induced draught fans, grit arrestor, gas-washing plant, chimney, ash-handing plant and boiler-house

building, the total evaporation has been raised by 15·9 per cent. This means that practically for the same capital expenditure 15·9 per cent. more steam is generated.

The relative size of economisers designed to reduce the flue gas temperature to 280° F. is shown in Fig. 117 (P. H. N. Ulander, *J. Inst. Fuel*, I., 188).

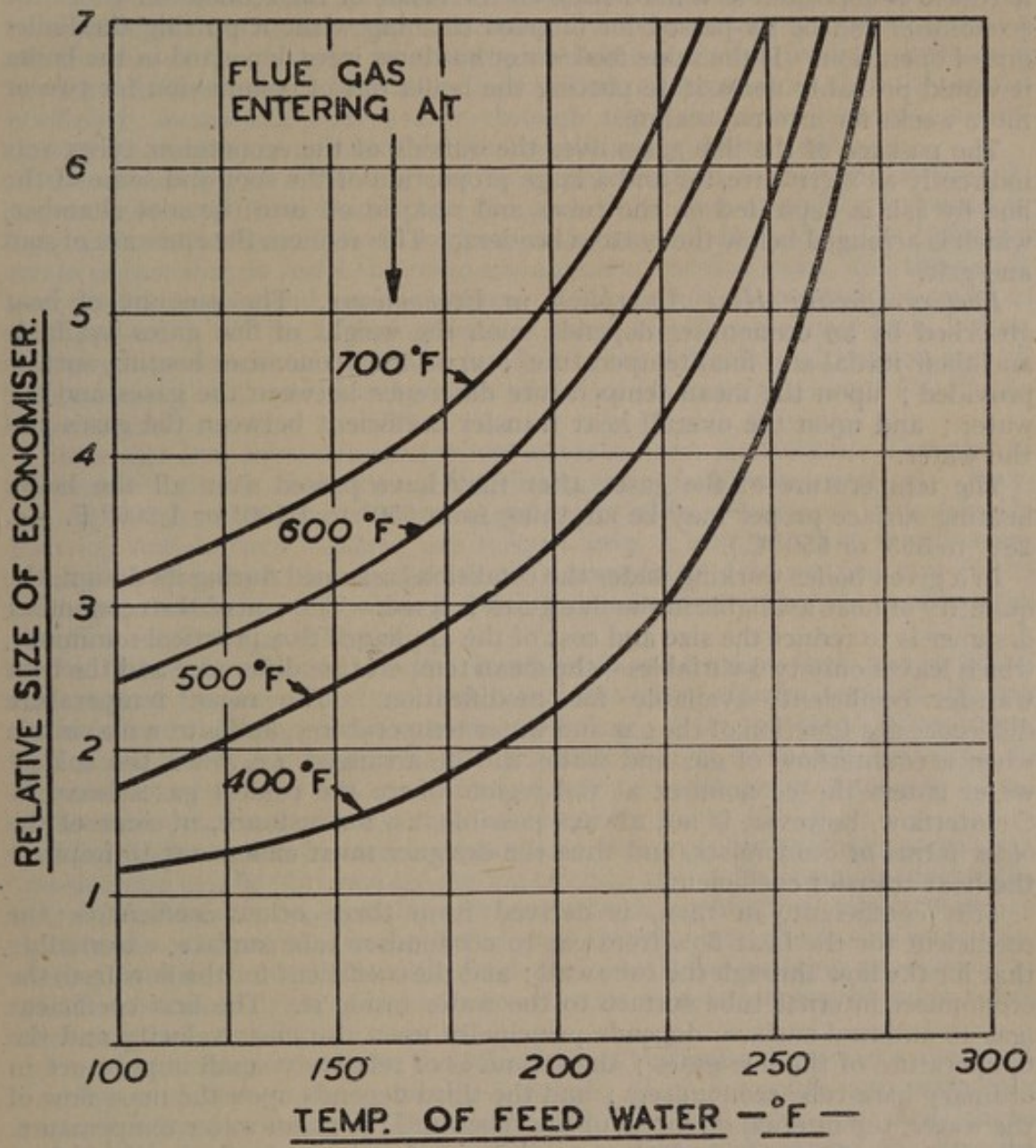

FIG. 117. Relative size of economiser to reduce flue gas to 280° F.
(P. H. N. Ulander, *J. Inst. Fuel*, I, 188.)

There are several indirect advantages obtained by the installation of economisers, the three most important of which are as follows :—

The feeding of the boiler with water at a temperature near the boiling point reduces the temperature differences in the boiler, prevents the formation of stagnant pockets of cold water and thus reduces greatly the temperature strains created in the pressure parts of the boiler and promotes better internal circulation.

Where the feed water is not as pure as it should be, the temporary hardness is deposited on the inside of the economiser tubes, and while this necessitates

internal cleaning of the economiser, the evil is usually not as great as internal cleaning of the boiler. The temperatures experienced in the economiser not being as high as those in the boiler, the salts are sometimes deposited as a soft sludge and can be blown down through the economiser blow-down valve instead of forming a hard scale on the inside of the boiler. Even where the unwanted hardness is deposited as a hard scale on the inside of the economiser tubes the economiser can be by-passed for internal cleaning without putting the boiler out of operation. If the same feed water hardness were deposited in the boiler it would probably necessitate putting the boiler out of commission for two or more weeks for internal scaling.

The passage of the flue gases over the outside of the economiser tubes acts indirectly as a grit arrestor and a large proportion of the soot and some of the fine fly ash is deposited on the tubes and scraped off into the soot chamber, which is arranged below the bottom headers. This reduces the emission of soot and grit.

Factors affecting Heat Absorption in Economisers. The amount of heat absorbed by an economiser depends upon the weight of flue gases available and their initial and final temperatures; upon the economiser heating surface provided; upon the mean temperature difference between the gases and the water; and upon the overall heat transfer coefficient between the gases and the water.

The temperature of the gases after they have passed over all the boiler heating surface proper may be anything from 550° to 1,100° or 1,200° F. (i.e. 288° to 593° or 650° C.).

In a given boiler working under the conditions assumed during its design, the quantity of heat available in the flue gases is fixed. The aim of the economiser designer is to reduce the size and cost of the appliance to a practical minimum, which leaves only two variables—the mean temperature difference and the heat transfer coefficient—available for modification. The mean temperature difference is a function of the gas and water temperatures, and is at a maximum when a counterflow of gas and water can be arranged, i.e. when the coldest water enters the economiser at the region where the coldest gases leave it. Counterflow, however, is not always possible, as, for instance, in some of the older forms of economiser, and thus the designer must endeavour to improve the heat transfer coefficient.

This coefficient, in turn, is derived from three other coefficients: the coefficient for the heat flow from gas to economiser tube surface, externally; that for the flow through the tube wall; and the coefficient for the flow from the economiser internal tube surface to the water inside it. The first coefficient (gas to external surface) depends principally upon the mass velocity and the temperature of the flue gases; the second is of relatively small importance in ordinary bare-tube economisers; and the third depends upon the mass flow of the water, the internal diameter of the tubes and the mean water temperature. The water coefficient is, in fact, so much larger in relation to the gas coefficient that it is usually sufficient to select a water velocity suitable for a given water pressure drop through the economiser and ensure that turbulent flow will be maintained at the lowest expected boiler rating.

Extended Pipe Surface. Designers have therefore been compelled to improve economiser performance by an increase in the gas coefficient, and this can be readily augmented by raising the gas velocity. The usefulness of this method, however, is restricted by the permissible cost of the extra fan power required, although it can be extended to the limit by the addition of extra heat-absorbing surface, in the form of gills, etc., to the conventional form of bare economiser tube usually fitted.

The bare tube has disadvantages in that whether it is arranged in plain or

staggered formation there are liable to be " dead " spaces behind the tube in the direction of the gas flow, and these spaces are untouched by the gases. Moreover, a certain amount of stratification is likely to occur (cf. Chapter IX), as a result of which layers of gas can pass right through the economiser without impinging upon more than a small proportion of the tubular heating surface.

The addition of extended surface to the bare tubes thus offers more surface for contact and increases the " scrubbing " action, but too much may defeat the purpose of the designer by causing an undue reduction in the temperature head available at the tube wall. It is in such circumstances that the second coefficient mentioned above—flow through the tube wall—assumes greater importance in the computation of the overall heat transfer coefficient.

Economiser Types. Turning now to a consideration of economiser types in general use, it will be found that the plain-tube type is widely favoured in Lancashire boiler installations working under natural draught. The tubes are made of cast iron to resist the corrosive action of the flue gases, and their ends are pressed into top and bottom headers. The method of construction restricts the application of this type to pressures below 250 lb. per square inch, but an improved construction in which a number of the tubes in each section is positively secured to the headers by bolts allows of its use up to about 550 lb. per square inch.

Plain cast iron tubes are used where space is not a restriction. The nest of economiser tubes is then arranged with the axes of the tubes vertical. Four, six, eight, ten or twelve tubes, machined at the ends and pressed hydraulically into top and bottom headers, are formed into a section. The sections are assembled in groups, the number varying according to the evaporation of the boiler plant and the duty required from the economiser. The standard lengths of tubes are 9, 11½ and 13 feet.

The bare-tube design has the inestimable advantage that the external surfaces can be continuously and directly cleaned by circular scrapers moving up and down the tubes, but if external corrosion is to be prevented, the entering feed water must be *at all times* at a temperature of at least 110° F. (43° C.) to eliminate " sweating." If corrosion is found to occur under these conditions the water inlet temperature should be raised by 20° to 130° F.

Gilled Tubes. A reduction in economiser size together with increased heat transmission can be obtained by casting circular gills on to the bare-tube walls, but at the higher pressures the cast iron tube is unsuitable. Steel tubes are then used, and have cast iron gilled sleeves shrunk on to them, the inner tube serving to withstand the pressure and the outer tube resisting any corrosive attack. Suitable spigot joints between the sleeve ends protect the inner tubes from external corrosion.

The ideal arrangement is for the gases to pass vertically downwards over such a group and the water vertically upwards, thus obtaining approximately contraflow heat exchange. The number of tubes and rows and their length is determined mainly by the total heating surface required and the draught loss available. It is usual to connect the tubes so that the water flows through all the tubes of the bottom row in parallel and then returns through all the tubes of the second row; then back through the third row, and so on. Thus the flow through the tubes of any one row is in parallel and the flow through the rows is in series. However, this arrangement can be changed to meet special circumstances, e.g. where a very low hydraulic loss through the economiser is desired, in which case the water passes through two rows in parallel.

Trouble with soot deposits, however, has necessitated modifications of the circular type of gill, and there are now available several forms of rectangular gills with " streamlined " surfaces designed to reduce draught losses to the

minimum consistent with high heat transfer. It will be appreciated that tubes of the extended-surface type cannot be cleaned by mechanical scrapers while in service, and must, therefore, be equipped with steam or compressed-air soot blowers. The blowers are operated at regular intervals and collect the soot in places convenient for removal.

Where weight is a major consideration as in marine economisers, an alloy of aluminium is used for the construction of the gills. Since the coefficient of expansion of aluminium is greater than that of mild steel, each aluminium gill is kept in intimate contact with the steel tube by the insertion of a ring of special steel with a coefficient of expansion less than that of mild steel.

FEED WATER HEATERS

The installation of an economiser cannot, of course, be justified in every class of steam-generating plant, as, for instance, in a moderate-sized laundry having an Economic boiler for all purposes. The use of feed water at as high a temperature as possible is obviously desirable in such circumstances, and some means of obtaining it must be arranged.

In small factories which utilise steam for heating and similar purposes, every effort should be made to return the hot condensate to the feed tank (see Chapter XXIII). If the condensate temperature is 180° F., 1 gallon of condensate carries 1,480 B.Th.U. Thus if the fuel burned in the boiler has a calorific value of 13,000 B.Th.U. per lb. and the boiler efficiency is 65 per cent., every gallon of condensate returned to the feed tank represents a saving of approximately one-seventh of a lb. of coal based on feed water at 60° F. In addition, the water so returned is free from the scale-forming compounds found in the raw water, thus assisting in reducing boiler cleaning charges.

Hot feed water can be obtained by blowing live steam into the feed tank through one of the " silent " jets now available, but the real economy of this practice is somewhat doubtful. When a factory develops all or a part of its power by means of a steam engine, feed water heating can be effected by discharging sufficient of the exhaust steam into the tank. In a multi-expansion engine steam can be tapped off between the expansions, but in either arrangement provision should be made so that a vacuum cannot form in the steam pipe and cause feed water to be drawn into the engine cylinder.

If the engine exhaust contains oil the open exhaust method cannot be used, and an oil separator must be incorporated in the system. If adequate volumes of exhaust steam are available, it may be advisable to install a feed heater of the direct-contact type which not only heats, but also de-ærates the water. Heaters of this kind are arranged on the suction side of the feed pump, at the requisite height, and as they are open to the atmosphere the maximum feed temperature obtainable is limited to 212° F.

ENCLOSED HEATERS

Higher feed temperatures up to that of the boiler water are possible by using one or more heaters of the enclosed type. These invariably assume the form of a closed body which houses tubular heating surface, the heating steam flowing over the external surfaces of the tubes while the water is pumped through them.

For the lower pressure and temperature ranges the tubes may be secured in two tube plates, as in the normal surface condenser, but for the higher ranges the relative expansion of tubes and body prohibits this construction. Various types of " floating " headers, U-tubes, etc., have been evolved to meet the need, and the principal differences between makes of feed heaters are in that respect.

Considerable use of surface heaters has been made in connection with turbo-generator sets, the heating steam being " bled " from the machine after the

steam has done a certain amount of useful work. To obtain the theoretical optimum efficiency, an infinite number of heaters should be employed, but in practice the number has so far been confined to five or six. Steam which has passed the saturation point is generally used, and when more than one heater is justifiable the remaining adiabatic heat drop may be suitably subdivided from this point.

If a high pumping loss is to be avoided, the velocity of water through the

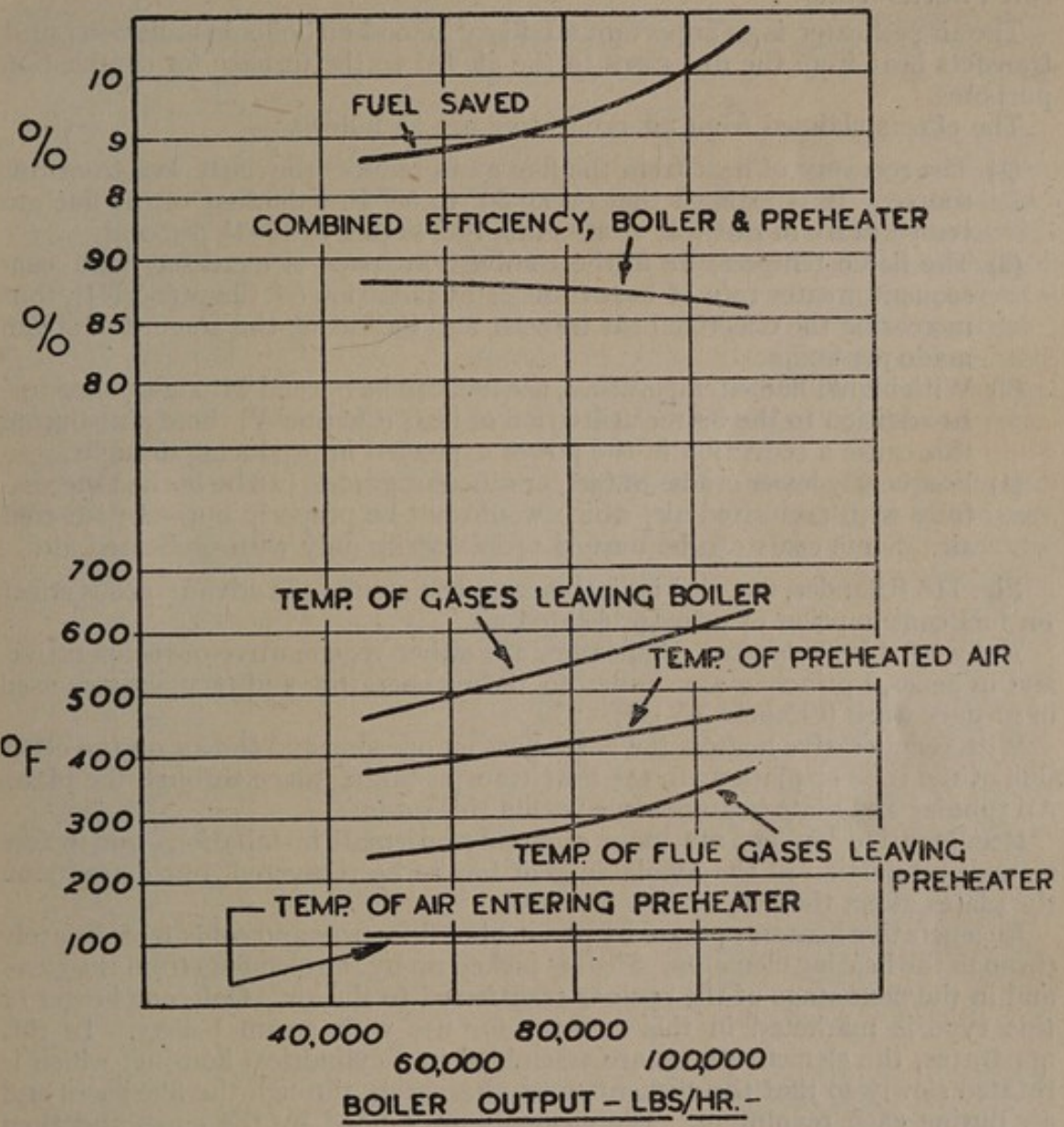

FIG. 118. Test results showing fuel saved by air-heater without economiser.
(P. H. N. Ulander, *J. Inst. Fuel*, I, 188.)

heater tubes is important, and this usually varies from about 4 to 7 feet per second. The size and cost of the heater can be reduced by increasing the water velocity, and, consequently, the heat transfer coefficient, but moderate velocities are desirable for the reason given. The tubes vary in external diameter from $\frac{1}{2}$ to 1 inch, and the number required to give the necessary cross-sectional area is directly related to the water velocity adopted.

The tubular surface area (and the tube length, when the diameter is determined), depends upon the weight of water to be heated ; its temperature rise ; the heat transfer coefficient ; and the mean temperature difference between the

steam and water. The number of water passes required in the heater will vary according to the permissible height or length of the heater, but when the two-tube-plate construction is used, the length between tube plates should not be greater than about 100 tube diameters, or tube-supporting plates may be necessary. These, it will be appreciated, interfere with the free flow of steam internally.

AIR PREHEATERS

The air preheater is an important auxiliary in modern boiler installations, and transfers heat from the flue gases to the air fed to the furnace for combustion purposes.

The effects claimed from air preheating are as follows :—

(1) The recovery of heat from the flue gases reduces the heat loss from this source. It is claimed that every 30° to 35° F. reduction of the flue gas temperature in this way results in a fuel saving of 1¼–1½ per cent.
(2) The flame temperature in the combustion space is increased with consequent greater rate of heat transfer by radiation (cf. Chapter VIII), thus increasing the effective heat transfer and increasing the amount of steam made per boiler.
(3) With higher flame temperature, the fuel can be burned with less excess air. In addition to the better utilisation of heat (Chapter V) there is also from this cause a reduction in the power expended in producing draught.
(4) Frequently lower grades of fuel, or different grades, can be burned successfully with preheated air, which would not be properly burned with cold air. Some coals can be burned satisfactorily only with preheated air.

Fig. 118 (Ulander, loc. cit.) indicates from test results the advantageous effect on fuel consumption of using preheated air.

Types of Air Preheater. Preheaters are either regenerative or recuperative, and in general principle are similar to the regenerators and recuperators used in furnace work (Chapter XVIII).

With recuperative heaters the gases pass on one side and the air on the other side of the tube or plate wall, the heat transfer taking place through the plate. All tubular and plate heaters come within this group.

Recuperative heaters are more economic on small installations and where a small proportion of the available heat has to be recovered, but deposits on the plates affect the rate of heat transfer.

Regenerative heaters operate by passing hot flue gases and cold air alternately through the heating elements. Heat is picked up by the elements from the gases and in the next stage of the cycle is transferred to the air. Only one heater of this type is marketed in this country for use with steam boilers. In this apparatus, the element plates are assembled in a cylindrical housing, which is rotated slowly so that the elements pass alternately through the flue gases and air during each revolution. The elements are heated by the gases and then cooled by the air, thus the heat transfer is continuous.

Temperature Limits. With each method of burning fuel in a boiler there is a limit to the temperature of the combustion air, and if this limit is exceeded the maintenance charges on stokers, burners or furnace parts increase rapidly. The maximum hot-air temperatures are approximately :—

Method of firing	Temperature	
	° F.	° C.
Hand	250–300	121–149
Sprinkler and chain-grate stokers	250–300	121–149
Retort stokers	450–500	232–260
Pulverised coal or oil firing	Up to 600	Up to 315

On the gas side of an air heater the maximum entering temperature is restricted to 750°–800° F. (399°–427° C.), which is the temperature that mild steel will safely withstand should the air flow cease. Beyond this limit it is necessary to introduce special designs or special operating precautions, or to make the heater elements of high-temperature resisting alloy steels, thus considerably increasing the cost.

Many air heater troubles result from working with too low an exit gas temperature. The corrosion difficulty has been explained when dealing with economisers. In addition, the deposition of moisture will cause dust to stick to the heating surfaces, which impairs heat transmission. The prevention of condensation within an air heater is therefore imperative.

The Prevention of Condensation. With recuperative heaters the most satisfactory method of maintaining the requisite element plate temperature at the air inlet end is to return some of the heated air to the forced draught fan inlet. Thus the temperature is raised to just above that at which condensation would occur in the flue gases. As the temperature of the element plates cannot fall below the entering air temperature, this method has successfully overcome corrosion and deposit troubles. An amount equivalent to 30 or 40 per cent., or even more, of the air passing through the heater may have to be re-circulated in this manner to ensure that the air temperature at the heater inlet is at the required figure, usually between 115° and 130° F., according to fuel and combustion conditions.

The makers of the regenerative heater claim that re-circulation of air is not necessary in their design. Because of its rotation and of the absence of sections with a cross-flow of gas and air, cool spots caused by uneven air and gas distribution at the air inlet end do not occur, and all the element plates are subjected to the same minimum temperature conditions.

In boilers of the Lancashire, Economic, marine Scotch types, etc., the gas temperature at the heater outlet should not at any time during operation fall below 280°–300° F. (138°–149° C.). In water-tube boilers, which are usually larger and obtain their air supply from above the boilers where the air temperature is already high, combustion conditions are in general steadier and more closely controlled, so that gas temperatures down to 240° or 230° F. (115°–110° C.) may be permitted when the fuel and combustion conditions are favourable.

The most satisfactory means of controlling the gas outlet temperature above the required limit at reduced loads is to provide air by-pass ducts and dampers. By opening the air by-passes, some air is short-circuited past the heater, less heat is taken from the flue gases and the gases then leave the heater at a higher temperature. The heater elements are thus maintained at the proper temperature.

For satisfactory control of an air heater it is usually necessary to provide a gas by-pass having a tightly shutting damper, isolating dampers to prevent gas flow through the heater during starting-up, banked-fire and shutting-down conditions, and air by-pass ducts with regulating dampers. On recuperative heaters, air re-circulating ducts and regulating dampers are advisable. A forced draught fan should be installed. Many air heaters are designed for a low draught loss on the gas side so that they can work in conjunction with chimney draught only, but it is usually found advantageous to fit an induced draught fan.

AUXILIARY POWER AND STEAM UTILISATION

The efficient generation of steam is of primary importance if fuel supplies are to be conserved, but the utilisation of the steam produced is also of equal importance. Unfortunately, the average works tradition is such that the

steam-plant engineer can rarely exercise any influential control over the production equipment in which the steam is used, so that boiler and power-house economies are liable to be neutralised by wasteful production methods. The plant engineer can, however, control the fuel-consuming appliances required to operate his auxiliary boiler and power-house plant, and brief mention will be made of some of the aspects of the subject that need consideration to achieve the best results.

Most modern auxiliaries are either steam or electrically driven, and the final decision of selection will depend largely upon individual conditions. The advantages of a steam drive are that steam plant will stand a certain amount of abuse before complete failure supervenes ; it is useful as a stand-by in the event of an electric power failure ; it can be used for starting up the plant when no electric power is available until the main generator is on load ; and the exhaust steam can be used for feed heating or for process work.

Steam plant has, on the other hand, usually high initial, maintenance, foundation and lubrication costs. Its thermal efficiency is low (unless the exhaust steam can be utilised), and it occupies a comparatively large floor area. Steam plant cannot be readily adapted for remote control, thus preventing the provision of centralised controls, and there are the inevitable losses and leakages which necessitate extra feed water make-up.

An electrical drive has the advantage of a good speed regulation, and it lends itself to centralised control, either automatic or remote. It is very flexible, and the initial, foundation and lubrication costs are low. Moreover, the floor space required is relatively small. Electrical drives are, of course, useless unless electrical power is available for starting-up purposes.

When electrical energy is privately generated for process work by a straight condensing turbine or reciprocating engine having no bled-steam connections, the provision of steam-driven auxiliaries exhausting to feed water heaters and evaporators, etc., will raise the feed temperature and thus increase the overall thermal efficiency of the installation. When both steam and power are generated for process work, the choice of drive frequently depends upon whether there is a surplus of steam or of electrical power. If there is normally a good balance, then the capacity of the main generator to meet auxiliary loads must be taken into consideration. In this respect, it must be remembered that a main turbine or reciprocating-engine generator has a higher efficiency than any small turbine or engine. Again, when steam is generated for process purposes only—all electrical power being purchased outside—the installation of back-pressure turbines and electrically driven auxiliaries may reduce power costs.

In the ordinary boiler house, power is required for operating mechanical stokers, fans, coal and ash elevators, feed pumps, etc., while steam may be required for feed water heating and occasionally to ensure efficient combustion. To these must be added the steam necessary to operate air ejectors, soot blowers and similar appliances.

Generally speaking, the equivalent steam consumption for boiler house auxiliary purposes in industrial plants should not greatly exceed about 10 per cent. of the total amount of steam generated, but if it does, an investigation into the cause of the excess is indicated. Mechanical stokers should not take more than 0·3–0·5 per cent. of the total steam, or its equivalent, and the coal and ash elevators may require from 0·8 to 1·0 per cent. The boiler feed pumps, on the other hand, may use from 2·5 to 3·5 per cent. of the total steam generated.

When steam is required for feed water heating it can usually be obtained by bleeding from the main prime mover, or by using the exhaust from the auxiliaries, but steam for forced draught steam jets as used on Lancashire and similar boilers must be taken from the high pressure line, and may use from 3 to 4 per cent. of the total steam generated.

BOILER FEED PUMPS

Boiler pumps fall into two broad classes :—

(1) Displacement pumps, in which the feed water is displaced from the suction to the discharge by the reciprocating motion of a bucket or plunger.
(2) Centrifugal pumps, in which the feed water flows through the pump by reason of the centrifugal force imparted to the water by the rotation of one or more impellers through which the liquid flows from the suction to the discharge.

Direct-acting feed pumps of the reciprocating type are displacement pumps ; they are by far the most common type of feed pump in use.

Centrifugal feed pumps are built for capacities up to 500 tons per hour, and are usually provided in the larger power plants.

A little consideration will show that the first requirement of a boiler feed pump is reliability. Operating, maintenance and first costs are important, but fade into insignificance when a pump ceases to function at the time of a heavy steam demand on the boiler or boilers that it serves. Due to the designs of a number of leading manufacturers, the direct-acting type of feed pump has a well established reputation for reliability in Great Britain, and these and other makers have built up a similar reputation for the centrifugal type of boiler feed pump now used in many industrial and power-station plants.

The direct-acting pump is designed for duties up to 200,000 lb. of water per hour, but is more generally applied as a unit to boilers totalling about one-half this capacity. The size of pump required for particular conditions will, of course, depend upon the maximum evaporative capacity of the boiler or boilers, due allowance being made for the pressure and temperature conditions obtaining. The direct-acting pump can be constructed as a single- or two-cylinder unit, vertical or horizontal, depending upon the purpose for which it is required. The vertical simplex type saves floor space and has only one steam and one water end to keep in order, while the horizontal type may cost slightly less.

The principal differences between the various designs of pumps apart from those just mentioned lie largely in the steam-distribution mechanism and the amount of expansive working thereby rendered possible. Normally the cut-off is not less than about 0·7 of the piston stroke. When the exhaust from the pump can be utilised in a feed heater, however, a lower steam consumption may be partly neutralised if a more intricate valve gear, necessitating frequent maintenance, is required. In such conditions it may be advisable to install a more rugged, if less economical form of pump.

Pump speeds vary from 50 to 80 feet per minute, but generally the slower pumps suffer the least wear and tear, and therefore have lower maintenance costs. Bearing surfaces, particularly the moving parts of steam valves, should be ample, and full provision should be made to allow of the rapid overhaul of steam and water ends.

The direct-acting pump is probably the best appliance for boiler feed purposes for pressures up to about 200 lb. per square inch pressure. In plants using high steam pressures and high superheat temperatures a reciprocating form of pump can still be employed, but in this instance a motor drive must be adopted. The pump is usually of the multi-throw type, driven through reduction gearing, and the discharge pressure varies almost directly as the output. As will be shown later, the centrifugal pump installed in many modern boiler houses can set up a high pressure differential when the feed check is practically closed. The efficiency curve of a reciprocating pump is reasonably uniform over the

range of its capacity, and it has, in fact, been installed to serve some of the high-pressure boilers now in service.

Centrifugal Boiler Feed Pumps. The direct-acting pump, as mentioned above, is constructed for duties up to 200,000 lb. of water per hour, but, according to individual circumstances, its economical upper limit is in the region of 150,000 lb. per hour. The lower economical limit of the centrifugal boiler feed pump is about 75,000 lb. per hour, so that a pump purchaser in the 75,000–150,000 lb. per hour range must decide on the respective merits of the

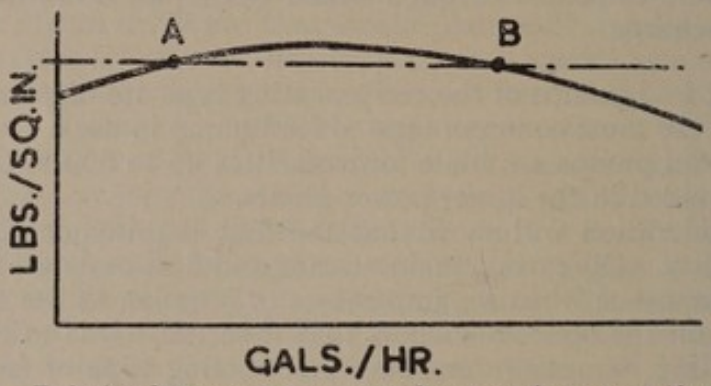

FIG. 119. Relation between pressure and output for a centrifugal pump.

two types. In such circumstances the best solution is probably to consult a manufacturer who produces both direct-acting and centrifugal boiler feed pumps, and carefully to consider the data thus obtained. In general, the centrifugal pump gives satisfactory service when its output is relatively large compared with its pressure.

The pressure developed by a centrifugal pump varies roughly as the square of its peripheral speed, and the blade angle at the circumference of the impeller. With an ordinary type of centrifugal pump, the pressure rises at first and then falls with increased output, so that a curve of the form shown in Fig. 119

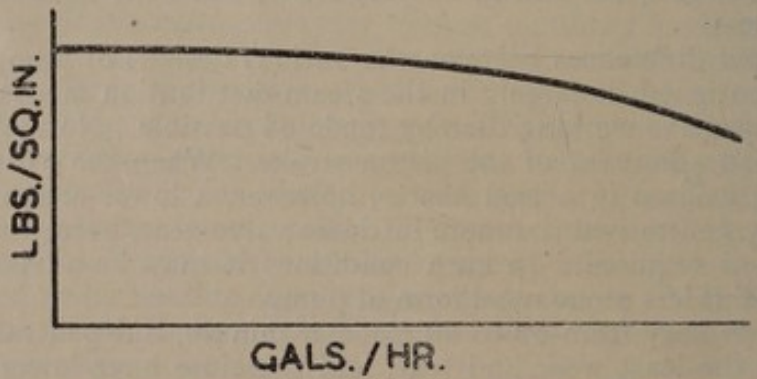

FIG. 120. Characteristic curve of a centrifugal pump.

results when pressure is plotted against capacity at constant speed. Obviously, a pump working under such conditions is liable to oscillate between two rates of delivery, A and B, and each change from one point to the other causes a pressure impulse to be set up in the pipe-line.

This unstable method of operation is unsuitable for boiler feed lines because of the danger of water hammer, and it is therefore necessary to use a pump that works on the " falling " part of the characteristic curve, as indicated in Fig. 120. This is particularly important when two or more centrifugal pumps are to serve in parallel, otherwise there is the possibility that one pump will give its maximum discharge while the others are practically idling and heating

up. The slope of a characteristic curve should not, however, be too steep, or there will be a large pressure difference between full-load and no-load conditions. The permissible pressure increase may vary from about 8 per cent. with a constant-speed electrical drive to perhaps 10 per cent., or slightly more, when a steam-turbine drive is used. Too large a pressure increase reacts on the pressure-bearing parts of appliances such as economisers, feed heaters, etc., located between the feed pump and the boiler.

Centrifugal Pump Construction. Centrifugal pumps can be designed to discharge into a volute surrounding the impeller, and in this arrangement the kinetic energy of the water is converted into pressure energy inside the casing itself. The method is unsatisfactory in that heavy bending stresses may be imposed on the impeller shaft, due to unequal radial thrust, and most designers of modern feed pumps provide a ring of diffusers around the impeller periphery. The diffusers gradually convert the kinetic into pressure energy, and also serve to guide the flow when a multi-stage pump is used.

For a single-impeller centrifugal pump to operate at the usual boiler pressures, the impeller speed and diameter must be large, and this may result in restricted water passages through the impeller. Again, the frictional losses of an impeller vary approximately as the cube of the diameter and the square of the velocity, so that a considerable reduction in losses can be achieved by using two or more comparatively small impellers working at a lower speed as against one large impeller working at a high speed.

The multi-stage construction has the further advantage that the subdivision of the total increase in pressure lessens the impeller clearance losses, and is now largely used for boiler feed pump purposes.

Selection of Pumps. Other points to be considered when analysing the construction of a centrifugal feed pump are the method of allowing for axial thrust, the prevention of leakage where the pump shaft projects through the casing, the provision for expansion at high temperatures, and the means adopted to minimise overheating of the pump under closed feed valve conditions.

One method of reducing the effect of axial thrust is to arrange the impellers so that the respective thrusts tend to cancel out, but this method usually also requires the provision of a thrust bearing to absorb any residual forces. Perhaps the most universally adopted construction is the balancing disc or drum, which, due to the water pressure on its face, opposes the axial thrust on the impellers.

The arrangements for preventing leakage between the pump shaft and its casing range from the simple packed gland to elaborate water-cooled devices. Some designers incorporate a form of labyrinth gland in the balancing drum which reduces considerably the pressure and temperature to which the outer glands are subjected, while others rely on separate labyrinth glands and water or air cooling.

In pumps for very high pressures and temperatures, it may be necessary to provide an auxiliary centrifugal pump solely for water-cooling purposes. Radial expansion of the pump can be largely counteracted by adopting centre-line suspension and longitudinal expansion by fixing one end and leaving the other free to move, but the merits of a vertical method of suspension should not be overlooked in this respect. For very high temperatures, at least one maker completely water-jackets his pump. Overheating under closed feed valve conditions is usually reduced by providing a leak-off valve, manually or automatically controlled.

PUMPING HOT FEED WATER

The saving by using hot condensate as boiler feed water is two-fold, firstly because its use means a direct economy in fuel, and secondly because its purity will do much to prevent scale in the boiler feed system. It is found, however,

that valuable condensate is often wasted because of the difficulties of pumping hot water, and this means that tons of fuel represented in the hot condensate are run down the drain while more fuel is used in heating cold feed water. While it is agreed that there are theoretical limits and practical limits in handling hot condensate, a very considerable economy can be effected if the necessary consideration is given to the design of the pump, suction piping and position of feed tank. Experience has shown that the source of the trouble is usually the suction piping or the position of the feed tank.

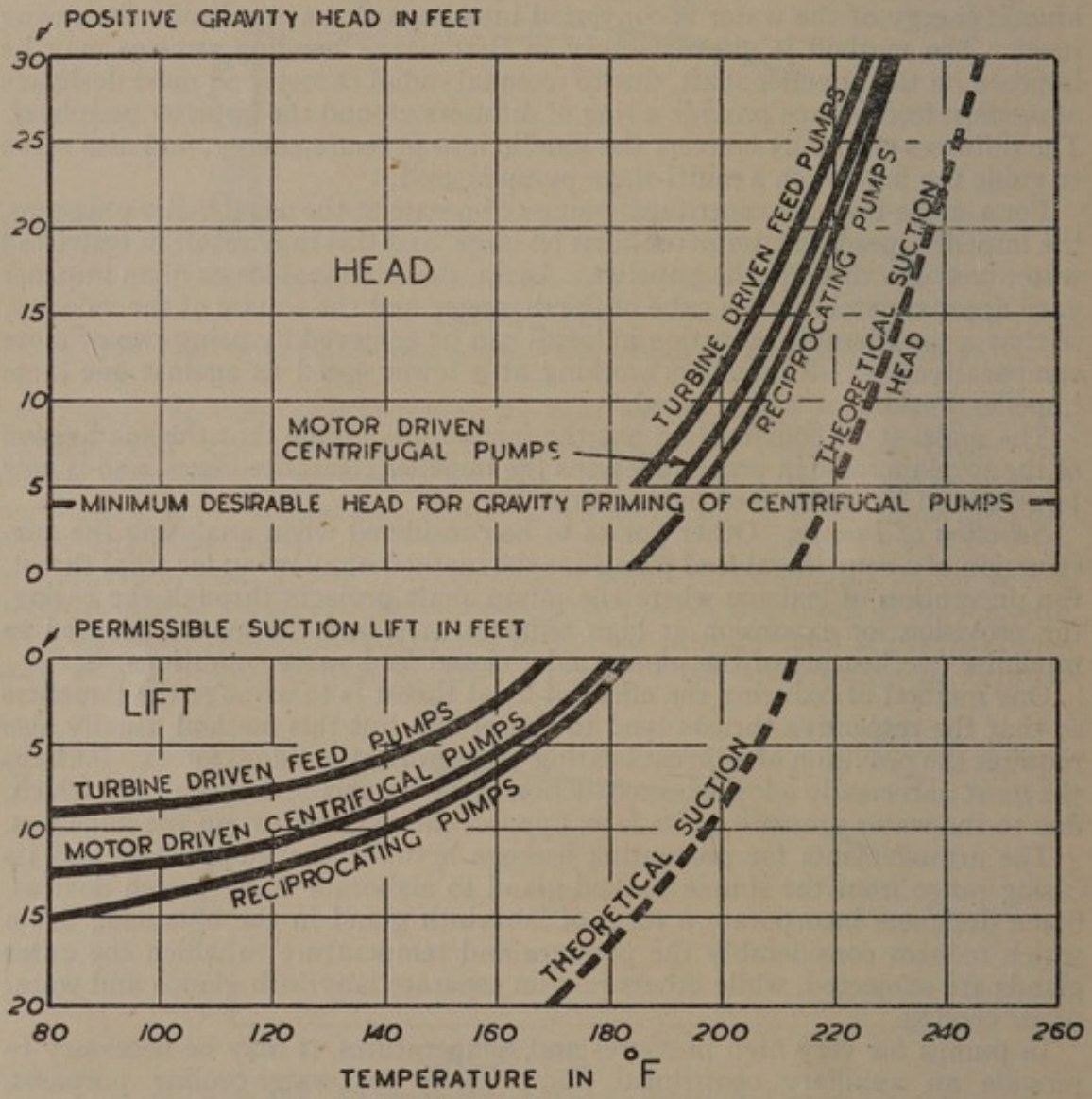

FIG. 121. Suction heads allowed for standard designs of pumps.

The secret of successfully pumping hot feed water is attention to the correct conditions on the suction side of the pump. Flow from a pump is caused by the pump, but the flow of a liquid into a pump is caused by forces outside the pump. A pump should, therefore, be so arranged that, from the source of feed water supply, a pressure is created in the suction chamber of the pump greater than the pressure at which the water will vaporise. When water is being pumped from a tank containing a free surface, the difference between the pressure on the surface of the water and the pressure in the suction chamber of the pump is the only pressure available to cause the flow of liquid into the pump. Fig. 121 shows the usual pressure difference allowed (expressed in feet of water)

for the average installation of various types of pump of standard design. The dotted curve shows the boiling point of water. The pressure difference allowed includes a reasonable allowance for the friction losses in the suction pipe line and suction passages and valves in the pump, and includes a margin to cause the liquid to follow the motion of the moving parts of the pump without vaporisation occurring.

Generally speaking, suction heads require to be greater for high speed or large capacities than for low speed or small capacities.

Before water can flow into a pump, the air or vapour in the suction line must be evacuated sufficiently to reduce the pressure in the pump to cause the liquid to flow into the suction chamber. Displacement pumps do this automatically when they are started. In centrifugal pumps, the liquid must be present in the first impeller before the centrifugal force to cause flow can be imparted to the liquid, and this requirement necessitates that the pump be placed relatively to the source of water supply so that the water flows by gravity into the impeller. If this cannot be done, special priming devices must be used to cause the evacuation of the air and flood the pump with water so that the motion of the pump can create the centrifugal force to cause a flow from the pump which will establish the flow from the feed tank into the pump.

A centrifugal boiler feed pump should never be arranged to have a suction lift unless the site conditions are such that it is impossible to arrange the pump and the feed tank to give a gravity flow into the pump.

Table 67 gives for direct-acting feed pumps the information in tabular form that is also in Fig. 121.

The suction piping from the feed tank to the pump should be as short and as free from bends as possible. Sharp right-angled bends should certainly be avoided, and the suction piping and valves should be designed to reduce the friction losses to a minimum.

It will be seen, from the figures given, that with the temperature above 175° F., the feed tank must be placed higher than the feed pump to ensure a pressure at the pump suction. The head required between the pump suction and the water level in the tank must be sufficient to prevent the water vaporising in the pump end.

TABLE 67. SUCTION LIFT AND HEAD FOR VARIOUS FEED TEMPERATURES. DIRECT-ACTING FEED PUMPS

Temperature ° F.	*Lift* in feet to suction valves
130	10
150	7
170	2
175	0
	Head over suction valves in feet
190	5
200	10
210	15
212	17

For feed temperatures *over* 212° F., it is necessary to provide a suction head of 17 feet plus a head equal to the gauge pressure corresponding to the feed temperature. If the feed temperature is over 212° F. and the heating is all carried out in the feed tank, the tank must be sealed and elevated at least 17 feet above the feed pump, or a booster pump provided to give the necessary suction head. Alternatively, the feed heating can be carried out in a direct

contact or a surface heater and suitable booster pumps provided to give the necessary pressure at the suction of the feed pump.

These briefly are the principal points requiring consideration in pumping hot condensate, but if they are carefully considered and the plant correctly arranged, there should be no difficulty in handling hot water, with increased plant efficiency and a considerable saving in fuel.

MAINTENANCE OF FEED PUMPS

In the maintenance of feed pumps, there are two broad principles to be followed :—

(1) The various component parts of the pump should be maintained in good condition so as to reduce to a minimum all possible leakages of water from the discharge of the pump to the suction of the pump or to any intermediate pressure within the pump. All discharges of water from the pump to the atmosphere should also be reduced to a minimum. Any leakage of water from a place of high pressure to another place at a lower pressure represents a loss in pumping energy, and such loss should be reduced to a minimum by maintaining the pump in an efficient condition.

(2) Similarly, every care should be taken to ensure that the steam end of a steam-driven pump is maintained in the best possible condition so as to reduce to a minimum all possible leakages of steam from the steam supply pressure to the exhaust pressure or to the atmosphere. Such leakages represent a loss of heat energy and, where the steam escapes to the atmosphere, there is also a loss of distilled feed water.

UTILISING EXHAUST STEAM

The subject of exhaust steam utilisation is discussed in detail in Chapter XXIII. Here reference is made only to its use for heating feed water.

The theoretical and practical limits in handling heated feed water have been illustrated and described in relation to Fig. 121, and all practicable steps should be taken to ensure that the feed water which the feed pump is required to handle should be at the maximum possible temperature which the pump is capable of dealing with. For this purpose, consideration should be given to the position of the feed tank and the design and layout of the suction piping to enable the pump to operate with the maximum possible feed water temperature.

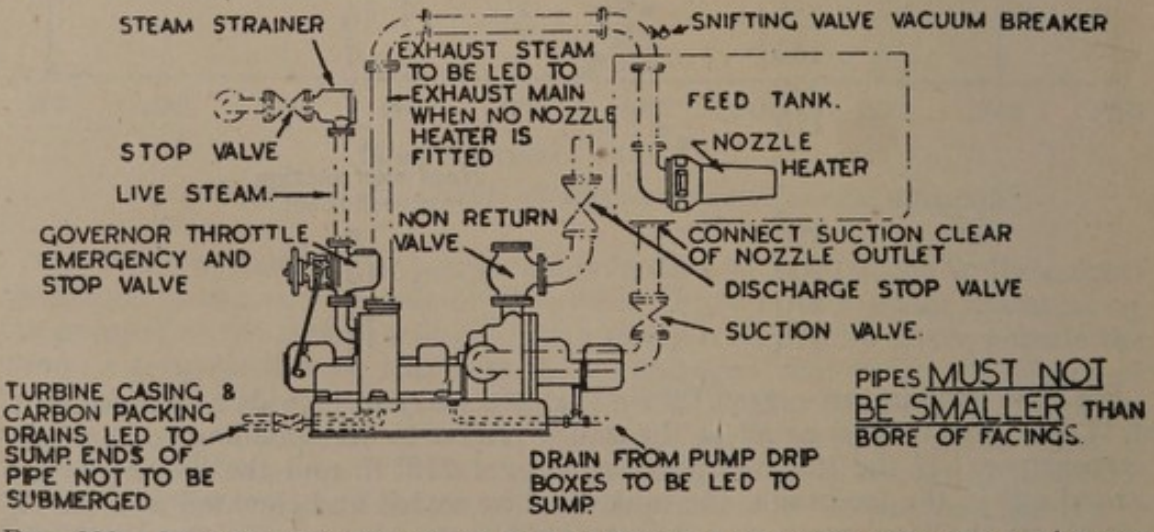

FIG. 122. Typical arrangement of turbine-driven feed pump exhausting to nozzle heater in feed tank.

The simplest method of using exhaust steam is to discharge it into the feed tank through a mixing nozzle. The exhaust steam from the turbo feed pump is not contaminated with oil and can, therefore, be condensed by direct contact with the feed water in the feed tank, thereby conserving the whole of the heat in the exhaust steam and also the distilled water formed by the condensation of the exhaust steam. A convenient arrangement for a turbo-driven feed pump is illustrated diagrammatically in Fig. 122. The quantity of exhaust steam to be dealt with is considerable and the guidance of feed pump manufacturers should be sought to ensure that the best possible arrangement of satisfactory heating is obtained and that the arrangement adopted will be entirely satisfactory so far as the operation of the feed pump is concerned. It may be that the arrangement of the plant and the operating conditions are such that it would be preferable to provide a surface feed heater on the discharge side of the feed pump into which the turbine of the feed pump discharges the exhaust steam for feed heating purposes, the feed water being discharged by the pump through the tubes.

DRAUGHT PRODUCTION

The rate at which coal can be burned in a boiler furnace per square foot of grate area depends, among other things, upon the characteristics of the fuel being burned : a non-caking coal, for instance, offers less resistance to air flow than does a caking coal, and thus absorbs less of the available air-pressure head. Again, some coals of the anthracite class will hardly burn at all unless provided with a high draught intensity, while others require only a low draught, or disproportionate fly-ash losses result. For every class of coal, therefore, there is a rate of burning which gives the most efficient combustion conditions, and the means of providing draught must be selected with this in mind.

Chimneys. The most common method of supplying air to boiler furnaces is by means of a chimney, and this has been discussed in some detail in Chapter VI. The height and diameter of a chimney must be determined from a consideration of present and future draught requirements, so that in a business likely to expand rapidly, it may be financially advisable to provide a larger chimney than can be justified on immediate needs.

Steam Jets. A chimney, however, is a very inefficient appliance for the production of draught, depending, as it does, upon a high gas temperature at the boiler flue outlet. Moreover, the flexibility of a chimney is limited. In such circumstances, a boiler operator may adopt some system of artificially-produced draught, the most common being that in which steam jets are fitted at the chimney base or below the firebars. The first method is used principally in vertical or locomotive boilers, and causes an induced draught which permits a higher rate of combustion or a reduction in the height of chimney required to produce a given draught. The second method is widely used in horizontal internal-flue boilers and in conjunction with various types of mechanical stokers fitted to such boilers. It causes a forced draught of air through the fuel bed, and consequently lessens the draught required to draw off the products of combustion.

Steam jet draught also assists in cooling the firebars and influences clinker formation and, in fact, is essential with certain classes of coal if combustion is to be ensured. When new, these jets serve the purpose and are at present fitted to many boilers having to burn inferior coals.

Steam jets should not consume more than 3–4 per cent. of the total steam generated by the boiler to which they are fitted, but unfortunately the jet orifices are liable to be considerably enlarged by erosive action. The greatly increased steam consumption that may arise from wear of steam jets has been indicated in earlier chapters.

Fan Draught. Artificial draught can be produced at a higher capital cost, but with greater economy and efficiency by installing a fan system. A great deal of the improved performance claimed for the modern internally-fired multitubular boiler can be attributed to the use of higher gas velocities, which the boiler operator must, of course, pay for in fan power. A saving in the height, and consequently cost, of a chimney is effected, but it must not be overlooked that most local authorities stipulate a minimum chimney height.

The use of fan draught allows the maximum amount of heat to be abstracted from the products of combustion before they are discharged into the chimney ; it makes possible enhanced rates of combustion of low-grade fuels together with higher furnace temperatures and rates of convective heat transfer ; and it permits rapid control of the boiler over a wide range of output. Fan draught can be applied as forced, induced or balanced draught, according to particular boiler requirements ; the first having the advantage that any air leakage is outwards ; the second that it probably gives a better distribution of air through the fuel bed ; and the third that it lessens the liability to cold air leakages into the furnace.

Fan Types. In considering the type of fan to be adopted for boiler draught

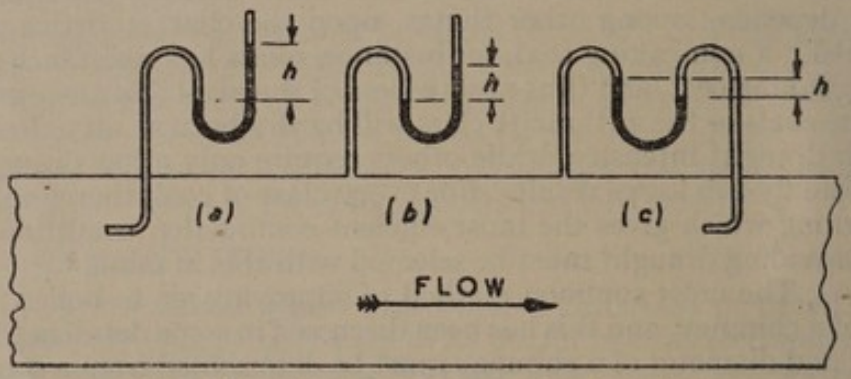

FIG. 123. Static and velocity heads at fan.

purposes it is necessary to enquire into the kind of pressure required. The total pressure produced by a fan is represented by the manometer at (*a*) in Fig. 123, and is the pressure due to the sum of the static and velocity heads. The static head can be indicated by a manometer of the type shown at (*b*), but to obtain the velocity head it is necessary to insert a tube into and facing the main stream of air and to connect its other end to the wall of the ducting, as at (*c*) in the figure (cf. Chapter IX). In boiler work it is the static head that has greatest importance, since it is by virtue of static pressure that the resistance to flow through, say, a compacted fuel bed, can be overcome.

The static head varies according to the fan blade curvature. Blades having a forward curvature, or those of the straight radial type, develop higher absolute air velocities than do those of the full backward-curved type, and they are less efficient. On the other hand, the backward-curved fan requires a higher peripheral speed for a given static pressure, and must therefore be of heavier build.

The resistance-overcoming properties of the backward-curved fan make it very suitable for forced draught work, but its necessarily heavier construction militates against its use for induced draught production. In the latter application there is a tendency for deposits to collect on the reverse faces of the blades, with obvious results on the fan balance, and extra thickness must be provided to allow for abrasion of the blades in service. A backward-curved blade would have to be of heavy section, as stated above, and it is therefore principally used for forced draught service, the radial self-cleaning form of blade being adopted for induced draught requirements. It will be appreciated, of course, that there

are several intermediate types of fan blades used for various boiler house purposes.

Fan Characteristics. As with the centrifugal boiler feed pump, the performance of a fan can be shown graphically by means of characteristic curves.

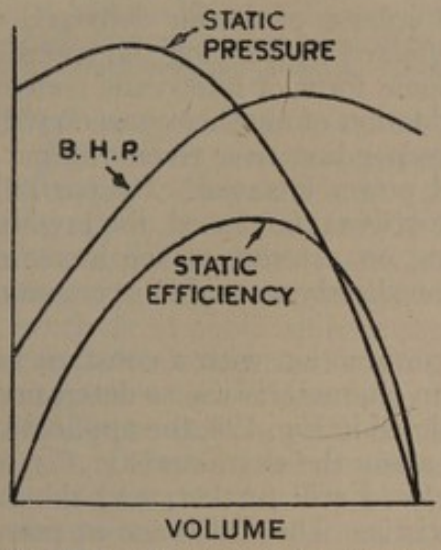

FIG. 124. Characteristic curve of forced draught fan.

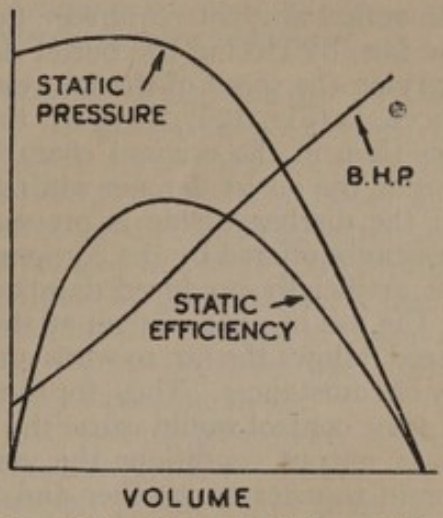

FIG. 125. Characteristic curve of induced draught fan showing diminishing B.H.P. with reduced consumption.

These are usually obtained by plotting static head against the volume discharged at a given speed, and the graphical information is completed by adding the corresponding curves of static efficiency and B.H.P., as indicated in Fig. 124. The characteristics shown are for a forced draught fan, and it will be noted that the pressure curve falls away rapidly after a maximum value. This is a

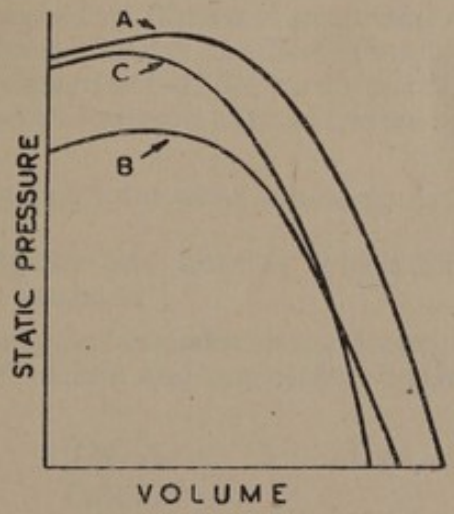

FIG. 126. Fan characteristics with different methods of control.

useful feature for forced draught work, as it means that the resistance must vary considerably before the volume delivered is appreciably affected, and it is also useful when two or more fans have to work in parallel.

A further point is that the B.H.P. curve attains a maximum value and then recedes, and this self-limiting characteristic prevents damage to the driving motor should the resistance suddenly drop to zero value. The difference between the characteristics shown in Fig. 125 and the curves in the preceding

figure should be noted. The pressure-volume curve is much less steep for the radial fan, and the B.H.P. increases as the volume discharged rises, so that provision must be made to prevent over-loading. The efficiency, also, is slightly less than that of the forced draught fan.

Fan Control. To operate a fan at maximum economy, some form of control must be introduced to relate the pressure and volume of the air delivered to the actual amount required. Control can be effected by varying the speed of the fan, by closing the outlet damper, or by some form of inlet-vane control. Varying the speed of the fan results in the production of another characteristic curve, B (Fig. 126), in which the pressure corresponding to a given volume is less than in the original characteristic, A, and power is saved. By partially closing the outlet damper while maintaining the original fan speed, the pressure on the discharge side is prevented from falling on account of the increased resistance offered by the damper, and power is needlessly wasted in overcoming the artificially-produced damper resistance.

The use of vane control at the fan inlet, in conjunction with a constant fan speed, allows the fan to work on a number of fan characteristics, as determined by circumstances. Thus, for the conditions depicted in Fig. 126, the application of vane control would cause the fan to operate along the characteristic, C ; for lower output conditions the vanes would be closed still further, and the fan would transfer to another and lower characteristic. The difference in power economy between speed and vane control depends upon the type of driving medium and its efficiency, and upon the fan characteristics.

CHAPTER XV

BOILER FEED WATER PURIFICATION

Mineral salts in water—Effects of scale—Water treatment—The lime-soda process—The zeolite process—Demineralisation—Internal treatment and water conditioning—Feed water for high pressure boilers—Corrosion—De-aeration—Caustic embrittlement—Blow-down, its control, technique and disposal—Control of water treatment—Sampling and testing.

ON its journey from cloud to earth, rain becomes saturated with the gases of which the atmosphere is composed. In the vicinity of industrial areas, where there is atmospheric pollution, gases such as ammonia (NH_3), hydrochloric acid (HCl), sulphur dioxide (SO_2), etc., will be taken into solution in addition to carbon dioxide (CO_2) and oxygen (O_2). The latter are, however, the most important, and rain when it reaches the earth may contain as much as 15 cubic centimetres of carbonic acid gas (CO_2) and 30 cubic centimetres of free oxygen per gallon. This dilute solution of carbonic acid dissolves limestone—which is insoluble in water free from CO_2—together with other soluble salts, and carries away insoluble matter in suspension as silt.

Some of these substances form solid deposits as the water is heated to boiling point ; others are not deposited until a higher temperature is reached or water is removed by evaporation. A water may therefore form deposits in both economiser and boiler and, as might be expected, waters from different sources show wide variations in behaviour when used as boiler feed ; some cause corrosion and little scale, others deposit thick scales of varying composition and texture. On the other hand, some waters can form soft deposits in the economiser and hard deposits in the boiler, the composition of the one differing radically from that of the other.

It will be readily understood, then, that no natural water is entirely suitable for use as boiler feed, and some form of treatment is required to counteract the effects of the dissolved salts.

The chief scale-forming substances which are deposited in boilers are compounds of calcium and magnesium (lime and magnesia) and as these substances are also the cause of hardness in water, it follows that hard waters usually form large amounts of scale, although not all the hardness salts take part.

The calcium and magnesium salts in natural waters may be divided into two groups :—

(1) The bicarbonates, which form what is commonly known as " temporary hardness."
(2) The sulphates, chlorides and nitrates, which form what is commonly called " permanent hardness."

When the water is heated, the bicarbonates are decomposed, giving off carbon dioxide and the resulting calcium and magnesium carbonates are deposited as solids.

$Ca(HCO_3)_2$	$\longrightarrow$	$CaCO_3$	+	H_2O	+	CO_2
Calcium bicarbonate		Calcium carbonate		Water		Carbon dioxide
$Mg(HCO_3)_2$	$\longrightarrow$	$MgCO_3$	+	H_2O	+	CO_2
Magnesium bicarbonate		Magnesium carbonate		Water		Carbon dioxide

Magnesium carbonate is more soluble than calcium carbonate, but it is decomposed under boiler conditions to form magnesium hydroxide, which is less soluble than calcium carbonate.

$$MgCO_3 + H_2O = \underset{\text{Magnesium hydroxide}}{Mg(OH)_2} + CO_2$$

A water containing temporary hardness will therefore deposit solids whenever the temperature rises to about 70° C. (158° F.), i.e. in economisers, preheaters, etc. as well as in the boiler itself.

The permanent hardness salts do not form deposits when the water is boiled at atmospheric pressure but under boiler conditions they concentrate in the boiler water. They are highly soluble with the exception of calcium sulphate, which crystallises out on the heating surfaces, and this substance is the chief constituent of thick hard boiler scales. Silica is also present in natural waters, though only to a small extent compared with the hardness salts, but it accumulates in the boiler water and thin hard scales of calcium and magnesium silicates can be formed. The magnesium permanent hardness salts decompose in the boiler and give rise to acidic conditions which may cause severe corrosion.

EFFECTS OF SCALE

It has long been recognised by boiler engineers that natural waters form scale in boilers and in economisers or feed water heaters. The amount and texture of the scales formed vary widely from one supply to another. Some scales are hard and very difficult to remove mechanically ; others are soft and easily removed ; there are also waters which form practically no scale but deposit sludges which can easily be removed from the boiler by means of a hose. It has become recognised boiler practice to shut down boiler plant periodically for the purpose of removing the solid deposits, and the expenses incurred will of course, depend on the frequency of shutting down and the difficulty of removing the scale.

The first effect of scale formation is therefore a recurrent expenditure in labour and a loss on capital while the plant is shut down for cleaning, together with the attendant disadvantages of having the boiler, and possibly other plant dependent on it, out of commission.

In view of the many statements which have been made regarding the loss of efficiency due to scale formation in a boiler, it seems worth while to try to get a clearer idea of the effects of scale. In a boiler the formation of scale on the heating surfaces reduces their conductivity and the rate of heat transfer to the water so that in scaled multitubular boilers the unused heat in the gases leaving the hottest banks will be absorbed by the more remote banks of tubes, and if these in turn became scaled the region of heat transfer would be removed to the economisers and preheaters. However, tube failure in the hotter regions of the boiler will usually occur long before this process is carried to the limit, particularly in boilers where the greater part of the heat is transferred by radiation.

In practice, therefore, scale is not likely to cause a big rise in stack gas temperature (with a corresponding large decrease in the efficiency of heat utilisation) in multitubular boilers fitted with economisers and air preheaters. On the other hand, in Lancashire or locomotive boilers not fitted with an economiser, scale on the heating surfaces may result in rise in stack gas temperature with loss of efficiency. The effect of scale on fuel consumption is therefore a function of the design of the boiler plant, and in general the simpler the design the greater the effect of the scale. The most important consideration is that the temperature of the metal increases and eventually reaches the point at which mechanical failure occurs (cf. Chapter VIII, pp. 121 and 126).

Many figures have been published for the loss in heat transfer due to varying thicknesses of scale, and a study of these has been made by E. P. Partridge

(*Engineering Research Bulletin*, No. 15, June, 1930: "Formation and Properties of Boiler Scale," Department of Engineering Research, University of Michigan). The following are quotations from the conclusions in this investigation :—

"Boiler scales have low heat conductivities; the maximum value of the heat conductivity coefficient is approximately 14·0 B.Th.U./sq. ft./hr./°F./in. Dense, compact scales, such as are typically formed by calcium sulphate, show an average value of approximately 15, while very porous scales may have coefficients as low as 0·6. The extremely low conductivity of porous scales is due to the fact that during boiler operation the interstices are filled with steam, forming what is, in effect, a vapour film mechanically held on the boiler heating surface. The high resistance of vapour films to heat flow is well known.

"While boiler scale is a material with low heat conductivity, its effect upon heat utilisation is slight. Older figures for heat loss due to scale range up to 15 per cent. loss for $\frac{1}{16}$ inch of scale, but it is probable that the actual loss is not more than 2 per cent. The latter figure is supported not only by theoretical calculations, but also by the result of recent experimental investigations.

"While the loss in boiler efficiency due to scale is small, the fact that there is any loss at all would still justify the application of scale prevention methods. Contemporary boiler operation demands the elimination of all losses up to the point of maximum economy of operation. The prevention of scale by adequate chemical control would be sound economics even if boiler efficiency were the sole consideration. Actually, however, there is another factor which is growing in importance as boiler pressures and rates of driving steadily increase. This is the failure of tube surfaces due to overheating as a result of scale formation."

An illustration of this was given in Chapter VIII when dealing with thermal conductivity. It was there calculated that when the safe working maximum temperature for boiler tubes is 900° F., a coating of scale of only $\frac{1}{7}$ inch thickness would be sufficient to cause this limiting temperature to be reached. The scale in this example had a thermal conductivity of 18 B.Th.U./sq. ft./hr./°F./in. thickness. Obviously if the scale had been of the porous variety, a very much lesser thickness than this would have led to tube failure.

The losses in money and time occasioned are sufficient themselves to warrant careful examination of any methods of preventing scale which are simple and relatively inexpensive, and for many years increasing attention has been paid to the conditioning of boiler feed water by the addition of suitable chemicals.

SCALE PREVENTION

It is essential to analyse any scales which are formed in a boiler as a knowledge of their composition is a valuable aid in devising water treatment for their prevention.

Chemical analysis has shown that the chief constituents of hard scales are calcium sulphate and/or calcium and magnesium silicates, and that the soft loose scales or sludges are composed of calcium carbonate and magnesium hydroxide. Broadly speaking, permanent hardness forms hard adherent scales; temporary hardness forms soft scales or sludges. It follows that the best method of preventing deposits is to subject the water to a preliminary treatment which will effect the removal of scale-forming salts and of substances that may cause corrosion. Any physical or chemical means whereby this can be done cheaply and efficiently can be used.

METHODS OF WATER TREATMENT

There are three methods of treating water for boiler feed and these can be modified or combined to suit the type of water or boiler operating conditions.

(1) Precipitation processes, in which chemicals are added to precipitate calcium and magnesium as compounds of low solubility. The lime-soda

process is typical of this class, but other precipitating agents such as caustic soda and sodium phosphate can be used where the composition of the water permits and having regard to its ultimate use in boilers.

(2) Processes in which the hardness salts are removed without visible precipitation. The zeolite or base-exchange process may be taken as an example of this class.

(3) Both the above processes leave sodium salts in solution, but these do not cause hardness or scale. All salts can be removed by distillation or the process described below under the head " Demineralisation."

A brief and very general description of these processes is given below, but it must be made clear that the choice of process depends on the composition of the water supply and other factors such as cost, boiler working pressure, etc., and the decision on such points is best left to an expert as indicated at the end of this chapter.

LIME-SODA PROCESS

In this process hydrated lime and sodium carbonate are added to the water, whereby the calcium and magnesium are precipitated and removed by sedimentation and filtration. It is known as the lime-soda, or lime-soda-sodium aluminate process.

The process can be carried out in the cold but better results are obtained at elevated temperatures, preferably above 70° C. (158° F.). Plants should incorporate automatic apportioning gear, to ensure accurate addition of reagent to the correct volume of water, ample retention time for settling the precipitates —about three hours cold and two hours hot—and filters to remove the final traces of suspended solids. By the hot treatment the hardness can usually be reduced to negligible proportions, i.e. of the order of 1·0 part $CaCO_3$ per 100,000, but in the cold such a low figure is not reached except under very favourable circumstances.

When the volume of water to be treated is small, the treatment may be carried out in a tank, or in two tanks used alternately, so that the clear soft water from one may be fed to the boiler while the precipitates are settling in the other. The capacity of the tanks should be sufficient to allow the settling times suggested above, and provision should be made for withdrawing the settled precipitates from the bottom of the tanks periodically. Some precipitate should, however, always be left as a nucleus on which the precipitates in the next softening can form.

The reagents are usually mixed with water and added as a " milk," and it is most important that the quantities shall be strictly proportioned to the hardness and volume of water treated, a small excess of reagent being added to ensure maximum hardness removal. After the addition the contents of the tank should be stirred for a few minutes to ensure thorough mixing before settling.

Whilst the use of aluminate is optional when the water contains little magnesium, when the water contains appreciable quantities of magnesia, sodium aluminate should always be used as an additional reagent. The magnesium hydroxide precipitate normally takes a considerable time to separate and settle, since it tends to remain in the colloidal state. Sodium aluminate hastens the softening reaction by coagulating the magnesium hydroxide into a large floc which settles rapidly, and thus expedites the removal of magnesia. The floc also entrains particles of calcium carbonate or other matter in suspension as it settles. In this way a stable softened water of low hardness can be obtained in a shorter time than with lime and sodium carbonate alone. The amount of sodium aluminate required is small, about 1·0 part per 100,000 in hot and 2·0 parts in cold softening. A reduction in the silica content may

also be obtained when aluminate is employed as an auxiliary reagent and the necessary slight excess of softening reagents is used.

For very soft waters the lime-soda process of softening is not suitable. Phosphates can be substituted for lime-soda in the treatment of these waters, and this process is widely used on the Continent, particularly for high pressure boiler feed. It is carried out at an elevated temperature.

THE ZEOLITE PROCESS

In this process the water is passed through a bed of mineral which contains sodium in combination and has the property of exchanging its sodium for the calcium and magnesium in the water. The effluent therefore contains only sodium salts and is soft. The calcium and magnesium are retained by the mineral and in due course its capacity for the exchange becomes exhausted. The flow of water is then stopped and the mineral reconverted to its original form by treatment with a relatively strong solution of common salt, the calcium and magnesium passing into the brine and being rejected with it. The cycle of operations is therefore (1) softening, (2) regeneration with brine, (3) draining and washing to remove brine, after which the cycle (1)→(2)→(3) is repeated. Artificial materials having the same properties as the mineral formerly used have now largely replaced natural zeolite.

The process is intermittent in character, but of course can be made to give a continuous flow of softened water by using two or more units alternately. It is carried out in the cold, there is no sludge, and the hardness is reduced to about 0·5 part $CaCO_3$ per 100,000, or less under favourable conditions. Since the calcium salts are replaced by sodium salts it follows that the temporary hardness is converted into sodium bicarbonate which decomposes and gives off CO_2 when the water is heated. Such a water may cause corrosion in the boiler and auxiliary equipment, and the process is not recommended for waters of high temporary hardness which are to be used as boiler feed.

The water supply must be clean, as the bed of exchange material is adversely affected by the presence of suspended matter or soluble substances which may deposit from solution in the bed, such as iron and alumina. When installing a softening plant of this type, therefore, careful consideration should be given to the quantities of these materials which may be present in the raw water, particularly in waters which are subject to variations in composition. Preliminary coagulation, sedimentation and/or filtration may be required to protect the zeolite bed.

DEMINERALISATION PROCESS

This has been made possible by two further discoveries, firstly that some artificial base-exchange materials, prepared from coal and other highly carbonaceous substances will convert all calcium, magnesium and sodium salts into the corresponding acids when regenerated with acid instead of common salt. Secondly, that an entirely different group of substances, of the artificial resin type, have the power of absorbing dilute acids. These substances are regenerated with alkalies such as caustic soda and sodium carbonate.

Thus, by passing a water first through a bed of hydrogen exchange material and then through the acid removing material all salts are removed. In practice it is found that the treated water contains silica, corresponding to that present in the raw water, together with carbon dioxide, oxygen and dissolved salts to the extent of only a very few parts per 100,000.

With the exception of its silica content, water prepared in this way is comparable with that obtained from evaporators. The silica content is not likely to be objectionable in the boiler, provided that conditioning chemicals are added to prevent the formation of silicate scales.

DISTILLATION

This method yields a very pure water, but it is fairly expensive and is used only when the volume of make-up water is small or the boiler pressure is high. The water derived from steam traps when uncontaminated with oil, etc., is virtually distilled water.

CONDITIONING TREATMENT

It must be emphasised that in addition to preliminary treatment it is usually necessary to apply chemicals to the final feed water in order to provide the right balance of chemicals in the boiler water to prevent scaling, corrosion, and caustic embrittlement (see later paragraph), or to modify the form of any precipitate in the boiler so that it can easily be removed in the blow-down. Such chemical adjustments are known as conditioning treatment, and their application will depend on the composition of the final feed water and the boiler operating conditions.

INTERNAL TREATMENT ; WATER SOFTENING IN THE BOILER

Circumstances sometimes arise in which the installation of a softening plant is impracticable. An alternative treatment is available in the form of the addition of chemicals direct to the water entering the boiler, by which means the whole of the hardness is precipitated in the boiler as a sludge instead of a scale.

This method has been applied with very satisfactory results to waters up to 25 parts $CaCO_3$ per 100,000 in boilers working at moderate pressures and rating. In general, the treatment is recommended for water-tube boilers only when the hardness of the feed water is low, but in boilers of the Lancashire type water of higher hardness may be treated by this method.

The basis of this treatment is the addition of sodium carbonate or phosphate to the water as it enters the boiler, in an amount sufficient to precipitate the permanent hardness. The temporary hardness is, of course, precipitated by heat. The precipitates are in a finely divided form and may tend to agglomerate and bake on to the heating surfaces. Where they collect in a mud drum it has been observed that they pack, and only a small amount of mud in the immediate vicinity of the blow-down cock passes out when this is opened.

An improvement in these conditions can be effected by the addition, with the sodium carbonate, of tannins or mixtures of sodium aluminate and tannins, by means of which the precipitate appears to be coagulated into a free flowing form which is maintained in suspension by the circulation of the boiler water and can be easily removed by intermittent or continuous blow-down. For waters of low hardness sodium phosphate is preferred as a precipitating agent, and the coagulation of the precipitate is effected by the use of tannins mixed with the phosphate. Similar principles may be used to prevent precipitation in the economisers.

HIGH PRESSURE BOILERS

The feed water for modern high pressure boiler installations is usually a mixture of condensate and distilled water as make-up. Most waters will form scale in the evaporator and it is advisable to consider a preliminary softening treatment to maintain the evaporator heating surfaces in a clean condition.

The distillate should, of course, be free from salts, but in practice small amounts of carry-over or priming often occur, and contamination of the feed water can also arise from condenser leakage. Conditioning chemicals will be required to counteract the contamination, but should not be regarded as a substitute for mechanical attention to condensers and mechanical repairs

generally. With phosphate conditioning agents it is an advantage to maintain a little free phosphate in the boiler water. These points are highly important.

CORROSION

All boiler feed waters should be alkaline in order to minimise corrosion, and where the feed consists of distillate and condensate it will be necessary to add a little caustic soda to make the feed water alkaline to phenol-phthalein. The small quantity of caustic soda normally required for this purpose should be added continuously. It should be emphasised, however, that alkalinity alone cannot be relied upon to give adequate protection against corrosion in boilers working at pressures of the order of 250–300 lb. per square inch or above, and the oxygen and carbon dioxide content of the feed water must be maintained at a low value. The suggested limits for the dissolved oxygen content of the feed water are :—

Boilers working at 300–450 lb. per square inch, not more than 0·02 cubic centimetre per litre. At higher pressures every effort should be made to reduce this below 0·01 cubic centimetre per litre.

DE-AERATION

Mechanical de-aeration is generally effected either by passing the feed water through special equipment installed for this purpose, or by de-aeration in the turbine condenser to which the distillate from the evaporator is introduced and mixes with the condensate.

In practice the residual oxygen content may exceed the recommended figure, e.g. when the de-aerator is working at peak loads, or from other causes of a temporary nature, and the additional oxygen then present in the feed water may be removed by chemical de-aeration with sodium sulphite. The sodium sulphite should be added continuously in the form of an aqueous solution at a point beyond which atmospheric contamination cannot occur. When caustic soda is added to the feed water the two reagents can be fed together and it is advisable to have the chemical solution tank covered with a close-fitting lid to minimise oxidation by contact with the atmosphere. The quantity of sulphite required is based on the oxygen content of the feed water with an excess sufficient to give a reserve of 3–5 parts Na_2SO_3 per 100,000 in the boiler water.

CAUSTIC EMBRITTLEMENT

The cracking of boiler metal as a result of attack by caustic soda has been the subject of extensive study on the laboratory and plant scale. These cracks can be identified under the microscope because they are intercrystalline and show no marked deformation of the grain, whereas those due to corrosion or fatigue failures are transcrystalline and are associated with grain deformation.

The investigations have shown that at least three factors must coincide in order to produce the characteristic intercrystalline attack.

(1) Unequal stress distribution in the metal, with high local stresses such as might occur in riveted joints due to misalignment of rivet holes, deformation of rivets, etc.

(2) The occurrence of crevices in the neighbourhood of the highly stressed metal, e.g. overlap joints, etc.

(3) The leakage of boiler water containing caustic soda into the crevices and its concentration therein in contact with the stressed metal. It has been demonstrated that such concentration can occur under boiler conditions especially when leakage to the outside occurs. Small leaks which are invisible but can be detected by condensation on a cool surface appear to be able to provide the necessary concentration conditions.

The attack usually takes the form of irregular crazy cracking on the dry side

of the plate starting from and running between rivet holes. Although the rest of the metal is quite sound the development of these cracks obviously leads to serious weakness which can result in sudden failure at the joint.

It is clear that both mechanical and chemical factors are involved and that the mechanical factors are particularly difficult to detect in a working boiler. Since leakages provide a means of concentration of the boiler water, however, it follows that the detection of small leaks is an important part of boiler maintenance and that any boiler with a history of leakage is suspect and should be carefully examined by an expert metallurgist. It is recommended that joints should be caulked internally and that welded drums should be stress relieved. There appears to be little evidence of embrittlement in operating boilers with forged drums.

Certain substances, such as lignins, tannins, sodium sulphate, etc., have been shown to have a retarding effect and up to the present it has been common practice in the United Kingdom to maintain definite concentrations of sodium sulphate in the boiler water as a safeguard. There are wide differences of opinion regarding the effectiveness of this chemical but at the time of writing no alternative material has been definitely established by large-scale experience as superior. It seems reasonable therefore to continue to use sodium sulphate for the present, and sufficient of this substance should be added to maintain in the boiler water at all times a weight ratio sodium sulphate/caustic soda (Na_2SO_4/NaOH) greater than 2·5, when riveted drums or welded drums which have not been stress relieved are used. Many waters contain sufficient sodium sulphate for this purpose, but other waters require a special addition of this chemical. It must be emphasised that no chemical additions to the boiler water can heal cracks which have once started because even if embrittlement factors have been removed from the water, the cracks act as stress raisers and failure may occur as a result of corrosion fatigue.

It is clear that the present position regarding this subject is not very satisfactory from the point of view of the engineer in charge of boilers, and it should be pointed out that boiler failures as a result of caustic embrittlement are not common, but that this does not justify neglect of the precautions which appear to be effective in the present state of knowledge. Further developments both in laboratory research and practical operating experience will no doubt be published from time to time and should be carefully studied by those responsible for boiler plant.

BLOW-DOWN

A boiler is blown down to prevent the unrestricted accumulation of suspended and dissolved solids which would lead to contamination of steam with boiler water, commonly called priming, foaming, or carry-over, and ultimately to the crystallisation of the very soluble sodium salts. It is an essential feature of all boiler operation whatever treatment is applied to the feed water.

AMOUNT OF BLOW-DOWN

The amount of blow-down should be controlled by the amount of total dissolved solids (hereafter referred to as T.D.S.) in the boiler water, which should be consistently maintained slightly below the maximum permissible amount. This maximum figure requires determination under normal operating conditions for each boiler plant and make of boiler. It is affected by such factors as the amount of suspended matter in the boiler water, the variability of the boiler load, and the level of the water in the drum. The highest figures will be obtained when :—

(1) The boiler load is steady.

(2) The boiler water is clean, i.e. free from all but small amounts of finely divided suspended matter.
(3) The water level is just above the centre line of the drum and not higher than, say, two-thirds the diameter of the drum.

The determination of the maximum permissible T.D.S. depends on observations of steam purity. When large quantities of boiler water are present in the steam the superheat temperature falls suddenly. This phenomenon is often called priming. There appears to be another condition, however, in which small amounts of boiler water pass off with the steam more or less continuously, and this occurrence is often known by the term "carry-over." It is doubtful whether the carry-over of small amounts of boiler water will appreciably affect the superheat temperature, and the method of detection usually employed is the measurement of the electrical conductivity of steam samples, i.e. the steam purity.

For this purpose a sampling nozzle (bore about $\frac{5}{16}$ inch for 175 lb. pressure and 6-inch steam pipe) is fitted into the steam line pointing up-stream and on the dead centre of the pipe; sharp bends in the pipe are to be avoided. The nozzle is connected to a cooling coil of sufficient capacity to reduce the temperature of the sample to atmospheric. The flow of the condensate must be adjusted so that the linear velocity of steam in the sampling nozzle is the same as that in the steam line from which the sample is taken.

When the boiler is working on normal load the conductivity of several steam samples should be measured and then, with the blow-down valve shut, 24-hour average samples of steam condensate and boiler blow-down can be taken and the conductivity of the steam sample plotted against the concentration of dissolved solids in the blow-down. That concentration at which a sharp increase in conductivity of the steam sample occurs may be regarded as the maximum permissible total dissolved solids content in the boiler water for the particular type of boiler and operating conditions. The effect on the maximum permissible T.D.S. of the other factors already mentioned may be observed by a similar set of tests. In practice it is found that certain maximum figures are fairly generally applicable and these are given later in Table 68. The actual T.D.S. must be maintained below the maximum (cf. p. 245).

If the maximum permissible T.D.S. in the boiler water is A parts per 100,000 and the T.D.S. in the feed water entering the boiler is B parts per 100,000, then the theoretical blow-down will be $\frac{100B}{A}$ per cent. of the total evaporation.

MEASUREMENT OF T.D.S.

There are four methods in use :—

(1) By hydrometer. These can be obtained from suppliers of laboratory apparatus, but they should be checked against actual determinations of T.D.S. by evaporation to dryness and weighing. When ordering, the purpose for which the hydrometer is required should be stated.
(2) By the conductivity of the blow-down water. This figure can be used alone after calibration against actual determinations. The method permits of a continuous record being kept when continuous blow-down is installed.
(3) By determination of chloride in both feed and boiler water. Chlorides are very soluble and are not eliminated as scale and, therefore, this chemical test is a fair indication of concentration when the composition of the feed water is fairly constant, but it is not as reliable or as easy to carry out as methods 1 and 2.
(4) Evaporation to dryness and weighing. This is a standard laboratory method rather tedious and unnecessary for routine plant control.

BLOW-DOWN TECHNIQUE

Continuous blow-down is the most satisfactory method of removing water from the boiler because it can easily be adjusted to maintain the T.D.S. in the boiler water consistently at the desired level, and takes the matter out of the hands of the operator. The method preferred is to fit a short horizontal length of pipe below the low-water level in the steam drum and at the end opposite to the feed entry. In cylindrical boilers an open-ended vertical pipe may be used and the open end should be more than 2 inches above the top level of the flue tubes so that the boiler cannot blow itself dry. A suitable size of orifice, or a special fine control valve, is fitted in the blow-down line outside the boiler so that the amount can be controlled. If the boiler water contains appreciable amounts of suspended matter the ordinary blow-down valve should be opened momentarily once per day to remove accumulated solids. Continuous blow-down is not recommended when the volume of water to be removed is less than 25 gallons per hour.

When the blow-down is intermittent and effected by means of the usual valve fitted at the lowest point of the boiler, this valve should be used as frequently as is practicable, taking into account the total volume of water to be removed. If possible, the valve should be opened once per shift and usually not less than once per day. In this way the composition of the boiler water will be kept as constant as possible. To ensure the maximum removal of accumulated sludge, the valve should be opened momentarily several times with a short pause in between so that the disturbance in the neighbourhood of the blow-down outlet can die down and fresh sludge can move into position ready to be flushed out. When starting up, boilers should not be blown down until the T.D.S. reaches the proper value, and it will be found advisable to have a clear set of instructions issued to each boiler house covering all conditions of operation.

DISPOSAL OF BLOW-DOWN

There are four methods of disposal in common use, viz. :—

(1) To waste : if this method be adopted the most rigid control is necessary to economise in fuel.
(2) Through a heat exchanger, in which the feed water is heated, or alternatively through a flash vessel from which the steam is passed direct into the feed water and the heat in the residue is recovered by a heat exchanger. A further economy of fuel can be secured by reducing this residue to a minimum.
(3) Through a flash vessel and heat exchanger as before, but part of the residue from the flash vessel is returned to the softening plant where its heat and chemical content are used.
(4) There is a fourth method which can be applied to boilers in which the feed water is condensate and evaporated make-up, namely, by returning a portion of the blow-down direct to the feed so that the chemicals in it are used for conditioning purposes.

While the importance of heat recovery by exchange is obvious, it is not always practicable with existing plant layout, particularly at the present time when economy of steel is also required. The installation of heat recovery plant may have to be decided on the basis of steel economy versus potential fuel economy, and careful consideration will be required in each case, but there is no doubt that much can be done to limit heat losses when the blow-down is sent to waste, and the data collected with this object in view will also be necessary for a consideration of heat economy by the use of exchangers.

CONTROL OF WATER TREATMENT

It should be emphasised that satisfactory softening and conditioning can be attained only by careful control, and routine testing is therefore essential. Simple and rapid methods of analysis have been developed for control purposes. For testing purposes a clean and well-lighted place should be provided since many of the erratic results in softening plants and with boiler feed water conditioning may be traced to errors in testing due to dirty testing apparatus, bad light, etc.

Except where otherwise stated, all tests should be carried out on the sample after filtration through filter paper.

A summary of the necessary tests is given below and for convenience these have been grouped under two main headings :—

(1) Water softening.
(2) The control of blow-down and boiler feed water conditioning.

(1) The determinations necessary for the control of a lime-soda water softener are as follows :—

(*a*) *Raw Water*

Temporary hardness.
Free CO_2 (not essential, as an extra quantity of lime can be added to combine with CO_2).
Total hardness.
Hardness due to lime salts.
Hardness due to magnesia salts (found by difference).

(*b*) *Softened Water*

Total alkalinity to methyl orange indicator.
Caustic alkalinity.
Residual hardness due to lime.
Residual hardness due to magnesia.
Hardness in suspension (sometimes necessary to check the efficiency of filtration in lime-soda plants).

Raw Water

These determinations should be carried out once a day and the quantities of reagents added should be calculated from the analysis of the raw water. If it is found that the composition of the raw water does not vary appreciably from day to day, then less frequent analyses may suffice.

Softened Water

These determinations are required for the control of the softening plant and should be carried out twice per day unless the raw water does not vary appreciably in composition and more often if the water is variable.

The sample of softened water should be taken from the softening plant outflow pipe some four hours after any change in the addition of reagents, in order to allow sufficient time for the change to become effective.

Zeolite softening plants are usually operated on a volume or time basis. The plant consists of several units each containing sufficient material to soften a definite volume of water of given composition. Provided that this composition remains constant it is then only necessary to carry out determinations of alkalinity and total hardness on the raw water once per day to establish that the composition is up to specification, or, if the composition varies, to alter the

volume passing through the unit before regeneration. The softened water should be tested in the same way several times immediately before it is expected that the unit will become exhausted, and as soon as the residual hardness in the softened water rises to about 1·0 part $CaCO_3$ per 100,000 the unit should be regenerated.

(2) The determinations necessary for the control of boiler feed water conditioning and boiler blow-down are as follows :—

(*a*) *Feed Water*

(Once per day or more often if required. Feed water is often the softened water already tested.)
Caustic alkalinity.
Chloride.
Sulphate.
Oxygen (for high pressure boilers a recorder is advised).
Hardness.

(*b*) *Boiler Blow-Down*

Total alkalinity.
Caustic alkalinity.
Chloride.
Sulphate.
Phosphate (where phosphate conditioning is used).
Hardness. (This is a detection test and not an estimation.) The detection of hardness indicates the possibility of scale formation and extra chemical treatment should be applied at once.
Sulphite (where sulphite is used for chemical de-aeration).
Total dissolved solids.

From these determinations it will be possible to ascertain without difficulty whether the composition of a boiler water satisfies the conditions laid down for the prevention of corrosion, caustic embrittlement and scale formation. The suggested tests may at first sight seem somewhat formidable, but for simple types of boiler a simpler scheme of tests can be worked out ; similarly for large boiler plants the number of tests can be reduced to a minimum by careful planning.

It will be appreciated that the characteristics of the feed and boiler water will require to be maintained at values which experience has shown to be necessary for the working pressure and type of each particular boiler installation. In general terms the specifications for the feed, softened make-up and boiler waters in low, medium and high pressure plants are as given in Table 68.

TABLE 68. SPECIFICATIONS FOR BOILER FEED WATER AND WATER IN BOILERS

	Softened water	Feed water
Total alkalinity ..	Approx. 8 pts. $CaCO_3$/100,000	Alkaline to phenol-phthalein at all times.
Caustic alkalinity..	,, 3 pts. $CaCO_3$/100,000	
Hardness	1 part $CaCO_3$/100,000 as near as possible.	As low as possible.
Oxygen	—	*Medium pressure* not exceeding 0·02 c.c. per litre. *High pressures* not exceeding 0·01 c.c. per litre.

	Boiler waters		
	Low pressure up to 250 lb./sq. in.	Medium pressure 250–500 lb./sq. in.	High pressure above 500 lb./sq. in.
Total alkalinity ..	15–20% of T.D.S.	15–20% of T.D.S.	15–20% of T.D.S.
Hardness (Wanklyn foam test).. ..	Zero.	Zero.	Zero.
Sulphate	Not below 2½ times caustic alkalinity.	Not below 2½ times caustic alkalinity.	Not below 2½ times caustic alkalinity.
Phosphate.. ..	8½–17 parts per 100,000 as Na_3PO_4.	8½–17 parts per 100,000 as Na_3PO_4.	8½–17 parts per 100,000 as Na_3PO_4.
Total dissolved solids (T.D.S.) for blow-down control.	Up to 700 parts/ 100,000.	500–200 parts per 100,000.	Not exceeding 100 parts/100,000.

(*N.B.*—For Economic boilers maximum T.D.S. recommended is 400 parts/100,000.)

ROUTINE TESTING OF WATER SAMPLES
RECOMMENDED PROCEDURE FOR SAMPLING

It will be realised that the method adopted in collecting samples of water for testing will have an important bearing on the control of water softening and of feed and boiler conditioning since the adjustment of the treatment is based on the results so obtained. No fixed system of sampling can be laid down to cover all boiler plants but the following notes, which are intended as a general guide, may be of use in developing a proper scheme of sampling.

The samples should, if possible, be obtained from permanent sampling points and these should be so arranged that there is no doubt about the samples being representative.

RAW WATER

More often than not raw water supply is subject to fluctuations in composition, these variations being dependent on a number of factors Daily average samples are therefore only useful for control purposes when the variations are small. Variable waters must be sampled as frequently as required for the control of plant. It is important to take samples at all times of the year so as to build up a history of the water composition.

SOFTENED WATER

Spot samples are required for plant control and daily average samples for checking and record purposes. The main precaution is that the sample should be taken at a convenient point on the softened water outlet line as near to the softening plant as possible.

FEED WATER

Where the feed water consists of condensate and make-up water mixed in a feed tank or hot well, it is inadvisable to assume that a sample taken from the feed tank is entirely representative. A sample taken from the boiler feed pump is therefore to be preferred.

CONDENSATE

Sampling of condensate presents rather more difficulty as although in general there are numerous points at which samples can be obtained, it is necessary to sample at a point where the various condensate returns mix together before re-entering the feed systems. If contamination occurs each source of condensate should be sampled.

BOILER WATER

Boiler water samples are usually taken from the gauge glass. This is done by shutting the steam cock on the top of the gauge glass, opening the gauge glass drain, and allowing the water to run to waste until the connecting pipe between the bottom of the gauge glass and the boiler is flushed through before the sample is taken. This is very important, as the continual condensation of steam in the gauge glass from the upper steam connection results in the water lying in the connecting pipe to the boiler being mixed with condensate.

In boilers fitted with continuous blow-down, the discharge to waste pipe provides a convenient sampling point. A sample taken from either point has the disadvantage of being subject to steam " flash-off " unless suitable precautions are adopted. This " flash-off " can be successfully obviated if the sample is passed through a cooling coil, and this method of sampling is advised.

In boilers receiving individual treatment it is advisable to test each boiler separately. Where there are a number of boilers utilising the same feed water supply it is suggested that they should be divided into groups of four, one boiler from each group being tested in rotation each day. This ensures that each boiler water is tested at least once every four days.

It is preferable to collect the boiler water samples in a copper cylinder or a welded sheet-iron bottle fitted with a close-fitting lid or stopper. The samples should be stoppered immediately, cooled as quickly as possible, and then transferred to a bottle and immediately stoppered. By this means the samples can be kept and stored without undergoing change until it is convenient to carry out the tests.

WORKS CONTROL

From this discussion it may appear that the treatment of water and its control is a somewhat complicated matter, but this is due to the attempt to cover the essential features of the process as applied to a variety of boiler plants. After a study of any individual boiler plant the picture which is revealed is usually fairly simple. Boiler plants have an individuality in design, layout, operation and staff and a study of these points in each plant is therefore essential so that a scheme of water treatment can be devised to fit the existing conditions. Long experience has shown that failure to devise such a scheme is rare provided that there is first a proper understanding of the special conditions at the plant concerned and that all changes in conditions which can be foreseen are taken into account.

A study of water conditions at a boiler plant should be carried out in three stages.

(1) PREPARATION

An appreciation should be made of the nature of the water supply, e.g. source, composition, variability, contaminations, etc., and of the boiler plant, e.g. size, type of boiler, working pressure, layout, etc.

(2) DESIGN

A scheme of water treatment and control testing should be devised, based on the appreciation of the factors in (1).

(3) OPERATION

The whole scheme should be discussed with the staff concerned and modified where possible to meet special objections or requirements. At least two members of the staff should be instructed in the basic principles of the treatment and the tests required, and when the scheme is put into operation various check tests will be necessary before it can be established on a routine basis. Minor modifications in treatment may have to be made in the light of operational experience.

It is highly desirable that some member of the staff should be made responsible for the maintenance of the water conditions laid down in the scheme and for the testing required as well as for the study of the principles involved. He should act in an advisory capacity to the manager of the boilers and should work in close co-operation with him. An arrangement of this kind is particularly necessary when the boiler plant is large or a number of boiler plants comes under one control.

It is clear that the development of such a scheme requires a thorough knowledge of the principles of water treatment at all stages of its progress from source to boiler, coupled with judgment and experience.

NEED FOR EXPERT ADVICE

The account here given of boiler feed water treatment is necessarily sketchy and incomplete. Reference is made particularly to the purification of feed water for low and moderate pressures ; for high pressure boilers distilled water free from dissolved gases must be used.

Sufficient will have been said to indicate that the treatment of boiler feed water is primarily a matter for a chemist and that the successful operation of a softening plant depends on chemical control, just as the installation of the right type of plant to meet local circumstances requires expert advice.

This advice is best obtained by employing a chemist. When that is impossible, a consultant may be able to help materially by paying periodic visits and making the necessary tests or teaching the boiler staff to carry out simple control tests. Many small boiler plants are controlled in this way after preliminary consultation with an expert. If no advice can be obtained, the technical staff of the Ministry of Fuel and Power are always ready to assist enquirers.


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CHAPTER XVI

BOILER EFFICIENCY AND TESTING

The methods of testing steam boilers to secure essential information regarding operating results—Examples of operating results for efficient and inefficient plants—The more detailed examination of boiler plants.

THE object of those in charge of boiler plant should be to realise the highest possible efficiency with the plant and fuel at their disposal. It follows that to know whether this object has been achieved, it is necessary to know what is possible with the particular plant. There is no other way of determining what is possible than by testing the performance of the plant.

If a proper standard of performance is established for a particular boiler, any falling off in performance can be detected and the cause traced.

The procedure in boiler testing will depend upon the extent of the information that is required and also, incidentally, upon the size and skill of the staff available for the work and the extent to which they are provided with instruments. There may well be a difference in procedure between routine testing undertaken to keep the plant in good order and a more comprehensive test undertaken at much longer intervals to ascertain a complete heat balance. Here the simple procedure is dealt with on the basis of British Standard Code 845. This code is not intended for use in the comprehensive tests, such as would be needed for large boilers, but it is accepted here as being a useful guide for routine testing for the majority of works boiler installations. Comprehensive tests can be performed on the basis of Table 73, or by using the code drawn up by the Institution of Civil Engineers.

The object here is to indicate the methods which should be adopted and the data which it is desirable to secure when carrying out a simple efficiency test at minimum cost on steam-raising plants using solid fuel, to obtain a satisfactory measure of performance. A warning must be given to the effect that if the load on the boiler fluctuates, the period over which the test is run must be sufficient to embrace a complete cycle of such fluctuations. In B.S. 845 the readings taken and the calculations made are reduced to the minimum. As will be seen from the test results given later in this chapter, this method determines the overall thermal efficiency, and the losses in flue gases and ashes. All other losses are taken as a difference figure. The results, moreover, are referred to the net C.V. of the fuel. The Institution of Civil Engineers code is based on gross C.V., and the gross C.V. is generally to be preferred.

Some specialised knowledge is required to ensure that reliable results are obtained by testing. Such knowledge can only be acquired by practice, but practice is often difficult to obtain, and therefore it is advisable to secure assistance for at least the first test from an engineer who has had practical experience in boiler testing, and who will advise as to details of methods, instruments, and how to avoid inaccuracies of measurement. Such assistance is readily available and can be obtained upon application to the Ministry of Fuel and Power.

Before making a test to set up a standard of performance, it is essential that the boiler and its auxiliaries shall be in such condition that all avoidable losses are eliminated. This means that all heating surfaces, both internal and external, must be clean ; that no leaks of water or steam occur ; that no air leaks occur through the setting of the boiler or the flues ; that the grate is in good condition, and that the air supply is effective.

The fuel used for a performance test should be representative of that which will normally be used.

The thermal efficiency of a boiler may be determined in two ways by :—

(*a*) Direct measurement, and

(*b*) Indirect measurement.

The direct method should give better and more positive results and it necessitates the measurement of the fuel burned and of the water evaporated. While this method is recommended it is recognised that in some works it may not be possible to make suitable arrangements to determine the weight of fuel and water. In such cases it is necessary to make use of the indirect method, which consists of ascertaining the various losses, and after expressing these as percentages, deducting their sum from 100 ; the resultant then gives the thermal efficiency of the boiler plant.

An example (Table 69) will be given set out in the manner recommended by B.S. 845.

TABLE 69. EXAMPLE 1

B.S. Item 7.	Boiler water tube heating surface	sq. ft.	Not required for this calculation.	
,, 8.	Superheater integral heating surface ..	sq. ft.	Ditto.	
,, 9.	Economiser heating surface	sq. ft.	Ditto.	
,, 10.	Air heater (not installed).			
,, 11.	Method of firing		Travelling grate.	
,, 12.	Grate area	sq. ft.	140	
,, 13.	Draught plant (chimney draught only) ..		Not required for this calculation.	
,, 14.	Duration of test	hrs.	8	
	A. Fuel			
,, 15.	Description		Northumberland slack.	
,, 16.	Proximate analysis as weighed :—			
	Moisture	per cent.		12·4
	Volatile matter	,, ,,		28·5
	Fixed carbon	,, ,,		47·7
	Ash	,, ,,		11·4
		,, ,,		100·0
,, 17.	Gross or higher calorific value as weighed ..	B.Th.U./lb.		11,000
,, 18.	Net or lower calorific value as weighed ..	B.Th.U./lb.		10,478
,, 19.	Fuel fired per hour	lb.		2,754
,, 20.	Fuel fired per hour per sq. ft. of grate area	lb.		19·7
	B. Ashes and Clinkers			
,, 21.	Collected per hour	lb.		342
,, 22.	Percentage of fuel fired			12·4
,, 23.	Percentage of combustible			8
	C. Water			
,, 24.	Temperature entering economiser	° F.		142
,, 25.	Temperature entering boiler	Not required as economiser fitted.		
,, 26.	Evaporation per hour	lb.		20,152
,, 27.	Evaporation per lb. of fuel as weighed ..	lb.		7,318
	D. Steam			
,, 28.	Gauge pressure..	lb./sq. in.		195
,, 29.	Final temperature	° F.		530
,, 30.	Heat per lb. above feed temperature ..	B.Th.U.		1,178

E. Flue Gases

Item 31.	Temperature leaving economiser	° F.	400
,, 32.	Analysis leaving economiser :—		
	CO_2..	per cent.	10
	O_2	,, ,,	9·3
	CO	,, ,,	Nil

F. Draught and G. Air

Items 33 to 38 inclusive	Not required for this calculation.	
Item 39. Temperature of air supply to furnace ..	° F.	60

RESULTS

,, 40.	Overall thermal efficiency on gross calorific value of the fuel	per cent.	78·4

A. Heat Account Calculated on the Net Calorific Value of the Fuel

Item 41.	Overall thermal efficiency	per cent.	82·3
,, 42.	Loss due to sensible heat in chimney gases..	,, ,,	11·9
,, 43.	Loss due to unburnt CO	,, ,,	Nil
,, 44.	Loss due to combustible matter in ashes and clinkers	,, ,,	1·4
,, 45.	Balance :—radiation and other unmeasured losses including grit emission	,, ,,	4·4
		,, ,,	100·0

B. Net Overall Thermal Efficiency

Not required for this calculation

CALORIFIC VALUE OF THE COAL

In the B.S. code, the net C.V. of the coal is taken as the basis. The net C.V. may be termed the " available C.V.," since it ignores the latent heat of the water produced on combustion. Because the net C.V. of the coal is the basis of the method here described, the latent heat of the water in the flue gases is is omitted from the calculations.

LOSSES

These are as shown by :—

(A)	Item 42.	Loss due to sensible heat in chimney gases ..	per cent.	11·9
(B)	,, 43.	Loss due to unburnt CO	,, ,,	0
(C)	,, 44.	Loss due to combustible matter in ashes and clinkers	,, ,,	1·4
(D)	,, 45.	Balance :—radiation and other unmeasured losses including grit emission	,, ,,	4·4

The sum of these four items, i.e. 17·7 per cent. represents the total losses and therefore 100 − 17·7 = 82·3 per cent. represents the overall efficiency of the plant on the net calorific value obtained by the indirect method.

Item 42. If only the theoretical weight of air required for the combustion of the fuel were supplied the chimney losses would be a minimum. This minimum, in the example set out above, would have been 6·4 per cent. ; the actual loss with 10 per cent. CO_2 was 11·9 per cent. Also, if only the theoretical weight of air is supplied the temperature of the gases at the chimney will be reduced. In practice it is impossible to operate with the theoretical weight of air as has been explained in Chapter VI.

The latent heat in the water vapour in the flue gases is ignored, as has just been explained. The sensible heat of the gases depends not only on their temperature but also on their composition, since this affects their specific heat

(Chapter V). As a simplified, though approximate method of calculation, in the B.S. code a method devised by Siegert is used :—

$$\text{Sensible heat in flue gases} = \frac{K\,(t_2 - t_1)}{\text{per cent. } CO_2 \text{ in flue gas.}}$$

where K = 0·35 for bituminous coal
0·37 for anthracite
0·39 for coke.

t_1 = temperature of air supply to furnace, °F.
t_2 = temperature of gases leaving boiler, economiser or air heater.

It has lately been found that whilst the maximum error in using this simplified formula is ± 1 per cent. for temperatures of the order of 500° F. commonly met with at economiser outlets, at higher temperatures of the order of 1,000° F. which are by no means unknown, the error rises to ± 3 per cent. or so. It is therefore recommended wherever possible that the calculation should be made from first principles as set out in Chapter V.

In this example the chimney loss is the largest of the four losses, and this is almost always so in practice. It will be seen that the magnitude of the chimney loss is directly dependent upon the temperature of the leaving flue gases, and on the percentage of CO_2; a reduction in the gas temperature or an increase in the CO_2 produces a smaller loss.

A low leaving gas temperature can be obtained by :—

(*a*) Installing in the first place ample heating surface of boiler and/or economiser, but in determining the size of these sections of the plant consideration has also to be given to the appreciable increase in capital cost that arises when a plant is designed for a very low leaving gas temperature.
(*b*) Maintaining the heating surfaces both externally and internally in a clean condition.

With plants that are in good condition and efficiently operated, losses under the headings of Items 43, 44 and 45 of the following approximate order should be expected :—

Item 43.	Loss due to unburnt CO ..	Nil.
,, 44.	Loss due to combustible matter in ashes and clinkers	Not exceeding about 2 per cent. unless the fuel burnt has abnormal characteristics such as high percentage of fines, non-caking qualities, etc., when this loss will be appreciably greater.
,, 45.	Balance—radiation and other unmeasured losses	Between 2 and 6 per cent. Where conditions favour grit emission (see Chapter XI) this figure may be very much higher.

BOILER TEST MEASUREMENTS

In what follows it is assumed that the object of the test is to ascertain what is the best performance that can be expected from a particular boiler plant of small or medium size in use in a factory, or alternatively to ascertain whether any improvement or deterioration has occurred since a previous test.

DURATION OF TEST

In considering the duration of the test, it is understood that the rate of evaporation is maintained practically constant throughout the period of the test and, once the combustion conditions are set for the kind of fuel being used, it should only be necessary to make minor alterations during the test. The load should, for at least 24 hours before the test commences, be as near as possible to the load carried during the test. Where these requirements cannot be fulfilled, a statement to this effect should be included in the report of the trial and it must be recognised that the recorded efficiency of the plant has been affected accordingly.

The trial should continue until the total fuel consumption per square foot of grate surface is equivalent to not less than 25 lb. for every 1 inch of fire thickness, with a minimum duration of four hours and this period is generally sufficient for the measurement of the other data with the necessary degree of accuracy. Thus, with a 6-inch fire, the trial should continue until $6 \times 25 = 150$ lb. of fuel have been consumed per square foot of grate surface. At a rate of firing of 30 lb. per square foot, the duration would be at least $150/30 = 5$ hours.

The plant should be run under test conditions for not less than one hour before and one hour after the trial, during which time sufficient data are to be collected to demonstrate that similar and steady conditions prevailed at the commencement and end of the trial. A four-hour trial plus these two control hours is less liable to error than a six-hour trial without the control periods. The fire should be as far as possible in the same condition (thickness, etc.) at the end of the trial as at the start. With hand fired furnaces, the trial should cover one or more complete cycles from clean fires to clean fires.

With pulverised fuel, direct-fired on the unit system, four hours with two control hours are usually sufficient. Where the pulverised fuel is not fired direct from the pulveriser, either in the unit or central system, a longer period is usually necessary for a correct determination of the fuel consumption. Such duration can only be determined by an examination of the plant and the possible variation of the quantity of fuel contained in the system between the point of weighing the fuel and its delivery to the furnace. The test should be of sufficient length to reduce this variation of the quantity of coal contained in the system to a percentage of the total fuel consumed, consistent with the degree of accuracy aimed at, which should ordinarily be about 1 per cent.

COAL

The coal used for the test, in so far as circumstances permit, should be the same as that normally used. The amount used should be weighed as indicated in Chapter X.

The coal should be sampled in accordance with B.S. Specification 1017 (see Chapter XXXI) and the calorific value, together with the proximate analysis, determined by a competent chemist. When a staff chemist is not available the services of a firm specialising in this class of work should be sought.

The accuracy of the result of the test is entirely dependent upon the accuracy with which the calorific value of the fuel is determined. This depends upon two factors, (*a*) proper sampling of the fuel and (*b*) determination of the calorific value. Any inaccuracy occurring in either of these factors will be directly reflected in the final result. The important part which proper sampling of the fuel plays in the determination of boiler efficiency is not always realised, and it is therefore essential that the instructions for sampling laid down by B.S. 1017 be rigidly followed.

At the same time as the sample for the determination of calorific value is taken, a further sample of at least 20 lb. in weight should be prepared for the purpose of making a sizing analysis ; determination of sizing should be undertaken by the chemist. Although details of the coal sizing are not necessary to determine the efficiency at which the plant is operating, they provide information which is essential before any recommendation for improvement in the efficiency can be made. The sizing specification should be drawn up by reference to the B.S.I. standard sieve numbers.

WATER EVAPORATED

Methods of measuring water and steam by orifice plate, and water by V-notch, are described in Chapter X ; other forms of water meter are mentioned in Chapter XXX. All instruments should be calibrated before the test.

If the weight of steam produced is taken as the weight of water fed to the boiler, when measuring this water every care must be taken to see that no water is used for any other purpose, that there is no escape of water from the feed system, no leakage from pump glands or economiser safety valves, and that water is not blown down from the boiler during the test. If the boiler is fitted with a continuous blow-down system, allowance must be made accordingly, or the blow-down must be shut off for the period of the test.

In addition to the methods of measurement given in Chapters X and XXX, water may be weighed directly in tanks placed on weighbridges, this being the best method for small installations ; or the water may be measured in calibrated tanks, the volumes at the beginning and ending of the test being corrected for temperature.

The water level in the boiler should be the same at the finish as at the start of the test.

It is frequently necessary to determine the wetness of the steam, especially when there are no superheaters, or when superheaters are installed on only half the boilers.

TEMPERATURES

The temperatures required for the test are as follows :—

Water entering economiser.
Water entering boiler.
Flue gases leaving the end point of the system (i.e. leaving boiler, economiser or air heater).
Air supply to furnace.
Air supply to air heater.
Steam temperature.

The measurement of temperature and the precautions necessary to obtain true readings are discussed in Chapter XXX, and also under " Stratification," Chapter IX. If a superheater is fitted, care must be taken to see that the thermometer records the true temperature of the steam leaving the unit. The thermometer pocket should be placed in a straight length of pipe some 2–4 feet from the superheater stop valve.

When taking the temperature of flue gases, special precautions such as a suction pyrometer need not be used if the temperature is below about 750°–800° F. and if the flue is below ground level or is insulated. Nevertheless, adequate precautions must be taken against the errors mentioned in Chapter XXX if the thermometer can " see " any surfaces markedly cooler than those of the gas. Temperatures should not be taken near the economiser outlet, since they are likely to be vitiated by stratification. If it should be necessary to attempt to take temperatures at that position, special instruments must be

used wherever the thermometer or pyrometer can " see " the economiser tubes (cf. Chapter VIII, " Radiation "). The higher the velocity of the gases at the point of measurement, the more nearly is the measured temperature likely to be accurate.

FLUE GAS ANALYSIS

It is essential to measure the CO_2, O_2 and CO content of the flue gases. The standard method employs an Orsat apparatus. There are many automatic CO_2 indicators and recorders, but if they are used it is essential that they be calibrated against an Orsat apparatus immediately prior to the test.

It is advisable to stress again the importance of ensuring that the sample of gas withdrawn into the Orsat is a truly representative average sample. This subject is discussed in Chapters XXX and XXXII.

ASHES AND RIDDLINGS THROUGH GRATE

The ash should be set aside, allowed to cool and then weighed. A sample should then be taken, the same care being exercised as in taking the sample of fuel. The sample is then sent to the chemist for the determination of the combustible content.

The riddlings should always be re-fired ; but if it is not practicable to do so during the test the riddlings should be added to the ash prior to weighing and sampling.

DRAUGHT GAUGES

Draught gauges are necessary for proper control of operation, but draught readings do not come into efficiency calculations.

PRESSURE GAUGES

The saturated steam pressure is the pressure of the steam in the boiler drum ; superheated steam pressure is measured at the superheater outlet. Bourdon gauges are quite suitable for this measurement, but they should be calibrated against a dead weight pressure tester or against a calibrated master gauge. Any correction necessary should be noted and applied.

READINGS

It will be found that usually all the readings of the various instruments are reasonably steady when the boiler is tested at a constant rate of evaporation, and therefore readings need only be taken at say 15 or 20 minute intervals. It is convenient to have readings taken by individual observers on foolscap sheets and transferred after each reading to a log sheet on which all readings are entered. This enables a watch to be kept on the effects of any changes and also on unforeseen results. The only exceptions are coal consumption and water evaporated, as it is the total of these for the whole test which matters. It is difficult to segregate coal consumption into accurate hourly periods. At the end of the test, the various readings should be averaged, and corrections applied where necessary.

POWER FOR AUXILIARIES

Power taken for auxiliaries strictly speaking is a debit against boiler efficiency. It is the general practice not to bring this item into normal efficiency calculations. It is essential to take account of this power when it is desired to arrive at a true value of overall output and efficiency, or when comparing different types of plant. For operating purposes, it will be sufficient to set up a standard

of performance for the auxiliaries and then to make a check test at reasonable intervals to find out whether the performance is deteriorating.

The consumption of steam by boiler auxiliaries should be measured with some accuracy. In particular the high consumption of steam jets is often a serious source of loss, and it is important to discover exactly how much steam is being used for this and other purposes about the boiler plant. The accurate determination of these steam quantities demands measuring accuracy of a fairly high order (see Chapter X). When the consumption of the auxiliaries such as fans, steam jets, and pulverised fuel equipment exceeds 1 per cent. of the heat in the coal, it must be deducted in arriving at the net efficiency.

When electrical power is used the equivalent steam consumption per hour is the input to the motors in kW. multiplied by the steam consumption per kW. hour of the prime mover producing the electricity. This conversion factor will vary over a very wide range according to the size and type of the prime mover, but since, as indicated, the evaluation of the power consumed by auxiliaries is only for comparative purposes, it is convenient to adopt the conversion factor given in the B.S. code, namely, 15,000 B.Th.U. per h.p. or 20,000 B.Th.U. per kW. hour.

For steam-operated auxiliaries the steam consumption can sometimes be measured without undue complications, but frequently this will hardly be practicable, and the maker's figures should then be adopted.

When steam jets are used below the fires either for cooling or for forced draught, the steam consumption can be calculated with reasonable accuracy if the diameter of the throat of the jet, and the steam pressure and temperature in front of the nozzle are known. A suitable formula is given elsewhere (Chapter VI). It may be recalled that the passage of high velocity steam leads to erosion and therefore in practice the steam consumption of nozzles gradually increases as wear proceeds. A watch should be kept on this wear, and as the cost of nozzles is very low, they should be renewed at reasonable intervals to keep down steam consumption.

It is usual to exclude feed pumps from individual boiler auxiliaries for various reasons. If it is considered necessary for any reason to undertake a test of the feed pumps, then this should be carried out by means of a separate efficiency test.

GENERAL REMARKS

Consideration will reveal that the whole object of boiler testing is to compare the results being obtained with some standard. It is therefore not only desirable to have a standard of performance for the particular plant which can be set by the tests outlined earlier, but also to have detailed particulars of what has been achieved on other plants of a similar type and size.

Comparison of the results of a number of efficiency tests (which must be at the same rating) made from time to time on the same plant may reveal a change in performance. When a reduction in thermal efficiency is suspected or known, it may arise from one or more of a large number of causes. A change in efficiency may be due to a change in the character of the coal used. A loss arising from fouling of the exterior heating surfaces, or due to scaling on the water or steam sides, will be indicated by a rise in the temperature of the flue gases; this rise may be due to fouling over many sections of the boiler, or it may be confined to one section. If, for example, the economiser is at fault, the temperature of the gases entering it will be normal, while the temperature of the gases leaving it will be higher than normal; the temperature of the water leaving the economiser will then be lower than normal. Whether the loss in efficiency is due to internal or external fouling must be ascertained by visual inspection. If a change in efficiency is shown by tests, particular care must be

taken to recheck the instruments used before making any major deductions from the results.

If the temperature of the superheated steam is low this may be brought about by one of three causes :—

(1) Fouling of the exterior surfaces.

This can be determined by visual inspection.

(2) Priming.

If water is carried over from the boiler drum into the superheater so that the superheater tubes are used for evaporating as well as superheating, a fall in the steam temperature will inevitably result and the separator in the boiler drum must therefore be examined. If solids are carried over with the steam from the boiler drum into the superheater, scale or sludge will be deposited on the inside of the superheater tubes. If this should occur attention must be given not only to the separator in the drum, but also to the treatment of the feed water (Chapter XV).

(3) Displacement of baffles.

Visual inspection will indicate whether all baffles are correctly located.

The quality of the combustion as indicated by the gas analysis (CO_2 and CO content) must be watched carefully as has been indicated in Chapters IV and VI.

If the best results are to be obtained from boiler working it is essential that a number of quantities and qualities should be under constant observation. The necessary observations can be made relatively easily if the boiler is properly equipped with suitable and reliable instruments maintained in good working order. The ideal is so to equip the boiler that it can be operated always under test conditions.

The smallest plant should be provided with some instruments so that the operators are aware of what is occurring. The minimum equipment may be considered to be that which would provide the following information :—

(1) Measurement of coal burnt over given periods.
(2) Output of boiler plant in lb. of steam or weight of water evaporated.
(3) Temperature of steam (if superheaters are fitted).
(4) Temperature of flue gases.
(5) Temperature of water entering economiser.
(6) Temperature of water leaving economiser.
(7) Draught at base of chimney.
(8) Draught at economiser inlet.
(9) Draught in furnace.
(10) CO_2 content of flue gases.

With larger plants, equipment on a more generous scale is a wise investment, which if properly used will pay for itself many times over.

Even with small plants, if mechanically fired, the efficiency should average between 70–80 per cent., and should be nearer the higher value than the lower. In large power stations average efficiencies of over 85 per cent. are secured over the year. This is achieved by installing plant which will produce a high efficiency, properly testing it under the best conditions, and then arranging the equipment and operation so that all losses are kept constantly to the minimum value.

Two test results (Tables 70 and 71) are set out in the manner recommended by the British Standard code to serve as a general guide.

TABLE 70. LANCASHIRE BOILER

		1. Specified Conditions and Performance		
Item	1.	Evaporation, normal load	lb./hr.	9,000
,,	2.	Pressure of steam	lb./sq. in.	230
,,	3.	Final steam temperature	° F.	675
,,	4.	Initial temperature of feed water	° F.	110
,,	5.	Class of fuel	Slack coal.	
,,	6.	Overall thermal efficiency	per cent.	76
		2. Particulars of Plant		
,,	7.	Boiler (type) Lancashire 30 feet × 9 feet, heating surface	sq. ft.	1,032
,,	8.	Superheater (type) downtake, heating surface	sq. ft.	282
,,	9.	Economiser (type) gilled tube, heating surface	sq. ft.	1,760
,,	11.	Method of firing	Coking stoker.	
,,	12.	Grate area	sq. ft.	48
,,	13.	Draught plant	Motor driven induced draught fan.	
		3. Data from Observations, Analyses, etc.		
,,	14.	Duration of test	hr.	8
		A. Fuel		
,,	15.	Description	Slack coal.	
		Size : Through 1 in. B.S. Sieve	per cent.	100·0
		,, ½ in. ,, ,,	,, ,,	92·4
		,, ⅜ in. ,, ,,	,, ,,	82·7
		,, ¼ in. ,, ,,	,, ,,	66·9
		,, ⅛ in. ,, ,,	,, ,,	46·7
		,, 1/16 in. ,, ,,	,, ,,	28·8
,,	16.	Proximate analysis as weighed :—		
		Moisture	,, ,,	7·9
		Volatile matter	,, ,,	29·8
		Fixed carbon	,, ,,	57·0
		Ash	,, ,,	5·3
				100·0
,,	17.	Gross calorific value as weighed	B.Th.U./lb.	12,890
,,	18.	Net or lower calorific value as weighed	B.Th.U./lb.	12,370
,,	19.	Fuel fired per hour	lb.	1,359
,,	20.	Fuel fired per hour per square foot of grate area	lb.	28·3
		B. Ashes and Clinkers		
,,	21.	Collected per hour	lb.	81
,,	22.	Percentage of fuel fired		5·9
,,	23.	Percentage of combustible		33
		C. Water		
,,	24.	Temperature entering economiser	° F.	109
,,	25.	Temperature entering boiler	° F.	291
,,	26.	Evaporation per hour	lb.	10,327
,,	27.	Evaporation per lb. of fuel as weighed	lb.	7·6
		D. Steam		
,,	28.	Gauge pressure	lb./sq. in.	229
,,	29.	Final temperature	° F.	670
,,	30.	Heat per lb. above feed temperature	B.Th.U.	1282·5
		E. Flue Gases		
,,	31.	Temperature leaving boiler	° F.	674
		Temperature leaving economiser	° F.	416
,,	32.	Analysis leaving economiser :—		
		CO_2	per cent.	9·1
		O_2	,, ,,	10·1
		CO		Nil
		F. Draught		
,,	33.	Suction over fire	in. w.g.	0·8
,,	35.	Suction leaving economiser	in. w.g.	1·9

G. Air

Item 39.	Temperature of air supply to furnace	° F.		82

4. Results

,, 40.	Overall thermal efficiency on gross calorific value of the fuel	per cent.		75·6

A. Heat Account Calculated on the Net Calorific Value of the Fuel

,, 41.	Overall thermal efficiency	per cent.		78·8
,, 42.	Loss due to sensible heat in chimney gases	,,	,,	12·8
,, 43.	Loss due to unburnt CO	,,	,,	Nil
,, 44.	Loss due to combustible matter in ashes and clinkers..	,,	,,	2·3
,, 45.	Balance—radiation, grit emitted, and other unmeasured losses	,,	,,	6·1
				100·0

Note.—The somewhat high loss due to combustible matter in ashes and clinker is attributable to the high percentage of fines in the fuel.

Table 71. Water-tube Boiler

1. *Specified Conditions and Performance*

Item 1.	Evaporation, normal load	lb./hr.		25,000
,, 2.	Pressure of steam	lb./sq. in.		160
,, 3.	Final steam temperature	° F.		550
,, 4.	Initial temperature of feed water	° F.		150
,, 5.	Class of fuel..	Scotch washed nuts.		
,, 6.	Overall thermal efficiency	per cent.		84

2. *Particulars of Plant*

,, 7.	Boiler (type) water-tube, heating surface	sq. ft.		6,182
,, 8.	Superheater (type) integral, heating surface	sq. ft.		1,540
,, 9.	Economiser (type) cast iron vertical tube, heating surface	sq. ft.		2,556
,, 11.	Method of firing	Chain-grate stoker.		
,, 12.	Grate area	sq. ft.		140
,, 13.	Draught plant	Motor driven induced draught fan.		

3. *Data from Observation, Analyses, etc.*

,, 14.	Duration of test	hrs.		6

A. Fuel

15.	Description	Scotch washed nuts.		D.S.
16.	Proximate analysis as weighed :—			
	Moisture	per cent.		10·3
	Volatile matter	,,	,,	32·5
	Fixed carbon	,,	,,	48·5
	Ash	,,	,,	8·7
				100·0
,, 17.	Gross calorific value as weighed	B.Th.U./lb.		11,980
,, 18.	Net or lower calorific value as weighed	B.Th.U./lb.		11,460
,, 19.	Fuel fired per hour..	lb.		2,878
,, 20.	Fuel fired per hour per square foot of grate area ..	lb.		20·6

B. Ashes and Clinkers

,, 21.	Collected per hour	lb.		193
,, 22.	Percentage of fuel fired			6·7
,, 23.	Percentage of combustible			12·2

C. Water

,, 24.	Temperature entering economiser	° F.		155
,, 25.	Temperature entering boiler	° F.		222
,, 26.	Evaporation per hour	lb.		24,773
,, 27.	Evaporation per lb. of fuel as weighed	lb.		8·61

D. Steam

Item 28.	Gauge pressure	lb./sq. in.		166
,, 29.	Final temperature	° F.		545
,, 30.	Heat per lb. above feed temperature	B.Th.U.		1,176

E. Flue Gases

,, 31.	Temperature leaving economiser	° F.	284
,, 32.	Analysis leaving economiser :—		
	CO_2	per cent.	10·6
	O_2	,, ,,	8·6
	CO		Nil

F. Draught

,, 33.	Suction over fire	in. w.g.	0·3
,, 34.	Suction leaving boiler	,, ,,	0·6
,, 35.	Suction leaving economiser	,, ,,	0·9

G. Air

,, 39.	Temperature of air supply to furnace	° F.	80

4. Results

,, 40.	Overall thermal efficiency on gross calorific value of the fuel	per cent.	84·5

A. Heat Account Calculated on the Net Calorific Value of the Fuel.

,, 41.	Overall thermal efficiency	per cent.	88·4
,, 42.	Loss due to sensible heat in chimney gases	,, ,,	6·7
,, 43.	Loss due to unburnt CO	,, ,,	Nil
,, 44.	Loss due to combustible matter in ashes and clinkers	,, ,,	1·0
,, 45.	Balance—radiation and other unmeasured losses	,, ,,	3·9
			100·0

The examples in Tables 70 and 71 are for efficient plants. In Table 72 are other results arranged on different lines, based on the gross C.V. of the coal and comparing the heat distribution of two plants, one of which is efficient and the other not.

Table 72

	Efficient plant		Inefficient plant	
	B.Th.U.	Per cent.	B.Th.U.	Per cent.
(1) Heat absorbed by water and steam in boilers, superheaters and economisers	9,925	79·40	7,588	60·70
(2) Heat loss due to moisture in coal	24	0·19	29	0·23
(3) Heat loss due to water from combustion of hydrogen	560	4·48	672	5·38
(4) Heat loss due to moisture in air	40	0·32	49	0·39
(5) Heat loss due to dry chimney gas	1,312	10·50	2,400	19·20
(6) Heat loss due to incomplete combustion of carbon	—	—	600	4·80
(7) Heat loss due to unconsumed combustible in ash	225	1·80	525	4·20
(8) Heat loss due to unconsumed hydrogen, and hydrocarbons, radiation, and unaccounted for	414	3·31	637	5·10
	12,500	100·00	12,500	100·00

Item (1) should be subdivided to show the heat absorbed by the boilers, superheaters and economisers separately. Typical values are as follows :—

	Efficient plant		Inefficient plant	
	B.Th.U.	Per cent.	B.Th.U.	Per cent.
(1a) Heat absorbed by water and steam in boiler	7,837	62·7	6,300	50·40
(1b) Heat absorbed by steam in superheaters..	688	5·5	525	4·20
(1c) Heat absorbed by water in economisers	1,400	11·2	763	6·10
	9,925	79·4	7,588	60·70

As a guide to those who are able to make more detailed examination of their plants, Table 73 is here included.

TABLE 73. STANDARD FORM FOR REPORTING BOILER HOUSE PERFORMANCE

PERIOD OF TRIAL

N.C.R. PER BOILER, LB./HR.

EVAPORATION

Total water evaporated corrected for blow-down.
" " " per boiler per hour—lb. actual.
" " " " " " " —lb. from and at 212° F.
Evaporation, from and at, per square foot of heating surface per hour, boiler only.
Evaporation—lb., actual, per lb. coal.
Evaporation—lb., from and at, per lb. of coal.
Per cent. of steam superheated.
Saturated steam to grate—lb. per hour.
Water to grate—lb. per hour.
Pressure (gauge) lb. per square inch at superheater exit.
Temperature of superheated steam.
Degrees of superheat.
Heat in steam above feed temperature.

COAL USED

Total fuel burned.
Fuel per boiler per hour.
Fuel per square foot grate area—lb. per hour.
" " " " " " " " " standard coal of 7,000 calories or 12,600 B.Th.U.
Heat release. Therms per square foot per hour.

HEAT BALANCE

Per cent. heat in process steam.
Per cent. heat in steam auxiliaries and grate cooling.
Per cent. heat in steam for forced draught steam jets.
Total heat of above.
Per cent. heat lost in flue gases. Total.

(*a*) by gases.
(*b*) from moisture in coal.
(*c*) from moisture in air.
(*d*) from hydrogen in coal.
(*e*) from unburned gases.
(*f*) from steam or water supplied under grates.

Per cent. heat lost in carbon in ashes.
,, ,, ,, ,, in dust.
,, ,, ,, ,, in blow-down.
,, ,, ,, ,, in radiation and convection.
,, ,, ,, ,, unaccounted for.

EFFICIENCY OF BOILER PLANT

Boiler.
Superheater.
Economiser.
Air heater.

DRAUGHTS

Induced draught fan.
In smoke chamber (Economic boiler).
In superheater chamber (Economic boiler).
Entering air heater.
Leaving air heater.
Entering economiser.
Leaving economiser.
Side flues (Lancashire boiler).
Over fire.

Forced draught.
Inlet air heater.
Exit air heater.
Under grates.
Secondary air pressure.

ANALYSIS OF GASES

$CO_2 - O_2 - CO$: at boiler exit.
economiser exit.
air heater exit or chimney.

Temperature at combustion chamber.
boiler exit.
economiser exit.
air heater exit or chimney.

FEED WATER AND AIR DATA

Temperature, feed water economiser inlet.
,, ,, ,, economiser outlet.
Per cent. blow-down.
Temperature, preheated air.
Temperature, air entering air heater.
Temperature, boiler house floor.

ANALYSIS OF CARBON LOSSES

Total weight of refuse.
Total weight of refuse as per cent. of coal.
Per cent. carbon in refuse.
Dust ex superheater, economiser, air heaters or grit arresters—lb. per hour.
Per cent. carbon in dust.
Dust in flue gases—lb. per hour.
Per cent. carbon in dust.

ANALYSIS OF STOKER OPERATION

Fuel bed thickness.
Grate speed.
Times fires levelled per hour.
Report on clinker formation.
Water added to coal—lb. coal.
Report on coal segregation.
Report on smoke formation.

POWER REQUIRED BY AUXILIARIES

Mechanical stoker.
Coal elevator.
Economiser.
Pump.
Fans.
P.F. mills.

ANALYSIS OF COAL

Moisture.
Ash.
Volatile on dry coal.
Volatile—dry ashless coal.
Calorific value as received—gross.
,, ,, ,, ,, —net.
Calorific value of dry ashless coal.

ULTIMATE ANALYSIS OF COAL

Carbon.
Hydrogen.
Nitrogen.
Sulphur.
Oxygen.
Caking and swelling properties.
Ash melting point (reducing atmosphere).

SIZE ANALYSIS OF COAL

0 to $\frac{1}{8}$ inches.
$\frac{1}{8}$ to $\frac{1}{4}$,,
$\frac{1}{4}$ to $\frac{1}{2}$,,
$\frac{1}{2}$ to 1 ,,
1 to $1\frac{1}{2}$,,
$1\frac{1}{2}$ to 2 ,,

CHAPTER XVII

GAS PRODUCERS

Gas producer plant—Nature and Characteristics of fuel—Blast and the admission of steam—Clinker—Channelling—Producer efficiency and heat balances—Operation of producer—Purification of producer gas—Utilisation of producer gas.

PRODUCER gas is made by blowing a mixture of air and steam through a bed of hot coal or coke when the combustible in the solid fuel is largely converted into combustible gas, consisting of carbon monoxide and hydrogen accompanied by nitrogen from the air. The resulting gaseous fuel may be used in the crude hot state, when it contains a proportion of tar and dust ; or it may be cleaned and distributed in a system of supply mains in a manner similar to town gas. It possesses most of the advantages of gaseous fuel, and in consequence it finds wide application in industrial processes.

This process of complete gasification is carried out usually in vertical cylindrical chambers which may be lined with firebrick or may be formed by an annular jacket containing water from which is raised by heat conduction from the fuel bed the steam required to be added to the blast. The fuel is fed by gravity from a charging hopper on to a hearth or grate, which supports the fuel bed, and provides the tuyeres or points of admission of the blast to the fire. The producer gas is conducted from the producer chamber by an off-take flue to a dust catcher or other cleaning plant (if any is provided) in which the gas may be purified from dust, tar and sulphur prior to distribution.

A clear distinction must be drawn between gas producer plants producing raw hot gas, usually distributed through insulated mains to furnaces and plants in their near vicinity, and plants in which the gas is cooled and freed from deleterious constituents. Cooled and cleaned gas may be produced in suction plants or large pressure units equipped with plant capable of cleaning the gas for distribution in relatively complicated systems of mains and for appreciable distances. The principles of gasification in the producer are the same whatever the conditions of purification.

The principles of the reactions between carbon, air and steam that lead to the formation of producer gas were discussed in Chapters IV and V. From what was there said it will have become evident that producer gas should contain as much combustible gas, CO and H_2, and as little CO_2 as possible, since CO_2 represents carbon that has already been completely burnt in the producer and is of no further value as a fuel. It is also clearly in the interests of fuel economy to effect the decomposition of the steam to the greatest practicable extent.

These objectives, as will also have been deduced from Chapters IV and V, can be achieved only by

(1) bringing the air and steam into intimate contact with the carbon and allowing an adequate time of contact ; this requires a uniform flow of gas through all parts of the fuel bed,
(2) by operating with a high temperature, well above 1,000° C., in the reduction zones.

ARRANGEMENT OF PRODUCER PLANT

Fig. 127 is a diagrammatic sketch of a gas producer. A mixture of air and steam is blown in at the base and passes through the air distributor into the ash zone. Since the ash serves further to distribute the blast uniformly it is therefore important that it should be in a good granular condition, free from large masses of clinker.

The air-steam mixture then passes up the fuel bed where the reactions described in Chapter IV occur. The oxidation zone extends to a depth of 4–5 inches above the ash zone. The depths of the other zones are not easily determined and will vary with the depth of the fuel bed, the gasification rate, and the reactivity and size of the fuel. As a rough approximation it may be stated that for a generator with 3 feet 6 inches depth of fuel bed the primary reduction zone and the distillation zone will each be about 1 foot deep, whilst the depth of the secondary reduction zone will be approximately 1½ feet. The reactions occurring in the several zones are summarised in Figs. 13 and 17.

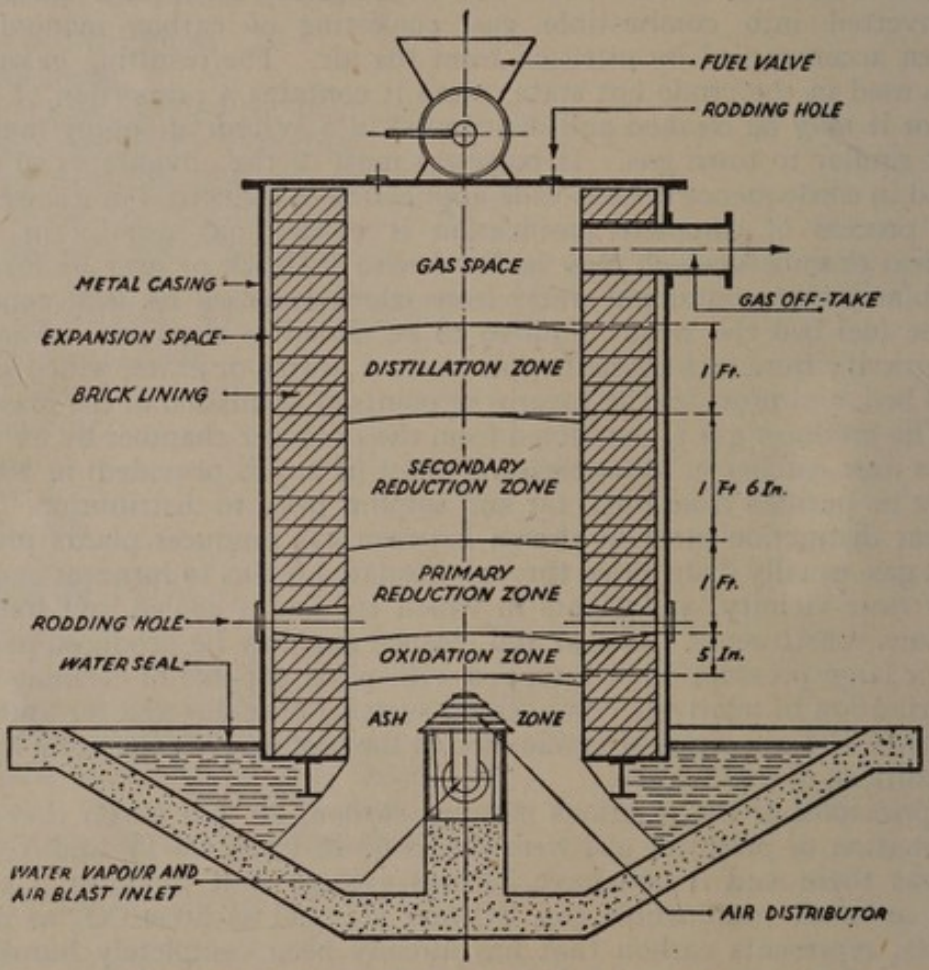


FIG. 127. Diagrammatic sketch of a gas producer.

(Reproduced by permission from "Producer Gas Plant for Industrial Purposes," published by The National Federation of Gas Coke Associations, The British Hard Coke Association, and The South Wales Anthracite and Dry Coal Committee.)

This subdivision into zones is conventional since there is no sharp line of demarcation; the depth of the zones may differ from the figures given.

At the top of the fuel bed is the distillation zone. Here the hot gases come into contact with the freshly charged fuel and the volatile matter is distilled into the gas space above the fuel in the generator. With most anthracites, for example, the volatile matter is converted into permanent gases of which methane is the most valuable on account of its high calorific value. Fuels of high volatile content, including some anthracites, evolve tar vapours in addition to permanent gases. With coke as fuel the distillation zone proper is absent, but some cokes contain "black ends" (i.e. incompletely carbonised portions) which give rise to small quantities of tarry vapours on distillation.

For some purposes the gas can be used in its crude, hot state, containing dust, sulphur compounds and (if made from bituminous, semi-bituminous coal or

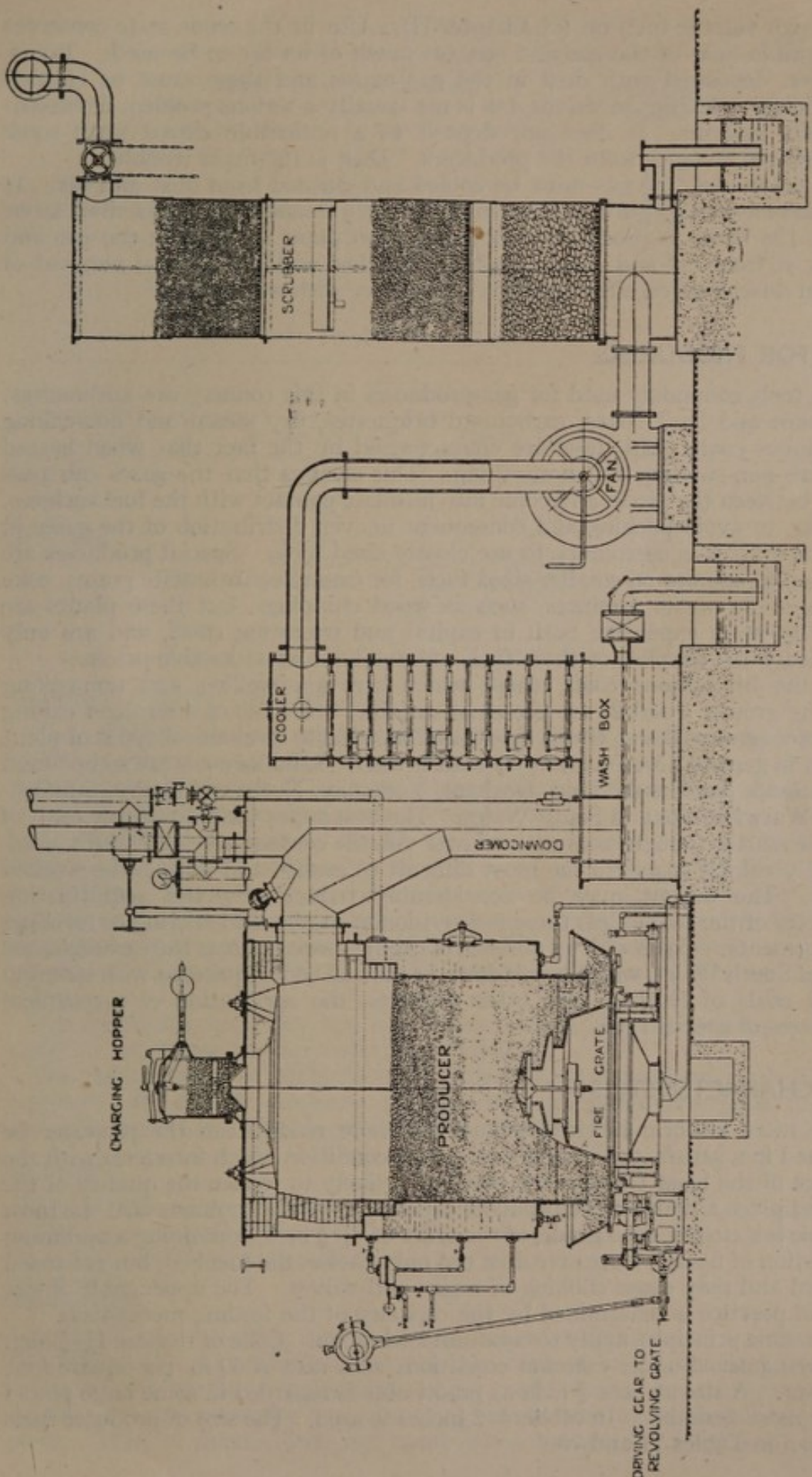


FIG. 128. Diagram of plant for cleaning producer gas.

(Reproduced by permission from "Producer Gas Plant for Industrial Purposes," published by The National Federation of Gas Coke Associations, The British Hard Coke Association, and the South Wales Anthracite and Dry Coal Committee.)

other high volatile fuel) tar (cf. Chapter II). Use in the crude state conserves the sensible heat of the gas and enables much of its tar to be used. Tar is, however, deposited with dust in the gas mains and these must be cleaned periodically. Except in valves, tar is not usually a serious problem in reasonable flue systems. It does not deposit to a noticeable extent until some considerable distance from the producers. Dust is the main trouble.

For other uses, the gas must be cooled and cleaned from dust and tar. It may also have to be purified from sulphur in gas cleaning plant described later.

Fig. 128 shows a producer with purification plant for cooling the gas and freeing it from tar and dust. Cooling, together with the partial removal of tar and dust, is often effected by direct contact with water.

FUEL FOR PRODUCERS

The fuels commonly used for gas producers in this country are anthracites, gas cokes and hard cokes, carbonised briquettes, dry steam and non-caking bituminous coals. All these are characterised by the fact that when heated they are non-swelling and non-caking. This ensures that the gases can pass freely between the pieces and come into intimate contact with the fuel surfaces. Further, to avoid packing and consequent uneven distribution of the gases in the generator it is customary to use closely sized fuels. Special producers are designed to operate on smaller-sized fuels, for example, anthracite grains, coke breeze, or on waste products, such as wood chippings, but these plants are generally more expensive both in capital and operating costs, and are only installed when supplies of these fuels are available at attractive prices.

Of the bituminous coals, those which are non-swelling and non-caking (Lessing groups 1 and 2) are the most suitable, but coals of a stronger caking character (groups 3 and 4) can be utilised in suitable mechanical types of plant so that in general a wide variety of coals is used. Suitable coals are to be found in Scotland, Northumberland, Durham, Yorkshire, Notts and Derby, Staffordshire, Warwickshire and South Wales. The best types yield 30–40 per cent. of volatile matter though coals with a lower volatile content are successfully used. Strongly caking coals are the most difficult to gasify, and have to be worked hotter. The output may be considerably reduced by the agglutinating character of the coal unless there is provision of mechanical stirring or levelling arrangements. Coals with an absence of caking power such as the carbonaceous coals of South Wales, which break down in the fuel bed, require, as with strongly caking coals of Durham and other districts, the application of mechanical producers of special type.

SIZE CHARACTERISTICS

The more uniform in size the fuel the more readily can the producer be operated in a satisfactory manner since any condition which interferes with the passage of the blast through the fuel bed is likely to reduce the quality of the gas and place a limit on the producer capacity. For bituminous coal the most suitable is a closely sized grade, of the order of $1\frac{1}{4}$–$\frac{3}{4}$ inch, containing a minimum proportion of fines. Excessive dust not only chokes the fuel bed, but is carried forward and may cause choking of mains and valves. The upper limit of size used in practice is determined by the capacity of the feeding mechanism.

The same principles apply to coke and anthracite. Coke of the size $1\frac{1}{4}$–$\frac{3}{8}$ inch has been gasified under extreme conditions at a rate of 60 lb. per square foot per hour. A size as near $\frac{3}{4}$ inch as practicable is regarded in some large plants as the most desirable. In others 1–2 inches is used. The size of producer fuels is shown in Tables 74 and 75.

TABLE 74. COAL

Rated capacity of producer		Fuel size inches
lb. per hour.	Therms per hour	
40–45	5	¾–1½
170–850	20–100	¼–¾ ¾–1½ 1¼–2
Over 850	Over 100	1¼–2

TABLE 75. COKE

Types of fuel	Rated capacity		Fuel size inches
	lb. per hour	Therms per hour	
Gas coke ..	45–50	5	½–1¼
	170–850	20–100	1–2
	Over 850	Over 100	1–2
Hard coke ..	45–50	5	½–1
	170–850	20–100	¾–1½
	Over 850	Over 100	½–1½ (as near ¾ in. as practicable)

REFRACTORINESS OF THE ASH

Fuels with ash of a low refractoriness cause clinkering, whereas those producing ash having a high fusion point, of the order of 1,400° C., permit of the use of less steam than when the ash has a low fusion point, a subject which will be discussed later ; the producer can be run hotter and a gas of higher calorific value results. Alternatively an ash with a refractoriness below 1,250° C. will require additional steam, and a limit may be put on the gasification rate practicable. The actual proportion of ash is less important if it is non-clinkering, and the producer is not driven at a rate requiring clinker removal. Clinker formation was discussed in detail in Chapter VI, from which it will be seen that ash fusion temperatures as ordinarily determined form only a very general guide.

For anthracite and coke the lowest permissible ash fusion temperature is 1,150° C. determined under reducing conditions.

Too high a content of ash in the fuel is especially undesirable in hand-clinkered producers, and even in mechanical producers may restrict throughput and give rise to operating troubles through uneven gasification.

SULPHUR

For certain uses the sulphur content of producer fuel must be kept low : below 1 per cent. for acid steel melting, 1·5 per cent. for metal heating, and 2 per cent. for basic steel melting.

MOISTURE

Where the gas is used in the crude state, excessive moisture even to the extent of 2–3 per cent. above the normal moisture content of the coal is deleterious, and will lower the flame temperature of the gas in the furnace partly by adding to the inerts and partly by heat-absorbing reactions in which the steam participates. This, of course, does not apply when the gas is cooled and cleaned.

Any water added with the fuel will be evaporated in the producer where it will reduce the outlet temperature of the gases ; the steam so evaporated must be condensed in the purification system and thus the lower outlet temperature does not mean less work on the cooler.

THE BLAST

The composition of producer gas depends more upon the proportion of water vapour present in the air blast than upon the type of fuel gasified. This statement applies more particularly to fuels of low volatile matter content, where the volume of gases obtained by distillation is only a small proportion of the total gas made, but even with high-volatile fuels variations of the water vapour-air ratio largely determine the composition and consequently the combustion characteristics of the resulting gas.

The amount of hydrogen present in the gas is the controlling factor and determines the rate at which the gas can be burned without the flame lifting from the burner ports. Where free-flame burners are used it is, therefore, important that the amount of water vapour present in the air blast should be such that the hydrogen content is not less than about 11 per cent.

The blast is formed by blowing a regulated quantity of steam into the air supply. The temperature of the blast is raised to a predetermined figure, at which the air is supposed to be just saturated with water vapour. This is not exactly correct but the blast temperature is a good and simple practical guide. The quantity of steam added to the blast is thus indicated by the blast temperature, which in its turn is controlled by the steam admission valve. Further steam, up to 25 per cent., is derived from the quenching of the clinker in the water seal of the ashpan.

Since considerable importance is attached to the correct proportion of water vapour in the blast, the design of the blast-mixing box, the steam injector or the vaporiser must receive special attention. The essential features are an instrument to measure the temperature of the blast, and ready means of control. Automatic controls which operate the steam valve on the indication of the temperature of the blast are available and should be used whenever practicable. Steam operated turbo-blowers are also used, the exhaust steam being used to saturate the blast.

Table 76, which is an approximate guide and is not to be taken as exact for all steam pressures and degrees of superheat, supplies the data to relate the saturation temperature to the quantity of steam used.

TABLE 76

Blast saturation Temp. ° C. (B.S.T.)	Wt. of steam, lb. per 1,000 cu. ft. of dry air as measured at N.T.P.	Lb. of steam per lb. of dry air	Lb. of steam per lb. coal gasified
50	6·95	0·086	0·26
55	9·28	0·115	0·34
60	12·35	0·153	0·46
65	16·55	0·205	0·61

Studies of the behaviour of steam in gas producers made by W. A. Bone and R. V. Wheeler (*J. Iron and Steel Institute*, 1907, i, p. 126 ; 1908, ii, p. 206) indicate that up to 55° C. blast saturation temperature the steam is completely decomposed. Their figures are shown in Table 77.

TABLE 77

Steam saturation Temp. of blast ° C.	Lb. of steam in blast per lb. of coal gasified	Per cent. steam decomposed approx.	Lb. of steam decomposed
45	0·20		0·20
50	0·21	100	0·21
55	0·32		0·32
60	0·45		0·36
65	0·55	80	0·44
70	0·80	60	0·48
75	1·10	50	0·55
80	1·55	40	0·62

Extensive and informative figures have been obtained by Gibson and Gwyther (Table 78) in relation to the gasification of gas coke.

In Bone and Wheeler's trials rather deeper fires were used than in current practice, and the Mond system superheaters were used in circuit. Whilst the actual blast temperature entering the fire was not given it has been stated that in the original Mond system preheats up to 250° C. were obtained. F. Clements's experiments (*J. Iron and Steel Institute*, 1922) showed a distinct relation between the thickness of the firebed and the steam decomposed, and the results differ from those of Bone and Wheeler. It is doubtful if the normal bituminous coal producers decompose steam as efficiently as Bone and Wheeler found. Slow-burning coke producers feeding engines present totally different conditions.

Experimental information from Clements's work may be thus summarised: As the blast saturation for any given fuel bed depth and rate of firing is varied from 45° to 70° C. blast saturation temperature (B.S.T.) the following changes are found to occur in the composition of the gas from bituminous coal.

(*a*) The gas composition shows a gradual change between the following ranges, advancing for each constituent from left to right as the blast saturation temperature increases :—

CO_2	CO	H_2	CH_4	N_2
3–9	30–20	11–20	0·3–3	52–46

(*b*) The heating value of the gas drops from 160 to 130 B.Th.U. per cubic foot (measured at 30 inches Hg. 60° F. saturated).

(*c*) The volume of gas produced per lb. dry coal increases from 3–10 per cent. according to the type of fuel and the conditions of operation.

(*d*) The combined effect of the above two changes is given by a statement of the cold gas efficiency, which is the gross heat of combustion of the gas expressed as a percentage of the initial gross calorific value of the fuel. An optimum value is found when the steam admission to the blast is about 0·3–0·4 lb. of steam per lb. coal. A corresponding blast saturation temperature range is 53°–57° C. Thereafter with increased admission of steam the cold gas efficiency falls appreciably, and at the same time the temperature of the issuing gases becomes reduced.

The optimum blast saturation temperature, which is generally 55°–62° C., will, however, vary with the nature of the fuel. A fusible ash may involve an increased B.S.T. to avoid trouble even at a cost of sacrificing the quality of the gas. Poor quality gas can result from excessive accumulation of ash and clinker just as easily as from too low a fuel bed temperature. Fuels such as anthracite, high-volatile non-caking coals and soft coke can stand a higher B.S.T. (e.g. 60–62° C.) than gas and hard cokes.

TABLE 78. TEST MADE ON 30 H.P. NATIONAL SUCTION GAS PRODUCER
(Gibson and Gwyther)

Fuel used. Gas coke of analysis :—

C	81·75 per cent,	Ash	8·3 per cent.
H_2	1·19 „ „	Calorific value	12,739 B.Th.U./lb.
O_2, N_2 and S	8·79 „ „	Size	¾–1¼ inch.

Number of trial	1	2	3	4	5	6	7
Saturation temperature of air supply, ° F.	86·0	106·5	120·0	124·4	132·3	137·6	147·5
Dry fuel burned per hour in lb.	28·16	26·00	24·90	26·15	21·70	23·78	29·40
Weight of H_2O vapour supplied/lb. dry fuel	0·1205	0·2170	0·3565	0·3990	0·5225	0·6310	0·850
Water vapour decomposed, per cent.	100·0	100·0	100·0	100·0	85·6	72·7	47·0
Composition of gas by volume, per cent. CO_2	2·91	3·48	4·73	5·10	5·06	7·64	10·4
Composition of gas by volume, per cent. O_2	0·84	0·74	0·84	0·75	0·57	0·87	0·68
Composition of gas by volume, per cent. CO	29·32	27·52	26·10	25·45	25·4	22·8	19·4
Composition of gas by volume, per cent. CH_4	0·20	0·35	0·36	0·28	0·33	0·36	0·41
Composition of gas by volume, per cent. H_2	2·61	8·10	11·0	13·10	10·6	11·0	9·64
Composition of gas by volume, per cent. N_2	63·12	59·81	56·97	55·32	58·04	57·33	59·47
Calorific value (calc.)* B.Th.U./cu. ft. at 60° F. sat. 30 in. Hg	103·7	116·9	121·8	125·7	118·1	111·3	96·6
Specific gravity (air = 1)*	0·95	0·92	0·94	0·87	0·89	0·91	0·94
Volume of air at 32° F. and 29·9 in. Hg per lb. dry fuel, cu. ft.	56·1	54·0	54·85	53·0	54·9	54·85	54·71
Thermal efficiency, per cent.	61·95	70·25	77·3	78·6	75·0	71·08	58·02
Theoretical flame temperature ° C. (assuming 10 per cent. radiation loss)*	1,535	1,622	1,634	1,643	1,594	1,544	1,429

* These figures have been added.

It is to be noted from Table 77 that the steam added to the air is not proportional to the B.S.T., but as the B.S.T. rises the steam added becomes proportionately greater.

PHYSICAL EFFECTS OF STEAM IN PRODUCERS

(1) Because of its cooling effect steam reduces the temperature at the base of the producer and thus minimises clinker formation. For this reason it has been the practice to increase the steam admission when experiencing difficulty due to clinker formation. This is not necessarily the best practice, because of the effects on the subsequent reactions.
(2) The passage of undecomposed steam through the producer cools the reaction zones by conveying heat from them to the upper part of the fuel bed. The importance of maintaining a high temperature in the reduction zone has been indicated in Chapter V. Excess steam must be avoided.
(3) The reactions between steam and carbon absorb heat and the addition of steam in moderate quantities thus serves in two ways to moderate excessive temperatures at the base of the producer which would otherwise cause clinker with even refractory ashes.
(4) A balance must therefore be struck between the steam admitted (which absorbs heat) and the air admitted (which generates heat).
(5) Since the reaction between steam and carbon yields essentially a mixture of $CO + H_2$ having a C.V. of 320 B.Th.U. per cubic foot gross, steam admission increases the C.V. of the gas. When the steam admission becomes so great as to cool the reaction zones unduly, both reactions $CO_2 + C = 2CO$ and $H_2O + C = CO + H_2$ are repressed, with the result that the CO_2 content of the gas becomes greater and its C.V. falls. The experiments of Gibson and Gwyther just cited illustrate this point.

CLINKER IN PRODUCERS

Clinker is formed by the fusion and liquefaction of ash in the hot zones of the generator. In the liquid state it is not likely to cause excessive difficulty, for in fact slagging producers have been used, but on reaching the ash zone it is cooled and forms a hard dense material which may be very difficult to break up and remove.

Causes of excessive clinker formation are :—

(1) Use of high ash content fuels.
(2) Inefficient or infrequent rodding.
(3) Use of fuels containing a high percentage of fines.
(4) An abnormally high rate of gasification.
(5) The development of hot spots in the fuel bed.
(6) The use of fuel having too low an ash fusion temperature.

A certain amount of clinker will almost always form on the walls of brick lined producers, due to the straight face of the wall giving an easier passage for the blast than the body of the fire, thus causing a greater tendency to channelling and hot spots.

The firebed must be cleaned at regular intervals, and the clinker must be removed when necessary, but never less frequently than at intervals which allow of an accumulation of more than 10 inches of ash in static producers. The exact period must depend upon the rate of gasification, the character of the ash, and the circumstances which govern uniform removal of ash. The smaller sizes of fuel generally contain higher relative percentages of ash and since they may be gasified more easily, high local temperatures occur, thus causing fusion of the ash. This condition may, to some extent, be mitigated by avoiding the use of " dusty " fuels.

Excessive rates of gasification result in the liberation of increased quantities of ash and the generation of abnormally high temperatures in the oxidation zone. These factors lead to the formation of a greater quantity of clinker.

Hot spots are usually associated with channelling in the fuel bed—notably at the walls—by hanging of the charge. The cores which form in the main combustion zone give rise to channelling. Blast passes through the channels and there results combustion of producer gas in the fuel bed of the generator. The development of a hot spot should therefore be taken to indicate the need for immediate rodding with the object of eliminating channelling.

Difficulties due to clinker may therefore be decreased by ensuring that the producer is not worked above its rated output, by constant attention to the even distribution and correct depth of the fuel bed, by adequate rodding or poking, which ensures the regular removal of any clinker formed before it can build up to excessive proportions, and by attention to the B.S.T.

CHANNELLING

Channelling through the fuel bed may be the cause of inefficient working since, as explained above, a vertical channel is formed through which a relatively large proportion of the blast will travel. From this arises a local "hot spot," and it leads to clinker formation.

The uprush of gases through a channel gives insufficient time for chemical reactions to be completed and prevents that intimate contact between blast and fuel which is necessary. Consequently channelling causes high CO_2 content accompanied by a great deal of undecomposed steam, leading to a poor quality of gas.

The remedy for channelling of this type lies in adequate poking.

There is another type of channelling, perhaps better termed irregular blast distribution, due to segregation. If the producer fuel is not closely sized the larger material will tend to roll to the sides of the hopper or producer, and zones (particularly around the edges) will be formed where the fuel bed has a comparatively low resistance. Thus in some experiments made jointly by the British Iron and Steel Federation and B.C.U.R.A., the resistance of a cold producer bed was found to be 4 inches w.g. for a fuel of average size 1·05 inch, and only $\frac{1}{2}$ inch w.g. for the same flow with a fuel of average size 2·94 inches. Obviously if these two sizes become segregated in a producer, the bulk of the blast will pass up the larger fuel and the effect will be to overload that portion of the producer; poor gas, high CO_2, much undecomposed steam and clinker formation will be the result.

One remedy lies in care in handling the fuel and in installing charging arrangements that will obviate segregation. A general remedy is to use fuel of a close size ratio. The size ratio may be thus defined: If the aperture of the smallest test sieve through which 95 per cent. of the coal will pass as the upper limit be a; and the aperture of the largest test sieve on which 90 per cent. of the coal will remain as the lower limit be b, then the size ratio is a/b and should not be greater than 3 or 4.

EFFECT OF DEPTH AND CONDITIONS OF FUEL BED

Generally, increasing the depth of the fuel bed results in markedly increasing the amount of steam decomposed. Thick fuel beds are not always so readily controlled as are thinner beds. Irregularities of feeding, and uneven distribution of the blast can produce channels and leakage of blast round the walls of the producer due to the greater resistance of the thick fuel bed. The results are inefficient operation; the percentage of carbon dioxide increases appreciably and the carbon monoxide falls proportionately; and there is a reduction in the hydrogen content of the gas.

RATE OF GASIFICATION

The rate of gasification is dependent upon the size and type of fuel, the fusibility of the ash, and the degree to which control of the fuel bed is mechanised by the use of mechanical rakes or pokers and grates. In static producers the rates of gasification giving the most consistent operation are of the order of 10–25 lb. per square foot of fuel bed area per hour. Mechanical producers usually operate in the range 30-70 lb. per square foot per hour. Generally at the high rates dust troubles are more frequent, and there is a greater possibility of clinkering troubles.

Essential conditions for the higher rates of gasification for any given design of producer are close grading of the fuel, whether large or small, and suitability of the coal as regards caking character and the refractoriness of the ash.

PERFORMANCE ON COKE

Coke of the size $\frac{3}{8}$–$1\frac{1}{2}$ inches has been gasified satisfactorily at rates of 60 lb. per square foot per hour, though as with coal the lower rates are less conducive to operating troubles, particularly with fuels of low ash refractoriness. Close grading favours the higher rates of gasification. With more than 15 per cent. of fines below $\frac{1}{8}$ inch screen, troubles are experienced with blow holes, channelling and excessive clinker. The maximum rate of ash removal is about 10 lb. per square foot per hour, though in practice much lower rates are generally operated, e.g. $2\frac{1}{2}$ lb. per square foot per hour on static producers. The average thickness of the fuel bed is from 3 to 5 feet.

With a coke of suitable size, $\frac{1}{2}$–1 inch, excellent gas is made with a blast saturation of 53°–56° C. With larger cokes the blast tends to short circuit through the centre of the fuel bed, causing channels and clinkering, requiring additional steam. The texture of the ash can be controlled by the steam, the following being the usual indications :—

Large excess causes fine mealy ash.
Moderate amount gives pea-sized nodules.
Marked deficiency causes large clinkers.

MOTIVE POWER FOR BLAST

Where the air and steam supplies are separately controlled, as in producers equipped with fans or other mechanical blowers, it will be obvious that the blast saturation temperature can be maintained at any desired figure but, where steam-jet injectors are used, the adjustment may be rather more difficult. With steam-jet injectors the air-steam ratio depends on the size and shape of the steam discharge orifice, the steam pressure, the design of the blast pipe and the pressure against which the blast is injected.

Variations due to a variable resistance in the fuel bed particularly affect the quantity of steam used when the pressure is supplied by a steam jet. The higher the pressure, the greater is the quantity of steam injected, thus leading to inefficient gasification. Where steam-jet injection is installed, it is particularly important to use fuel of uniform size which remains uniform over successive deliveries. In a great many plants, particularly in bituminous producers, the ash bed grading is more important than the fuel grading. This can be controlled by the blast temperature and the technique of cleaning. A further method of supplying the blast is the application of the turbo blower, that is to say, a steam turbine drive operating a fan. The exhaust steam from the turbine is used to saturate the blast. Considerable improvement in blast conditions, and economy of steam has been claimed for this method of operation.

EFFICIENCY OF A PRODUCER

According to the British Standard method of testing gas producers (B.S. 995), the efficiency of a producer plant can be expressed in one of three ways :—

(i) Hot gas efficiency

$$= \frac{\text{Potential + sensible heat of gas per ton of dry fuel}}{\text{Total heat per ton of dry fuel}} \times 100$$

(ii) Cold gas efficiency

$$= \frac{\text{Potential heat of gas per ton of dry fuel}}{\text{Total heat per ton of dry fuel}} \times 100$$

(iii) Comprehensive efficiency

$$= \frac{\text{Potential + sensible heat of gas per ton of dry fuel + thermal equivalent of surplus steam per ton of dry fuel}}{\text{Total heat per ton of dry fuel + thermal equivalent of make-up steam per ton of dry fuel + thermal equivalent of energy consumed per ton of dry fuel}} \times 100$$

Each efficiency can be expressed as desired,

(*a*) Taking gross calorific values for potential heat of fuel and gas.
(*b*) Taking net calorific values for potential heat of fuel and gas.

HEAT BALANCE

The cold gas efficiency in the generation of producer gas usually ranges between 63–80 per cent. Radiation and cinder loss is not over 9 per cent. Consequently the sensible heat in the gas is from 11–28 per cent. of the total heat input. About 97–98 per cent. of the total heat input usually comes from the fuel itself, the balance being the heat in the steam contained in the blast.

CONDITIONS FOR EFFICIENT OPERATION OF PRODUCERS

The following summary of conditions favourable for efficient gasification may help to crystallise in a practical way much of the foregoing information.

(*a*) The most important individual factor in efficient gasification in blast producers operating on bituminous fuel is the selection of a suitable coal. The relative properties of coal vary very considerably, not only as between one coalfield and another, but also from seam to seam in a given district, and sometimes even in a particular colliery. Broadly, the properties which are desirable are reasonable uniformity of size, low or moderate caking power and a relatively refractory or infusible ash having a minimum tendency to fuse into clinker or on to the brick-work lining of the producer. Some operators prefer that the coal used shall have a moderate caking power in order to avoid disintegration of the fuel in the producer. A useful survey of the suitability of British fuels for this purpose is given in the *Fuel Economy Review* for 1942, page 4.

(*b*) A correct depth of fuel bed requires that the grate or tuyeres should be covered with a sufficient but not excessive depth of well-burnt ashes, this depth in static producers being probably not less than 6 inches or more than 12 inches above the highest point at which the blast enters the producer, and that this ash bed should support an adequate depth (usually falling within the limits 2 feet 6 inches to 3 feet 6 inches) of fuel in progressive stages of carbonisation and combustion ; both the ash layer and the fuel bed should be of reasonably uniform texture and depth. Wide variations from the above conditions occur in gas machines, depending on the design and operation of the machine.

(*c*) Efficient distribution of the blast requires that there should be an equable flow throughout the entire cross-sectional area of the fuel bed, resulting in a uniform rate of gasification. Difficulties in this respect usually arise either from the formation of clinker fused on to the producer lining and thereby restricting the effective area of gasification or from the formation of a hard and relatively impenetrable central core of coke resulting from the carbonising of fines in the fuel and undue concentrations of fine ash locally in the ash bed.

(*d*) The blast saturation temperature should be maintained constantly at the figure best suited to the requirements of the fuel and plant and should rarely exceed 60° C. and never 62° C. unless there are exceptional reasons to the contrary.

(*e*) The gas pressure in the producer and mains should be maintained at a constant and moderately low figure, which, usually, should not exceed about 1 inch water gauge. An excessive gas pressure causes a material increase in the gas leakage from the poker holes and fuel charging mechanism and, in producers operated by a steam-jet air injector, also tends to increase the difficulty of maintaining a low blast saturation temperature.

(*f*) The maintenance of a reasonably low gas exit temperature is desirable as indicating an equable rate of gasification over the entire fuel bed area and a correct depth of fuel ; whilst, on the other hand, an abnormally high exit temperature always suggests an impoverishment in the quality of the gas, and sometimes combustion of gas above the fuel bed. A low gas temperature occurs with excessively high blast saturation temperature, and low rates of gasification.

PRACTICAL OPERATION OF A PRODUCER

FUEL CHARGING

Hopper charging with manual operation is the usual practice ; mechanical feeding is a general feature of mechanised units. The hopper has a gas-tight lid and is provided with a bottom bell or valve capable of being raised or lowered from the outside to admit the charge of fuel into the hopper and thence to the generator chamber. Some types of mechanical feed are provided with a counter mechanism to indicate the rate of feed. The uniform distribution of the fuel may be assisted by the use of some form of fuel-distributing device which must be adequately protected against damage by heat from the hot gas.

LEVELLING OR POKING MECHANISMS

In rotary producers, in which the body of the machine revolves, the levelling of the fuel bed may be effected by a leveller or rake. In both mechanical and static producers oscillating pokers are also used to ensure even distribution of the fuel. In all these types water cooling is used and accordingly detection of water leakage is important. Further it may be necessary to overcome erosion of the wearing parts by the use of special materials.

THE GRATE OR BLAST DISTRIBUTOR AND ASHING PLOUGHS

Grates vary from a simple construction of straight grate bars to complex mechanical contrivances. Where the grate or " mushroom " is made to rotate, a helical contour is given to the tuyere rings to cause the ash to be removed radially from the ash bed. In the simpler stationary producers the ash is removed from a water-sealed trough by long-handled shovels or from the ashing doors in dry-bottom producers. Mechanical grates may be provided with ashing

ploughs the use of which at stated intervals determined from the indications of the ash-level test, described later, ensures the correct thickness of ash bed above the grate.

Observation of the condition of the clinker removed will ensure the correct use of steam in the blast. An ultimate feature of all gas producer operation is that the operator becomes concerned mainly with clinker trouble. Continuous mechanical extraction of ash gives a gas of more constant composition, and is stated to increase the overall efficiency by about 5 per cent.

LIGHTING UP AND CLOSING DOWN PLANT

For lighting up it is usual to build a fire of kindling material such as shavings and timber, on a bed of suitable ash. The kindling may then be set alight, or alternatively, a layer of 1 foot of fuel may be placed on the top first, and the shavings then lighted. When the bed is alight, the fuel is fed gradually, while the lean gases escape at the blow-off pipe. The newly lighted fire is blown by a gentle blast.

In large coke-fired producers, the kindling materials are built into robust beds in which are contained the full burden of coke. The lighting is then effected from tinder boxes.

Explosions both in producers and furnaces during starting-up can be avoided if it is recognised that an explosion can only occur when a sufficiently rich air-gas mixture has accumulated. The ignition temperature of producer gas (Chapter V) is about 600° C. A furnace can be more safely "gassed" with lean gas.

When it is necessary, as in shutting down, to purge the producer and furnaces, purging should be continued until the air has been dislodged from pockets. With clean gas a satisfactory and simple test is to observe the condition of the flame at a test burner. This method is quite safe provided a proper test burner is used. The flame in the burner shows a double cone, similar to that of a lightly aerated bunsen burner, while there is air in the gas. As soon as the inner cone disappears, the gas may be safely used. With dirty gas a cautious trial at a poke hole will show clearly enough what is the composition of the gases. In burning out or shutting down, the main safeguard is to avoid the possibility of the formation of explosive mixtures.

RUNNING AND CONTROL

In order to put the principles which have been discussed to practical effect, care in operation should be devoted to three essentials. These are :—

(1) Control of the steam supply.
(2) Maintenance of the correct depth of fuel bed by attention to the feed. This is usually indicated by maintaining a constant gas exit temperature. The attention to the fuel bed also involves cleaning off the walls any clinker that may form there.
(3) Efficient removal of the ash, including judicious movement or poking of the fuel bed to prevent the formation of channels and to break up any masses of partly carbonised fuel that may form.

(1) CONTROL OF STEAM SUPPLY

The blast saturation temperature should be maintained constantly at the figure best suited to the requirements of the fuel, the plant and the conditions of gasification. It should generally be kept below 60° C. Higher rates of gasification may require a somewhat higher temperature to be carried in the blast.

(2) CORRECT DEPTH OF FUEL BED

The grate or tuyere should be covered with sufficient but not excessive depth of ashes, preferably between 6 and 12 inches above the highest point at which the blast enters the fuel bed. The hot zone and the distillation zone should also be kept within regular limits by means of tests with the test rods. These should be $\frac{3}{8}$ inch diameter and long enough to reach to the top of the blast tuyere. They are inserted vertically through the poking holes and left in the fuel bed for a fixed interval of time, usually one to two minutes, according to the class of fuel and conditions, and then withdrawn. A red-hot length on the rod will indicate the position of the oxidising zone. This should be sharply defined and only a few inches long. The presence of two or more such zones is an immediate indication of poor fuel bed conditions.

A second rod with a small L-bend at the end may be inserted afterwards to locate the position of the surface of the fuel bed in producers not filled up to the top of the generator.

A graduated rod painted to mark the correct limits should be used to determine the exact measure of the various levels. These should be recorded in the log book or marked in chalk on a board set up on the control stage, together with the times and degree of ash removal, e.g. with mechanical grates the number of revolutions during which the plough has been operating.

The fuel must be fed at a regular rate, otherwise in bituminous coal producers the gas will be of irregular quality. The spread of the coal must be watched, and if there are no mechanical aids to spreading such as deflection plates, the operator may control the spread of the fuel by varying the load of each charge.

One of the hardest and consequently most shirked tasks in producer operation is the poking of the fuel. A light rod may be used for the purpose of levelling.

(3) CLINKERING AND REMOVING ASHES

The avoidance of excessive clinkering is mainly effected by steam control, but if clinker has once begun to form, it may have to be broken up by the use of the heavy poker, particularly if it is building up on the wall of the producer. Clinkering may arise from the formation of channels, and these may have to be filled by poking, but in carrying out this necessary action care should be taken to avoid too much disturbance of the fuel bed.

OBSERVATIONS OF THE GAS TEMPERATURE

The temperature of the gas should be measured in the main and the off-take of the producer. The most reliable means is the use of a base metal thermo-couple. The use of a recorder for the purpose is one of the most effective means of control, since for a given fuel and fixed rate of gasification and blast saturation temperature the temperature of the issuing gas should remain constant. By considering the temperature of the gas in relation to that of the blast and the fuel bed levels as shown by the rod test, a reliable judgment of the condition of the fuel bed can be formed.

The temperature of the hot gas will generally be for bituminous coal between 500°–750° C. and for coke and anthracite, 300°–600° C. Gas temperature variations may result from channelling in the fuel bed or inleakage of air and burning of the producer gas ; an unusually low gas temperature may be due amongst other causes to a high moisture content in the incoming fuel.

GAS QUALITY

The best method of checking gas quality is to make an analysis of the gas. The interpretation of the analysis will be apparent from the earlier discussion

(cf. also Chapter II). Judging the quality of the gas by observation of its character as it leaves the poking holes requires experience tempered with knowledge derived from gas analyses. If, however, the gas from bituminous coal ignites at the poking hole, or appears to be smoky, either the fire is too shallow or channelling may be occurring.

Too high a blast saturation temperature is apparent when the gas appears to be dull and cold, and moisture is deposited if it is brought into contact with the hand. The amount and kind of volatile matter which is yielded from a particular fuel will influence the colour of the gas and therefore the conclusions derivable.

CONTROL

Among essential instruments are those for measuring the temperature of the blast and the gas at the off-take. The maintenance of the pressure of the gas at a satisfactory and constant value requires either the use of automatic control or a readily visible pressure gauge, placed in a prominent and convenient position. The installation of modern instruments is a powerful means of attaining efficiency of operation in all classes of plant, and should be looked upon by the engineer as a profitable form of scientific control. The metering of coal, air, gas and steam can be readily and efficiently applied.

The most desirable instruments are :—

(*a*) Exit gas temperature recorder.
(*b*) Blast saturation temperature recorder.
(*c*) Blast pressure indicator.
(*d*) Gas pressure indicator and controller.
(*e*) On plants blown by live steam, a steam pressure gauge.
(*f*) On plants blown by electric fans, an air flow indicator.

This equipment is sufficient to disclose all that need be known for the day-to-day operation of most plants. On large batteries of producers more elaborate equipment may be installed in control panels, and the refinement of automatic control used.

The blast saturation temperature should be measured by a thermometer placed immediately before any preheat is given to the air-steam blast and with the bulb immersed to a depth at least one-third of the diameter of the main.

The pressure conditions on the plant are of great importance as fluctuations in these readings give indications of faults which would otherwise be unsuspected.

The actual pressure readings will depend upon the size and the design of plant and the characteristics of the fan employed, but a gradual building up of pressures (in clean gas units) may in general be taken to indicate the need for inspection of either the washer-cooler or dry scrubber, and gives an indication that a fresh charge of the scrubbing medium is required.

On producers equipped with gas-cleaning provision, four gauges of the ordinary water U-tube type are suggested and these should be placed in the following positions :—

(*a*) Washer-cooler inlet.
(*b*) Washer-cooler outlet.
(*c*) Dry scrubber outlet.
(*d*) Fan outlet.

GAS ANALYSIS

It is recommended that analysis of the gas should be made from time to time, preferably at least once a day. A suitable gas analysis apparatus for works purposes is the Orsat, though this is primarily of service for rough works

control. Chapters XXX and XXXII should be consulted for further information.

THE PURIFICATION OF PRODUCER GAS

The purification of bituminous gas in all but the smallest quantities presents a more difficult problem than that of coke and anthracite gas and is more rarely undertaken. For coke and anthracite gas the usual equipment consists of a washer-cooler in which the gas is cooled and freed from the greater part of its dust content and almost all of any traces of tar by passing upward through a tower charged with coke and sprayed with water. This may be followed by a filter box packed with wood chippings, wood wool, sisal fibre or some other filtering medium, to remove the last remaining dust. This box may also be charged with iron oxide to remove hydrogen sulphide. A gas-washing fan or other mechanical washer may be substituted for the filter box.

In large plants operating on coke the cooling tower is followed by one or two centrifugal disintegrators, containing rotating blades, which are supplied with a stream of water fed to the machine. Violent turbulence between gas and water is produced, and the latter is separated from the gas by centrifugal action. The " disintegrator," intended mainly to remove dust, is followed by a " spray separator."

The washer-cooler is generally packed with graded coke, 1–2 inches in size, over which water is sprayed amounting to some 25 gallons per 1,000 cubic feet of gas make, or some 4,000 gallons per ton of fuel gasified. This water may be run to waste if water is plentiful, or it may be circulated through a cooling pond. Where the water is re-circulated it must be remembered that it will become acid through absorption of oxides of sulphur from the gas and it must be neutralised with lime to avoid corrosion. The presence of sulphur dioxide in coke producer gas has been established, and evidence of serious corrosion adduced where the alkalinity of the wash water has not been watched.

The loss of pressure across the washer-cooler is kept as low as possible by using a suitable packing. If it becomes high the inference is that a blockage is occurring and the scrubber must be repacked with fresh coke of the right size.

The quantity of water used can be best regulated by placing a thermometer in the outlet gas main and adjusting the water consumption to give the required temperature. To avoid difficulties in the cleaning plant the following precautions are suggested :—

(1) Examine the water sprays regularly.
(2) (*a*) Inspect the water overflow at the seal pots and make sure there is no blockage caused by accumulation of sludge.
    (*b*) In frosty weather take steps to prevent freezing.
(3) Prevent wastage by using just sufficient water to cool and clean the gas effectively.
(4) Take records of the pressures at the inlet and outlet connections of both the washer-cooler and the filter.

Difficulties sometimes occur with gumming or sticking of valves and suction fans on producer plants. This is frequently due to traces of tar escaping from the tar scrubber, the presence of which may be shown by allowing the gas to play on a wetted filter paper. If tar is absent, the cause may be the formation of gum by the action of small traces of nitric oxide formed in the producer on certain unsaturated hydrocarbons (e.g. butadiene and cyclo-pentadiene) formed during the distillation of the coal. This type of deposit would not occur with coke. Some cokes, as has been mentioned before, will contain traces of tarry matter, and it is not good practice to do without the tar removal apparatus when using coke as a producer fuel.

Another cause may be deposits due to salt. The salts are derived from the coal, or they may be equally derived from coke which has been quenched with water containing salts. The use of brackish water in the seals of the producers or in the scrubbers, etc., of the purification plant may also give rise to salt deposits.

In most installations it is not considered necessary to remove sulphur, and unpurified gas containing up to about 70 grains of sulphur per 100 cubic feet is fed direct to the appliances. This practice is justified providing :—

(*a*) The sulphur compounds in the gas do not adversely affect the material being processed.
(*b*) All appliances using large volumes of gas are fitted with flues which carry the products of combustion outside the building.
(*c*) Adequate ventilation is provided where flueless apparatus is in use.

Most fuels recommended for use in gas producers contain not more than 2 per cent. of sulphur and on gasification some 75 per cent. of this appears in the gas as a mixture of hydrogen sulphide and organic sulphur compounds. The hydrogen sulphide content of producer gas is generally between 35 and 60 grains per 100 cubic feet according to the sulphur content of the fuel. With high sulphur fuels sulphur contents as high as 140–150 grains may quite well be encountered.

Where it is found necessary to remove hydrogen sulphide the oxide purification system is used which is already well-established in gas works. The producer gas is passed through layers of solid iron oxide specially prepared for the purpose which is thereby converted into iron sulphide. The purification material is removed from the boxes from time to time, when the air serves to reconvert the sulphide into oxide with liberation of free sulphur. The oxide can then be used again until it contains over 50 per cent. of sulphur when it can be sold for the manufacture of sulphuric acid.

Where producer gas containing tar has to be cleaned it is of fundamental importance that the greater part of the dust is removed before any of the tar vapour is allowed to condense. This is accomplished by providing a large and well insulated dust catcher before the cooler. The most satisfactory design of dust catcher is that involving the principle of the cyclone separator. That is to say, the gas enters the dust catcher towards the top at its widest part. It moves horizontally and tangentially to the inner surface of the dust catcher and leaves through a central off-take passing down the axis of the chamber. The use of baffles has been shown to be of little service, whilst unduly increasing the resistance to flow of the gas. From the dust catcher onward the plant does not differ from those for coke and anthracite gas except that the washer-cooler tower must not be packed with coke. In some plants electrostatic precipitators are substituted for the filter boxes for removal of fine dust and tar fog with considerable success. When cleaning bituminous fuel gas, arrangements must be made to separate the cooling water from liquid tar and to dispose of the tar.

UTILISATION OF PRODUCER GAS

Producer gas may be utilised in two ways. It may be decided to operate furnaces or gas engines on producer gas as a regular practice and in this event the whole installation would be designed to suit the producer gas. On the other hand, producer gas may be used only in an emergency ; producers are installed at many works to-day where town gas is used in order to provide a standby for emergency purposes if war damage should affect the supply of town gas.

Where producer gas is to be used for emergency purposes care should be taken in advance to ensure that the conditions under which it is to be used are known and can be fulfilled. The essential conditions are :—

(1) That the gas mains in the works under the conditions of pressure to be applied are large enough to convey the necessary quantity of producer gas to the furnace.
(2) That the arrangements at the furnace are suitable for burning producer gas.

For all practical purposes, and at low pressures, the pressure required to convey gas through a main is given by Pole's formula (Chapter IX).

$$Q = 1{,}350\sqrt{\frac{d^5p}{sL}}$$

where Q = gas discharged through the pipe in cubic feet per hour.
d = diameter of pipe or main in inches.
p = loss of pressure in pipe between inlet and outlet, in inches w.g.
s = specific gravity of gas (air = 1).
L = length of pipe (yards) 5 feet being added to the measured length for each sharp elbow or tee, and 2 feet for each 90° streamlined bend.

In making the calculation it is to be remembered that the quantity of producer gas to be used will be slightly greater (10 per cent. more would be allowed) than would be given by the expression :

$$\frac{\text{Maximum quantity of town gas (cubic feet per hour)} \times \text{(C.V. of town gas)}}{\text{C.V. of producer gas}}$$

Thus if a works using 5,000 cubic feet per hour of town gas of 500 B.Th.U. per cubic foot has to change over to producer gas of 125 B.Th.U., the quantity of gas now required is $5{,}000 \times 500/125 = 20{,}000$ cubic feet per hour. To this should be added 10 per cent., making 22,000 cubic feet per hour as a margin for the slightly greater amount of producer gas that may be necessary.

To carry the example further, if the existing gas pipes were 5 inches internal diameter and 100 yards effective length taking 5,000 cubic feet per hour of gas of specific gravity 0·4, the fall in pressure would be practically 2/10 inches (0·2 inch) w.g. To carry the equivalent quantity of producer gas (specific gravity 0·9) through the same mains, would require a pressure drop of 7·6 inches w.g.—an increase of 38 times.

If this does not provide sufficient pressure at the burners, a booster, which can be by-passed when on town gas, must be installed in the system, preferably near the gas plant.

In practice, just as the delivery pressure of town gas is frequently of the order of 2–3 inches w.g., so with producer gas plants, it is the practice to ensure a minimum delivery pressure much higher than the figure calculated above, since there are other resistances than those shown in the simple example cited.

The size of mains thus calculated will be reasonably correct for purified gas, though generally a small allowance should be made for unexpected resistances by making the mains a little larger than indicated by this calculation. For unpurified gas a liberal allowance should be made. The attack by unpurified gas on iron services and copper alloy fittings, particularly at the elevated temperatures frequently to be found around furnaces may be serious.

Whether the pressure obtained at the burners will be enough can be calculated from the flow of gas through burner orifices. This is given by the formula :—

$$Q = 1{,}658{\cdot}5\,a\,k\,\sqrt{p/s}$$

where Q = gas discharge, cubic feet per hour.
a = area of orifice in square inches.
k = coefficient of discharge, normally 0·75 for gas burners.
p = initial gas pressure at orifice, inches w.g.
s = specific gravity of gas (air = 1).

The pressure at the outlet of the producer gas cleaning plant is known and is generally between 12 inches and 20 inches w.g. Let it be supposed to be 15 inches w.g. The pressure loss in the mains is 7·6, making (15 — 7·6) = 7·4 inches effective pressure available at the burners.

If the coal gas were fed to the furnaces through 10 similar burners of $\frac{9}{16}$-inch bore, it could be replaced by 2,200 cubic feet per hour of producer gas through each burner, and the pressure then required at the burners would be given by

$$2{,}200 = 1{,}658{\cdot}5 \times 0{\cdot}249 \times 0{\cdot}75 \sqrt{p/0{\cdot}9}$$

whence p = 45 inches w.g.

It would clearly be necessary to install a booster to overcome part of the pressure drop in the mains, and to enlarge the burner orifices so that the required amount of gas can be passed by some 10–15 inches w.g. pressure. In the foregoing example the booster should be capable of increasing the effective pressure at the burners to, say, 12 inches w.g., and the burners should be enlarged to 13/16 inches internal diameter; under these conditions each burner will pass 2,350 cubic feet per hour.

Given a pressure loss of 7·6 inches w.g. and a gas pressure at the producer outlet of 25 inches w.g., there would be left a pressure of the order of 17 inches w.g. on the burners, so that the mere drilling out of the furnace jets would be quite sufficient to ensure satisfactory operation on producer gas without installing a booster, and it is in fact for this very reason that the large majority of makers of these producers have adopted the higher gas pressure referred to above. Lower outlet pressures are only adopted in small units which can be mounted near the point of service.

FURNACE HEATING BY PRODUCER GAS

In most furnaces combustion takes place in refractory burner throats. Because of this restriction of the gases and the close proximity of hot brickwork to the burner no difficulties are experienced with flame stability, and the relative proportions of hydrogen and carbon monoxide present in the gas are immaterial. The composition of the gas, however, determines the setting of the primary and secondary aeration controls and it is therefore important that the composition should be kept as constant as possible. Much of the efficiency of utilisation depends upon the correct setting of the primary air controls and the secondary air dampers. On most furnaces operating on low pressure producer gas at temperatures up to about 1,000° C., it will generally be found advisable to close the air slide at the injector and thus use no primary air. The secondary air dampers should then be adjusted so that the CO_2 content of the flue gases is between 17 and 19 per cent., at which value the atmosphere in the furnace will be oxidising.

Furnaces designed for town gas and working at temperatures up to 1,000° C. will give an equal performance on producer gas provided the heat input is the same, but the heating-up time may be 5 per cent. longer with producer gas.

The maximum temperature which can be attained by any furnace depends upon the flame temperature of the gas used, and the intensity of mixing of gas and air. For producer gas the theoretical flame temperature is approximately 1,700° C. (Chapter V), whereas for town gas the value is some 400° C. higher. It has been found that on a small high-speed furnace without preheating the gas and air a temperature of 1,325° C. can be obtained with producer gas. Above this temperature preheating of air and gas is essential.

The efficiency of some high temperature furnaces heated with producer gas is considerably lower than that of similar furnaces heated with town gas. Comparative tests made on a town gas high-speed steel furnace converted to use producer gas show that the heat input with producer gas is about 25 per cent.

greater than that required with town gas. The heating-up rate is also slower when using producer gas.

Whilst the example of the small high-speed furnace is an experimental fact it is not the practice to operate such furnaces without some degree of preheat, for the reason that in order to avoid an oxidising flame an excess of gas is necessary. A simple form of preheater applied to tool hardening furnaces is effective in giving the required flame intensity. In general furnace practice a much lower minimum limit of temperature of operation on cold gas and air is used. A temperature of 1,050° C. can readily be obtained, but higher temperatures must depend entirely on conditions and particularly on the intensity of mixing and conditions in the combustion chamber. When considering the application of preheat to producer gas it must be realised that preheating of both gas and air is readily practicable, and is facilitated by the fact that the relative volumes of gas and air are very nearly the same. Their densities are also not greatly different so that mixing conditions in the burner under preheat are not likely to interfere seriously with the stability of the flame. The loss of sensible heat in the waste products of combustion for similar furnace temperatures is higher with producer gas than with richer gases, and this conduces to the efficiency of the preheater.

EFFECT OF HYDROGEN

The linear velocity of the ignition wave of hydrogen-air mixtures is immensely greater than for other gases (cf. Chapter V).

Many industrial and domestic gas appliances are heated by burners where the gas flames burn freely in air and do not impinge on any solid object. These are known as "free flame" burners and the stability is particularly susceptible to changes in the hydrogen content of the gas.

Experiments have been shown that with these burners gases with a low hydrogen content tend to lift off the burner head unless fed with a low linear velocity. What this limiting velocity in the burner head should be can be deduced from the following experiments (Table 79) carried out by the Gas Light and Coke Co., London, which were conducted on a range of orifices between $\frac{1}{8}$ and $\frac{3}{4}$ inch diameter.

TABLE 79

	Producer gas				Coal gas
Percentage hydrogen in gas	2	7	12	15	50
Mean velocity of gas in tube in ft./sec. at which the flame lifted from the burner ..	8	17	48	78	134

LITERATURE

Among recent books on the subject of gas producers particular reference is here made to two small practical works, to which the authors of this chapter have been indebted for some of the material here published :—

"The Efficient Working of Gas Producer Plant," being Fuel Economy Monograph, No. 3, issued by the Technical Department of the Federation of British Industries, 1941. (Price 2*s.* 8*d.*)

"Producer Gas Plant for Industrial Purposes," issued jointly, in 1942, by the National Federation of Gas Coke Associations, the British Hard Coke Association, and the South Wales Anthracite and Dry Coal Committee. (Price 5*s.*)

CHAPTER XVIII

FURNACES

The selection of furnaces—Furnace atmospheres—Combustion and heat transmission—Sources of heat loss—Thermal efficiency—Recovery of waste heat—Insulation—The heat losses associated with continuous and intermittent operation—Radiation losses through openings—Other losses—Scientific control—Instruments—Records—Practical hints to operators of furnaces fired by coal, coke and gas, pulverised coal and oil.

THE first function of an industrial furnace is to produce a satisfactory product and other considerations must be subordinated to this. In the past this principle has so prevailed that the study of the fuel efficiency of many furnaces has been neglected. When the cost of fuel is only a small proportion of the total manufacturing costs there is a disposition to regard fuel conservation as unimportant; only in those operations where the cost of fuel is important in relation to the total cost is there the incentive to watch fuel economy closely.

The efficient use of industrial furnaces, therefore, requires the consideration of many factors quite distinct from economy in fuel. Over the many uses to which furnaces are put conditions may be expected to vary widely. Certain principles are, however, common to all furnaces and it is to these principles that attention is primarily directed in this chapter.

The fundamental principles of combustion and heat transmission which have been discussed in earlier chapters are of prime importance in the operation and design of furnaces.

THE SELECTION OF FURNACES

In selecting a furnace the purpose for which it is required and the probable fuel consumption are points of primary importance. Its capacity should be closely related to the probable output since one of the principal sources of heat loss lies in incompletely filled furnaces. It is obviously a great deal more wasteful both in heat and labour to use two furnaces each loaded to less than full capacity to do work for which one would be adequate if operated at maximum capacity.

The cost of the fuel is a continuous charge on the heating operation, usually far outweighing the capital charges. Consequently the use of mechanical aids to operation, control instruments, and means of heat recovery should not be excluded on grounds of capital cost. An apparent high capital cost taken over the life of the furnace may be a considerable economy in the long run.

Other matters for special attention are :—

(1) Robustness of the furnace structure. Besides a mechanically stable framework, good quality refractories and attention to efficient laying of the bricks are essential.

(2) Ease of control. All valves, damper controls and instruments should be situated preferably in a convenient and, as far as possible, central position to give ready accessibility for control purposes. Automatic control, provided that over-elaboration is avoided, is an instrument of efficiency.

(3) Ease of maintenance. This implies the use of readily replaceable parts, particularly those which are likely to be affected by heat or wear, and the means for inspection and cleaning.

(4) Insulation, dealt with later, is to be regarded as an essential feature of almost any type of furnace.

(5) Provision for adequate draught. A chimney is not necessarily the most efficient means of providing draught. Fans should be carefully rated for the duty intended. Flues, dampers and sight holes should be so arranged and constructed as to reduce to a minimum the possibility of air inleakage.
(6) Furnace atmosphere. It may be necessary to heat the goods in the furnace in an atmosphere having a composition that will not injure them.
(7) Recovery of waste heat.

FUEL SELECTION

Fuel selection, which is a highly important matter in furnace management, is discussed in Chapter XXXIV.

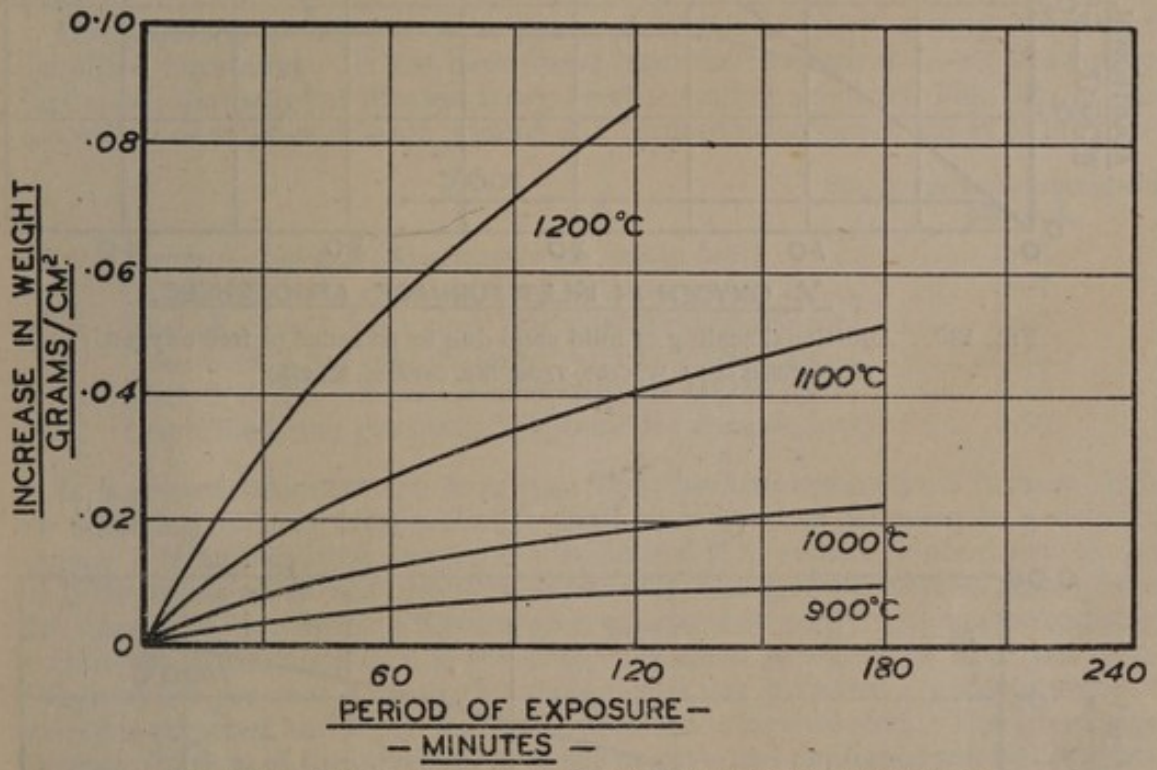


FIG. 129. Rate of scaling of mild steel in a neutral atmosphere.
(H. C. Millett and J. W. Cobb, *Trans. Inst. Gas Eng.*, 85, 672.)

FURNACE ATMOSPHERES

The importance of the control of the atmosphere in a furnace has been widely recognised and this may be the dominant consideration in the selection of a fuel.

(*a*) *Internally Heated Furnaces.* The scaling of metals, particularly the readily oxidised non-ferrous metals, is of particular importance in furnaces in which the products of combustion of the fuel come in contact with the goods in the furnace. Similarly in the steel industry a reduction in scaling during heating could effect important economies. Thus the loss of 1 per cent. by scaling on an annual output of 10,000,000 tons would mean a direct loss of 100,000 tons of steel, valued at over £1,500,000. Steels are heated more than once in manufacturing processes and in addition to the direct scaling losses there are the indirect losses due to additional machining, wasters and roughened surfaces which affect coating operations. Not only does this mean a heavy wastage of steel, but also of the fuel used in making it.

Scaling has long been known to be due to the presence of oxidising gases and is in fact oxidation of the steel which becomes appreciable at temperatures above 700° C. The presence of free oxygen is not essential to scaling because

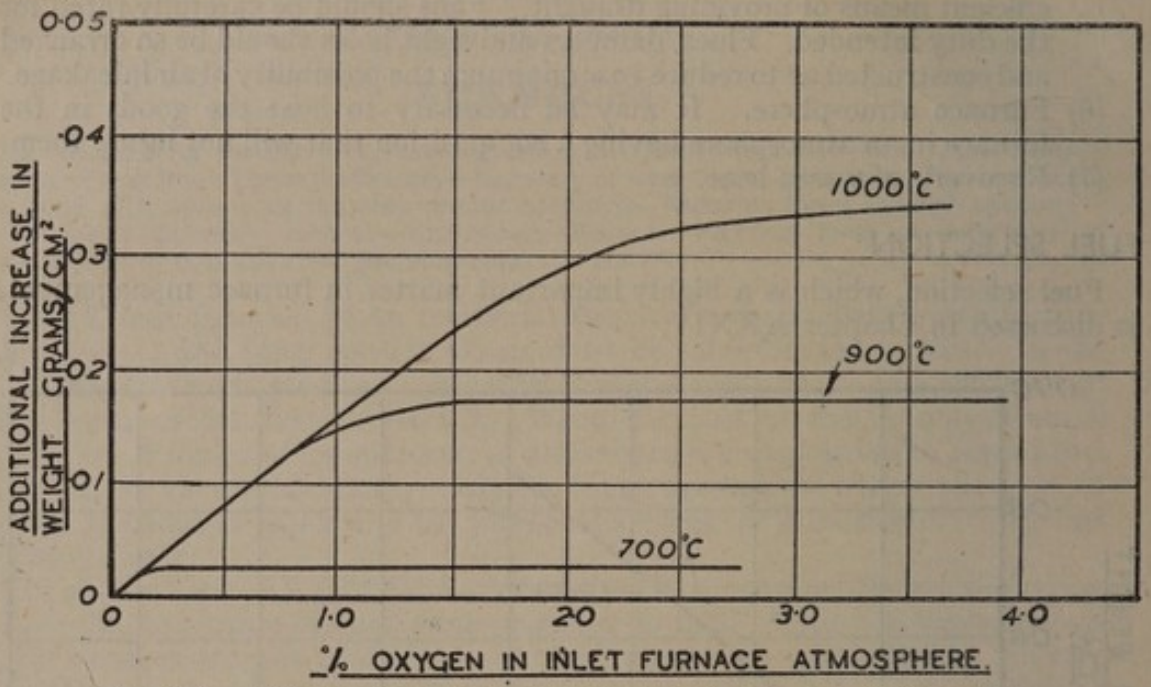


FIG. 130. Additional scaling of mild steel due to presence of free oxygen.
(H. C. Millett and J. W. Cobb, *Trans. Inst. Gas Eng.*, **85**, 675.)

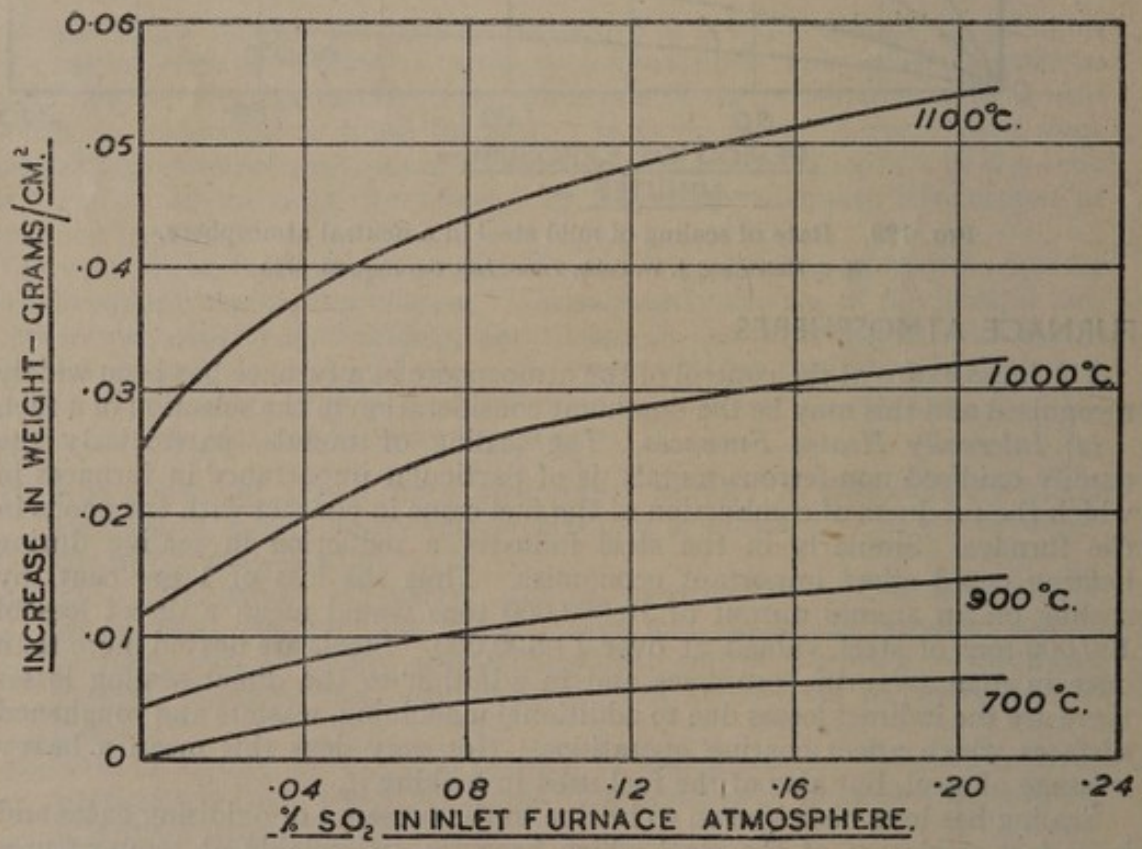


FIG. 131. Additional scaling of mild steel due to presence of free sulphur dioxide.
(H. C. Millett and J. W. Cobb, *Trans. Inst. Gas Eng.*, **85**, 671.)

carbon dioxide and water vapour may both give up their oxygen to iron. Fig. 129 shows the result of scaling experiments at Leeds University (*Trans. Inst. Gas E.*, **85,** 672) indicating the effect of time and temperature in a "neutral" atmosphere, consisting of 10 per cent. CO_2, 20 per cent. H_2O and 70 per cent. N_2. The effect of a small percentage of oxygen to this atmosphere is given in Fig. 130.

If it is desired to inhibit scaling completely it is necessary to add hydrogen, carbon monoxide or methane in such large quantities as to be impracticable. The exclusion of oxygen to the extent that the products of combustion contain comparatively small amounts of carbon monoxide and hydrogen is sufficient so to reduce the oxidising effect as to minimise considerably the scaling of the steel.

The presence of oxygen or oxygenated bodies is not the only important condition for scaling. It has been found from the research at Leeds University just cited, that sulphur dioxide is a powerful scaling agent (cf. Fig. 131). The quantities of sulphur dioxide present in a furnace atmosphere are as follows :—

	SO_2 in furnace atmosphere per cent.
Purified coal gas (30 grains S per cubic foot)	0·01
Coke with 1 per cent. S burnt with no excess air	0·07
Fuel oil, S = 1·5 per cent. burnt with no excess air ..	0·09
Coke with 2 per cent. S burnt with no excess air	0·14
Coal, S = 1·5 per cent. burnt with no excess air	0·15
Unpurified coal gas (600–700 grains S per cubic foot) ..	0·20

It is sometimes necessary to ensure that the atmosphere in a furnace shall be oxidising and sometimes that it shall be neutral or reducing to a definite extent. If an oxidising atmosphere is desired it is evidently necessary to use as little excess air as will complete combustion and give the desired atmosphere (cf. Chapter IV). When adjusting an appliance heated by coal gas to oxidising conditions, if the appliance is governed it is usual to adjust to 10·5* per cent. CO_2 and 1·5 per cent. O_2; if the appliance is not governed a wider margin of safety is required and 9 per cent. CO_2 should not be exceeded. The advantage in lower flue loss of the governed over the ungoverned appliance will be apparent and this is an indication of the desirability of expending money on control.

When the furnace atmosphere is to be adjusted to reducing conditions, as for steel hardening or similar processes, the primary and secondary air controls are regulated in conjunction with the flue damper to give a faint curtain of flame as the furnace door is opened. A piece of wood or paper thrown into the furnace should char, but not flame. This practical test indicates the absence of any appreciable amount of oxygen but gives no indication of the amount of CO present. Gas analysis should be used for this purpose and should show some 1–2 per cent. of CO present according to the process. Each 1 per cent. of CO produced by incomplete combustion is accompanied by 1–2 per cent. of hydrogen so that the potential heat loss in the unburnt gas is likely to be of the order of 7 per cent. for each 1 per cent. of carbon monoxide contained in the flue gas.

So-called reducing atmospheres can therefore be obtained by so adjusting the quantity of secondary air admitted to the furnace that there is produced a small quantity of carbon monoxide in the furnace gases, the amount that is present being determined by a simple form of gas analysis apparatus. Particular attention is directed to this fact because experience shows that many furnace operators seem to regard it necessary to pass large volumes of uncon-

* This figure will vary somewhat according to the composition of the coal gas.

sumed gas through the furnace under the impression that only so will the atmosphere be reducing.

It has been established that carbonaceous particles such as smoke, exert little or no reducing action in a furnace. In order that any substance shall exert an effective reducing action it is clear that it must be brought into contact with the oxidisable materials in the furnace. The molecules of a gas are in a

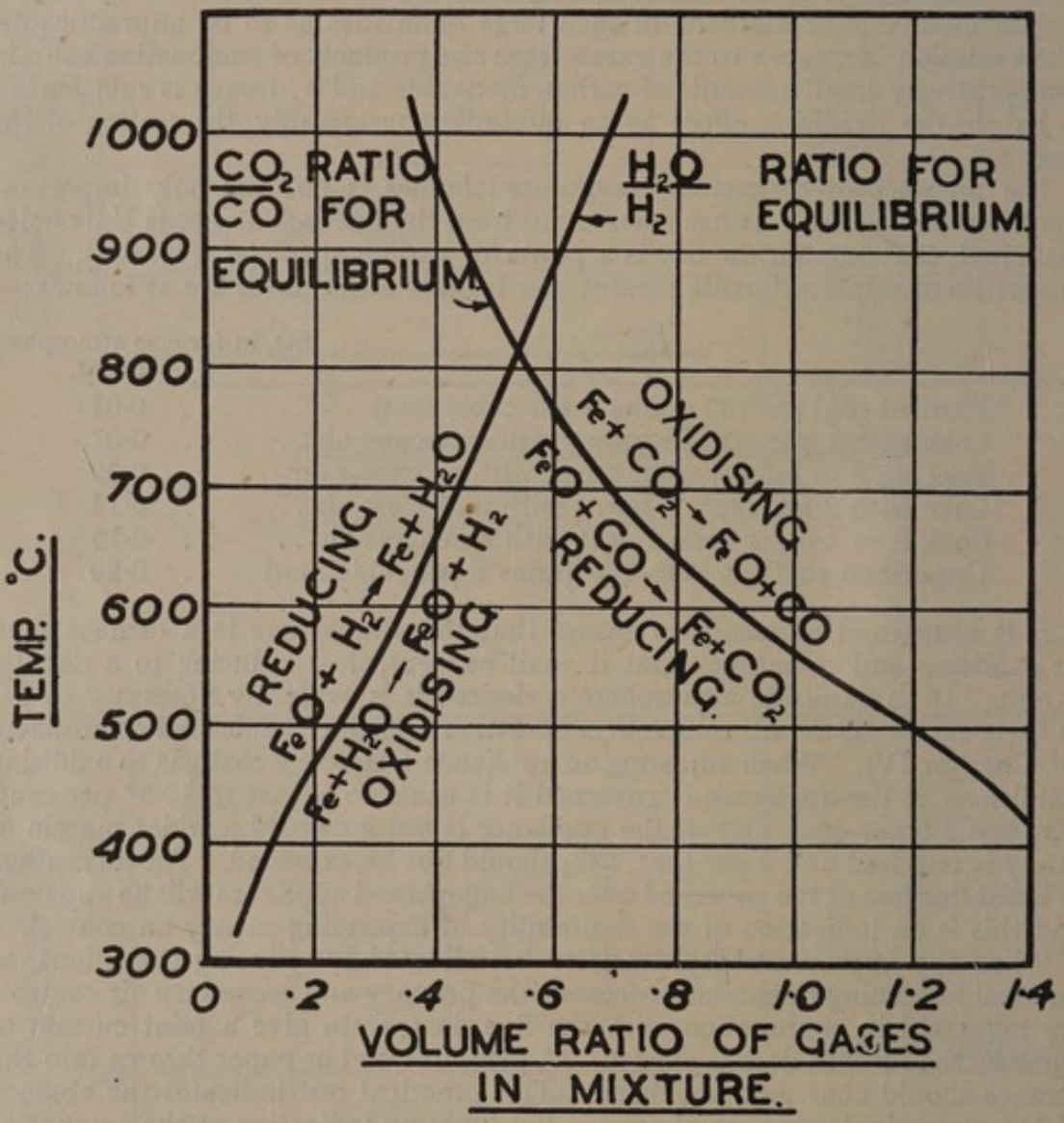


FIG. 132. Equilibria curves for reactions between steel and gases arising from combustion. The reactions printed on the left-hand side of the curves take place under the conditions existing on that side of the curves until the gas composition becomes that given by the curves; similarly for the reactions printed on the right-hand side, until the equilibrium composition of the gases in contact with the steel given on the curves is attained. Thus for a reducing atmosphere at any given temperature the gas composition must be maintained below the CO_2/CO or H_2O/H_2 ratios given by the respective curves for that temperature.

(Data due to Marshall and to Jominy and Murphy—reproduced from *G.E.C. Journal*, XII, No. 1.)

state of very rapid motion (the molecule of hydrogen, for example, at atmospheric temperature travels at the rate of 1 mile per second) and thus will come into contact with adjacent surfaces more freely and more frequently than particles of carbon which are conveyed through the furnace by the slow and comparatively gentle convection currents. Contrasted with the motion of gases smoke particles are relatively almost stationary.

(*b*) *Externally Heated Furnaces.* The bright annealing of metals is another process for which a carefully controlled reducing atmosphere is peculiarly essential. The primary object of most heat treatment operations is the attainment of a metallic structure having certain desired physical properties; the deleterious effect of the furnace atmospheres on the surface of the metal being

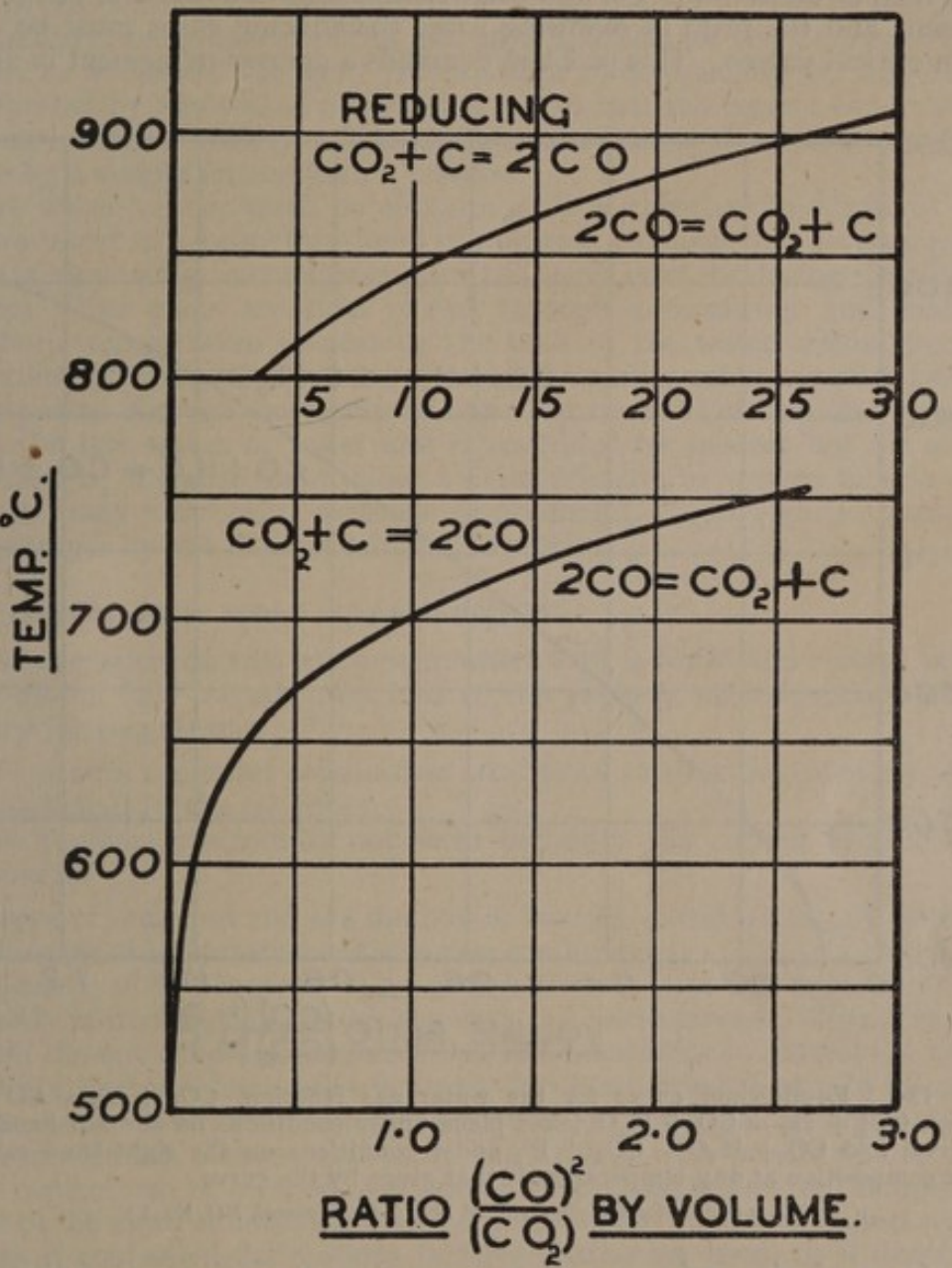


FIG. 133. Equilibria curves for the producer gas reaction, $CO_2 + C = 2CO$. Reaction $CO_2 + C = 2CO$ takes place under conditions given on the left-hand side of the curves, and $2CO = CO_2 + C$ under conditions on the right-hand side, until the gas composition at any temperature is that given by the curve.

(Upper curve due to T. F. E. Rhead and R. V. Wheeler and the lower curve to Marshall—reproduced from *G.E.C. Journal*, XII, No. 1.)

treated has been decreased for the relatively cruder operations of internally heated furnaces in the manner just described. One of the objectives of bright annealing is to eliminate scaling, decarburisation and other chemical reactions usually due to sulphur or oxygen which impair the surface of the metal. The problems of bright annealing vary with each metal used and the composition of the furnace atmosphere may have to be different for each metal.

There are very few metals or alloys which cannot be bright annealed, but with some the problem is more difficult than with others. For stainless steels with a high percentage of chromium the atmosphere must be completely free from oxygen and from oxygen-containing gases. The immunity from attack of this class of metal depends on an invisible film of oxide which would be broken up by further oxidation. For the bright annealing of steel, too, oxygen must be absent and the ratio of oxidising gases to reducing gases must be outside certain critical values. This problem demands a greater refinement in arrange-

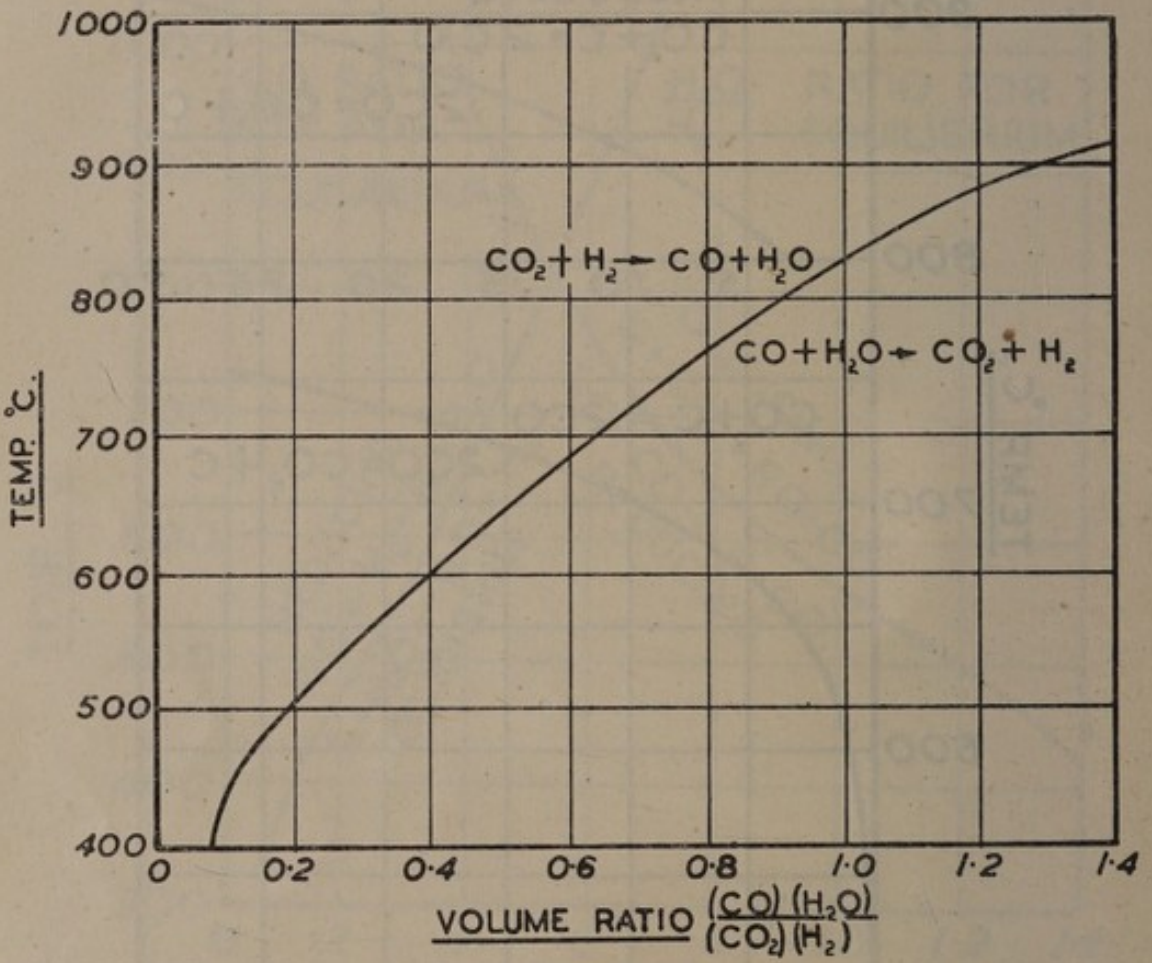


FIG. 134. Equilibrium curve for the water gas reaction, $CO_2 + H_2 = CO + H_2O$. Reaction $CO_2 + H_2 = CO + H_2O$ takes place under conditions on the left-hand side of the curve, and $CO + H_2O = CO_2 + H_2$ under conditions on the right-hand side, until the gas composition at any temperature is that given by the curve.

(Data due to Marshall—reproduced from *G.E.C. Journal*, XII, No. 1.)

ments for producing a furnace atmosphere than the scaling of ordinary mild steel previously discussed.

The necessary exact control of the atmosphere cannot be attained if the combustion gases circulate in the furnace. The furnace must be heated indirectly and the gases that are to constitute the furnace atmosphere must be produced separately in an ancillary apparatus.

Space does not permit of a detailed discussion of this problem here, but the oxidation of bright steel is preventable by keeping the ratios: CO_2/CO; H_2O/H_2; $(CO)(H_2O)/(CO_2)(H_2)$ and $(CO)^2/(CO_2)$ above or below certain limits, the values of which depend on the temperature, see Figs. 132, 133 and 134. A detailed discussion of this subject is to be found in the literature.

In general there is now no difficulty in partially burning gas so that the

resultant atmosphere satisfies the required conditions of composition. It is evident that the most careful control must be exercised over the combustion of gases in a furnace when the products can be affected by the furnace atmosphere.

(*c*) *Burners for Production of Furnace Atmosphere.* Special burners have been patented for this purpose in which gas and air are fed into the system by power-driven blowers with vernier proportioning control. These mixers govern both quantity and quality of the gas, and will control the CO_2 content for example, to within 0·5 per cent. In this type of equipment, the unit may be pre-calibrated by analysis of the purging gas so that the correct conditions for bright annealing a variety of materials can be secured without trouble in practice by a simple setting with a pointer.

Where water vapour must be eliminated, it is absorbed in silica gel. The gases are burnt in a refractory-lined combustion chamber and give up part of their heat by passing in counter-current to the air used for drying the silica gel chambers. The gases are then passed through a condenser and cooled to atmospheric temperature to remove the bulk of the water arising from the combustion of hydrogen. They are then dried by silica gel in a special chamber, and are passed through oxide of iron boxes for removal of sulphur. Silica gel absorbs the last traces of water and is revivified by passing hot air over it. The furnace is, of course, kept under a slight pressure to prevent ingress of air. There are many alternative methods of producing the necessary atmosphere, as for example by the decomposition of ammonia.

COMBUSTION AND HEAT TRANSMISSION

In the operation of any equipment where fuel is burnt the correct proportioning of the fuel and air, and also of the primary and secondary air, are necessary for two reasons :—

(1) To ensure optimum combustion conditions so that the quantity of fuel used shall be the minimum.
(2) To produce continually and with certainty the correct furnace atmosphere.

The type of fuel used and the method of burning it will be largely controlled by the degree of importance of these two conditions.

(*a*) *Control of Combustion.* The combustion of coal has been dealt with extensively in earlier chapters dealing with the boiler furnace. From what has been said there it will be appreciated that it is impossible to control the furnace atmosphere by hand firing solid fuel and that hand firing is only to be used in furnaces where fine control of temperature or atmosphere is not required.

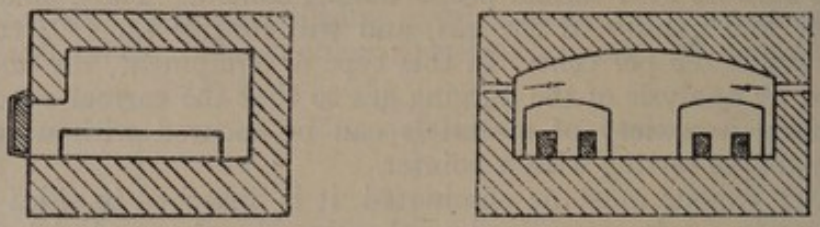
With mechanical stokers, and more particularly with powdered fuel firing, greater control can be secured over both furnace atmosphere and temperature because of the more uniform admission of fuel and air. Solid fuel is less suitable than gas if controlled fluctuations in temperature are used, or if exact temperature control is required. The use of hot crude producer gas is susceptible of greater control than solid fuel and is approached by powdered fuel firing in this respect. For certain operations powdered fuel firing has the disadvantage of introducing all the ash from the coal into the furnace whereas in crude producer gas the content of inorganic dust is very much less.

For exact control of combustion conditions it is necessary to use cold, cleaned producer gas or coal gas both of which can be valve controlled and can thus be used to produce not only the required temperature but also the required furnace atmosphere.

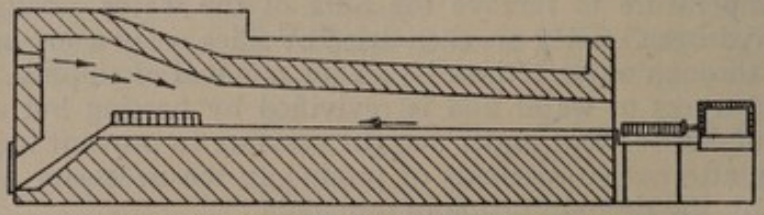
(*b*) *Control of Flame.* The general principles of combustion have already been described in earlier chapters and no further reference to them need be made here. In furnaces it is necessary to control the temperature, and the method

by which the goods are heated. Furnaces can be subdivided into two general types :—

(1) Furnaces indirectly heated (e.g. muffle furnaces) in which the heat is generated in flues or in a separate combustion chamber and must pass through a refractory or other material forming the furnace wall into the



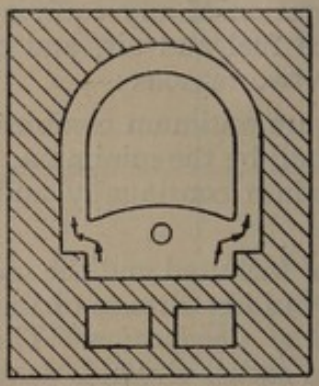
BATCH FURNACE.



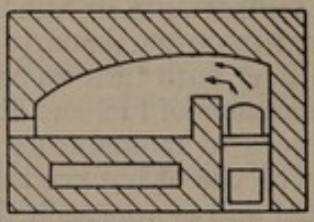
CONTINUOUS FURNACE WITH END DISCHARGE.



OVER-FIRED FURNACE.



MUFFLE FURNACE.



COAL-FIRED FURNACE.

Fig. 135. Types of furnaces.

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heating chamber proper in which the goods are placed. The products of combustion do not come in contact with the goods.

(2) Directly heated furnaces in which the flame may be developed within the chamber in which the stock is placed or in a separate furnace, but in either event the products of combustion come into contact with the goods.

In the first type of furnace the principles of combustion and heat trans-

mission which have previously been established in earlier chapters hold without modification. The stock is to be heated to a certain temperature and according to this temperature the temperature in the flues is regulated by the total quantity of fuel burnt. The objective must evidently be to burn the fuel with the maximum efficiency and to pass the maximum quantity of the heat generated through the furnace wall. The heat is generally transmitted to the stock partly by convection and partly by radiation from the interior surface of the furnace walls.

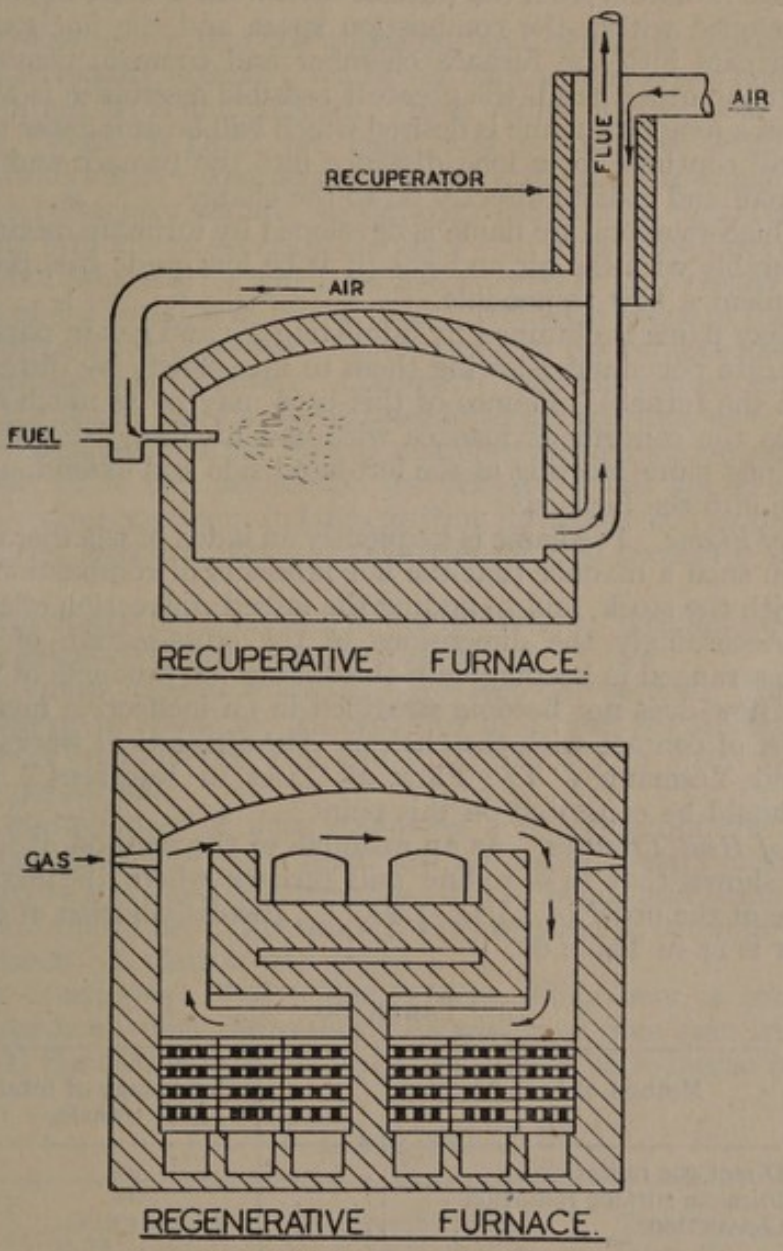

Fig. 135. Types of furnaces.

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The directly heated furnace involves all methods of heat transfer—radiation from the roof and sides of the furnace ; convection and conduction from the hot gases flowing through the furnace ; and gas radiation from CO_2 and H_2O in the heating gases. A number of furnaces are sketched diagrammatically in Fig. 135 (pp. 384/5) which will illustrate these general types.

Attention has been previously directed to the fact that flames can be chilled and caused to deposit carbon by contact with cold surfaces. If, therefore, a

flame from the volatile products of coal or from coal gas is allowed to impinge upon cold stock in the furnace, smoke will be produced and soot will be deposited on the stock. Any combustible, even the constituents of gaseous fuels or oil, can be chilled in the same manner. Moreover, heating by flame impact would frequently be too fierce and would spoil the material heated. The flame must generally therefore be so controlled and directed that it does not impinge directly upon the stock.

If the heat is to be transmitted primarily by convection and gas radiation aided by gentle radiation from the furnace brickwork a short high-temperature flame is developed within the combustion space and the hot gases are then allowed to stream into the furnace chamber and come in contact with the stock. If, on the other hand, the greatest possible assistance is to be derived from radiation a long lazy flame is desired which will be at a lower temperature, but which will continue for a long distance into the furnace and so will heat the furnace roof and radiate directly on to the charge.

The short high-temperature flame is developed by intimate mixing of the gas and air, preferably with the air and gas (if it be low grade gas) preheated and with as turbulent a flow as possible.

The long lazy flame is obtained by admitting air and gas in parallel streams through separate ports and allowing them to mix slowly by diffusion as they pass through the furnace. Flames of this type may be as much as 30 feet in length. Thus the continuous furnace with end discharge shown in Fig. 135 may have a long flame starting at the left-hand side and extending a considerable distance into the furnace.

(*c*) *Depth of Flame.* The flame is frequently an index of whether the hot gases are flowing in such a manner that the hot products of combustion are coming in contact with the stock, and so adding the forced convection effect to that of radiation. Accordingly the dimensions of the cross-section of the furnace require to be arranged in keeping with the volume of products of combustion, whereby the flow does not become stratified in an ineffective manner such as to keep it out of contact with the charge. The theoretical works of Groume-Grjimailo and Yesmann ("The Flow of Gases in Furnaces" by Groume-Grjimailo) should be consulted on this point.

(*d*) *Mode of Heat Transfer.* As an example of the method of heat transfer, it has been shown that in a rolling mill furnace where the maximum temperatures are of the order of 1,100°–1,300° C., the distribution of the modes of heat transfer is as in Table 80.

TABLE 80

Method of heat transfer	Percentage of total heat transfer
Direct gas radiation	54
Furnace surface radiation	34
Convection	12

(*e*) *Size of Combustion Chamber.* The rate at which fuel can be burnt determines the size of the combustion chamber. The condition of the fuel and the temperature of preheat together with the method of mixing have an important bearing upon the dimensions. Some good practical rules are as follows :—

(1) Poor mixing of fuel with cold air and poor utilisation of combustion space :
Approximately 1·5 B.Th.U. per cubic foot per second.

(2) Reasonably good mixing of fuel with preheated air and fair utilisation of combustion space :
Approximately 3–4 B.Th.U. per cubic foot per second.
(3) Good mixing of fuel and preheated air and good utilisation of combustion space :
Approximately 6–10 B.Th.U. per cubic foot per second.
(4) Very fine atomisation of fuel and perfect mixing with highly preheated air, with excellent use of combustion space :
Approximately 10–20 B.Th.U. per cubic foot per second.

Experience has shown that for annealing and reheating furnaces generally, condition (2) is usually attained, whilst for high temperatures such as melting furnaces condition (4) is not unusual. Much higher intensities are practicable, for example, 70–190 B.Th.U. per cubic foot per second, with the type of heating known as surface combustion, in which the gases are made to unite in contact with a refractory surface, accelerating combustion by surface catalysis. A burner firing into a hemispherical refractory cavity has been developed in the United States of America, for which far higher rates of heat release are claimed. The problem of the location of burner and combustion space in these examples is resolved into one of the mode of heat transfer in the furnace itself.

(*f*) *Draught and Gas Velocities.* Conjointly with the estimation of the heat release from the flame, there must be considered the question as to whether the gases can be evacuated from the furnace chamber without undue loss of draught. An upper limit of 30 feet per second for velocity of the hot gases is usually required for this reason except in large melting furnaces, such as steel and glass-making furnaces, where higher gas velocities may be necessary to give quick melting.

For practical design the rates of flow are important. The following velocities are useful in calculating flue and valve sizes :—

Cold air under natural draught, 5–10 feet per second. Air or gas for temperatures up to about 500° C. (940° F.) 10–15 feet per second.

Products of combustion from 750° C. (1,380° F.) to 1,250° C. (2,280° F.) 15–25 feet per second.

When calculating sizes of flues and ports and laying out the design, care must be taken to ensure that there are no sudden restrictions where the velocity has to be increased considerably, and all bends and corners should be eased as much as possible. This subject was discussed in detail in Chapter IX, where a specimen calculation was given.

(*g*) *Rate of Heating-up Stock.* The rate at which heat is transmitted to materials depends on their diffusivity. As will have been seen from Table 36 (Chapter VIII) the thermal conductivity of metals is high, but so is the specific

TABLE 81

Metal	Conductivity, B.Th.U./sq. ft./° F./hr./ft. thickness k	Specific heat, B.Th.U./lb./° F. over range 0–600° F. s	Density lb./cu. ft. ρ	Diffusivity sq. ft./hr. $\frac{k}{\rho s}$
Copper ..	220 (200–1,000° F.)	0·11	558	3·58
Aluminium ..	108 (at 600° F.)	0·24	168	2·68
Brass ..	66 (at 600° F.)	0·092	530	1·37
W.I. ..	35 (at 200° F.)	0·12	458	0·64
C.I.	26 (at 200° F.)	0·13	442	0·44
Mild steel ..	25 (at 200° F.)	0·115	488	0·45
Firebrick ..	0·6 (at 600° F.)	0·23	125	0·024

gravity. The value of the diffusivity compared with that of refractory materials is high. Thus, expressing the diffusivity as the rise in temperature produced in 1 cubic foot of the material by 1 B.Th.U. per hour, the diffusivities of several metals are as in Table 81.

Approximate practical rates of heating have been given for iron and steel (Webb, *Trans. Inst. Chem. E.*, 1938), as follows :—

Heat treating	30	lb./sq. ft./hr.
Annealing	60	,, ,,
Forging	80	,, ,,
In continuous furnaces	50–100	,, ,,

All the foregoing figures depend on the thickness of the stock.

The figures refer to a hearth of which the area is completely covered by the material being heated. For large sized pieces useful figures for low carbon steels are 16–20 minutes per inch thickness and for high carbon steels 30–35 minutes per inch thickness.

As an example of the use of these figures (quoted by C. Webb, loc. cit.), let it be assumed that bars 2 inches square, 6 feet long, are to be annealed and an output of 30 bars per hour is required. The weight of the bars is 81·6 lb. each ; therefore the output is 2,448 lb. per hour. Taking a heating rate of 60 lb. per square foot per hour the area required is 40·8 square feet. As the bars are 6 feet long, a hearth width of 6·8 feet will be required.

For a continuous furnace, for 2 inch square bars, 6 feet long, to give an output of 100 bars per hour, the heating rate with a well-designed furnace would be about 90 lb. per square foot per hour : the length of the furnace would be 15·2 feet.

As an example of heating a large round ingot, say 20 inches diameter by 6 feet long, the maximum figure above should be taken, namely, 20 minutes per inch, giving 400 minutes for heating. This would apply if the ingot were heated on its side. If the ingot were placed vertically, as in a soaking pit, so that the heat could penetrate from all sides, the heating rate could be taken as 16 minutes per inch, reducing the time to 320 minutes.

The rate at which non-ferrous metals can be heated compared with iron and steel is proportional to the following fraction :—

$$\frac{\text{Conductivity of steel} \times \text{specific heat of non-ferrous metal}}{\text{Conductivity of other metal} \times \text{specific heat of steel.}}$$

To obtain the size of furnace required, divide the lb. per square foot per hour for steel by the value of the fraction. This rule does not apply to metals of high conductivity such as copper and aluminium, which can be heated at rates giving approximately 55 per cent. to 65 per cent. and 30 per cent. to 35 per cent. respectively of the times for heating steel.

Although the rate of heating depends upon the diffusivity as mentioned, it must be confessed that heating rates are in the present state of knowledge based on empirical rules, arrived at as the result of practical experience ; as examples there are the " practical " rules (1) that heat penetrates ordinary carbon steel at the rate of $\frac{1}{8}$ inch in five minutes ; (2) that highly alloyed steels, which have generally a lower thermal conductivity, require at least one hour per inch, and that whilst heating must be slow up to 500°, it can be speeded up beyond this point. Another basis is to relate the times of heating to the output on a given hearth area ; thus practical figures quoted by the steel industry are 30 lb. per square foot per hour for heat treating, 40 lb. for general rolling and forging of large blooms, and 60–80 lb. for carbon steel billets below 4 inches in linear section. These rules are not altogether in agreement with the figures quoted

from C. Webb's paper. Some further discussion on this subject will be found at the end of Chapter XIX.

Recent investigation has shown that with metals more depends upon the manner in which the heat is distributed to heat the furnace walls and bring them to temperature uniformity than upon the thermal constants or dimensions of the charge.

The practical points of control in arriving at efficient working are as follows :—

(1) The effect of different methods of disposing the charge should be tried until that which gives the maximum output per unit time is obtained.

(2) The mass of the charge should be varied, if the operation admits of such a procedure, until a point at which an increase of weight per unit area of hearth either disturbs the final desired distribution of temperature at the finishing point or the rate of output becomes reduced. Generally the limiting factor is determined by the nature of the operation, and under-charging is more common than overcharging. The ideal hearth area for a specific duty should be sought. Much fuel can be wasted in wrong-sized furnaces.

(3) The rate of firing should be changed experimentally until the optimum conditions are found, and then valves and dampers should be marked to show the ideal settings.

SOURCES OF HEAT LOSS IN FURNACES

Fig. 136 derived from Trinks' " Industrial Furnaces," summarises the movement of heat in a furnace. The heat is generated in the combustion chamber

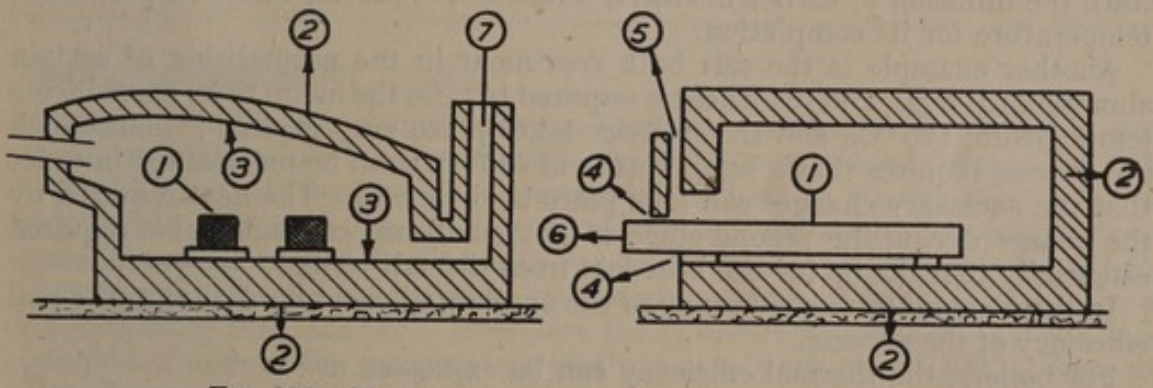

FIG. 136. Heat transmission and heat losses in furnaces.

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at the left, and travels from there to the right. The passage of heat into the stock, as indicated by arrows (1), is desired. But heat also goes elsewhere ; some of it passes into the furnace walls and some into the hearth, increasing their temperature as indicated by the arrows (3). Another portion of the heat is lost to the surroundings by radiation and convection from the outer surface of the walls, or by conduction into the ground (2). Through cracks or other openings, heat radiates away (4) ; and furnace gases pass out around the door (5), frequently burning in the open, and carrying off heat. Then there are the special losses, peculiar to certain types of furnace : for example, in furnaces which heat only part of a long piece of metal, heat passes out along the metal from the part which is in the furnace to the part which is outside, and is there dissipated to the surroundings (6). Radiation from the exposed liquid surface of salt baths or lead baths, and conduction of heat through the terminals or electrodes of electric furnaces, are other cases in point. Water cooling of skid pipes and of conveyor rollers absorbs large quantities of heat and lowers thermal efficiency. If the charge is heated in containers or on travelling

chains, a large part of the heat is dissipated in the open by these devices, after they have been brought up to furnace heat. Finally, heat passes out with the products of combustion (7), either in the form of sensible heat or as undeveloped heat of combustibles escaping unburned (incomplete combustion).

THERMAL EFFICIENCY

The term thermal efficiency as used for boilers is not necessarily applicable to furnaces because of the different purposes for which heat is used. In a boiler, heat is applied to water for the purpose of converting it into steam which is used in that condition. In a furnace, the heat is given to a substance not for the purpose of raising its temperature *per se*, but in order that certain chemical or physical changes shall take place in the material.

In the burning of bricks for example it is not only necessary for the material to be raised in temperature to, say, 1,500° C., but a certain time must be taken over each of the stages in order that (1) the shapes may be dried without cracking, (2) carbonaceous matter contained in them may be oxidised, and (3) that clay may be converted into mullite, or (with silica bricks) that quartz may be converted into tridymite and cristobalite. The process requires that the temperatures shall be attained slowly and shall be held for a period which may be measured in days. Thus the term " thermal efficiency " as defined in a straight heat transmission process is not necessarily applicable to furnaces.

In the carburising of mild steel the useful heat employed in raising the steel to carburising temperature is only a part of the process ; during by far the greater period when the steel is in the furnace there is no heat flow to the steel, since the diffusion of carbon in steel is a reaction requiring time at a particular temperature for its completion.

Another example is the salt bath treatment in the normalising of certain aluminium alloys. But little heat is required to raise the metal to its normalising temperature, 495° C., and the transfer takes place very rapidly ; thenceforth the process requires that a temperature of 495° C. shall be maintained in order that the necessary changes can take place in the metal. The heat required by the charge during this second stage is very small indeed, and fuel is required essentially to make up for the heat loss from the salt bath.

In these examples the efficiency of the appliance bears no relation to the real efficiency of the process.

For boilers, the thermal efficiency can be expressed as :—

$$\text{Boiler efficiency} = \frac{\text{heat in steam}}{\text{total heat input}}$$

$$= \frac{\text{Total heat input} - \text{flue loss} - \text{surface loss}}{\text{total heat input.}}$$

But, the efficiency of a furnace must usually be related to the real work done as expressed in the term :—

$$\text{Real efficiency} = \frac{\text{total fuel consumption}}{\text{quantity of work heat treated.}}$$

Thus the expression thermal efficiency as a general criterion for furnace operations is meaningless and each operation must be considered on its merits. It is for this reason that thermal efficiencies in furnace operation are often very low, not because of inefficiency, but because of the characteristics of the process.

Tihs, however, does not mean that there is not a great deal of inefficient operation in furnaces and the low heat efficiencies often obtained can be very materially improved by making use of the principles of fuel technology.

HEAT BALANCES

As examples of the heat balances of furnaces and as indicating the distribution of the heat, there are reproduced in Table 82 details of heat losses from typical furnaces.

TABLE 82. DISTRIBUTION OF HEAT LOSSES IN TYPICAL FURNACES

Type of furnace	Ceramic kiln	Producer gas fired recuperative billet heating	Recuperative coal fired forging	Non-continuous regenerative gas-fired heating	Oil-fired forge	Coke-fired forge	Coal-fired annealing	Recuperative gas-fired reheating
Source of loss.. ..								
1. Useful heat transferred to charge ..	14	23	16·1	40·9	9	16·1	18	48·7
2. Combustion and chimney losses								
(*a*) Unburnt combustible	3	2·6	3·0	2	3	5	5	1·0
(*b*) Sensible and latent heat in waste gases..	50	22·4	47·6	26·7	47	52·3	45	3·9
(*c*) Gas generation loss in producer gas fired furnaces	—	15·0	—	—	—	—	—	—
3. Furnace structure in brickwork and heat lost to surroundings..	33	37	33·3	30·4	41	26·6	32	46·4

UNAVOIDABLE HEAT LOSSES

As in boiler plant so also in furnaces there are certain heat losses which may be taken as unavoidable. Reference has been made to some of these in Fig. 136.

(*a*) There must inevitably be a certain loss of heat in the flue gas. After all industrially possible has been done to extract the useful heat from these gases there are certain losses which are inevitable. If a chimney is used for creating the draught the requisite temperature must be retained in the gases for operating the chimney. Even if a fan is used for operation it is still not possible to reduce the temperature of the gases below about 300° F. because there is no way of making use of low grade heat usefully. The latent and sensible heat of the water arising from combustion and (with wet material) evaporated from within the furnace must also be lost.

(*b*) However excellent the insulation of the furnace may be there must inevitably be a small escape of heat through the walls to the atmosphere.

(*c*) An important source of heat loss is that arising from the sensible heat of the products in the furnaces. In some continuous processes, as for example in continuous gas retorts and continuous brick kilns, it is possible to recover the heat in the products by using them to produce steam or by circulating air round them which is subsequently used for combustion. In a great many instances, however, as for example in rolling mills, the purpose of the heating operation is to raise the stock to a high temperature at which it is to be used for some succeeding process. In that event the heat taken away from the furnace in the sensible heat of the stock cannot be recovered.

A borderline example is that of the coke in by-product coke ovens. This is discharged at 1,000° C. and represents something like 40–45 per cent. of the heat put into the flues to effect the carbonisation, but generally it is lost when the coke is cooled with water ; in some installations this heat is recovered by a

dry quenching plant in which inert gases are circulated through the hot coke until it is cooled and are then circulated through a waste heat boiler so that something like 900–1,000 lb. of steam can be raised per ton of hot coke. This, however, is not a pure problem in heat recovery but depends on the economics of the process.

(*d*) If gas producers are used in connection with the furnaces there must also be a certain thermal loss due to the operation of this section of the plant. A cold gas efficiency of some 60–75 per cent. or hot gas efficiency of 70–85 per cent. may be expected in efficient producer practice.

AVOIDABLE HEAT LOSSES

There are several avoidable losses to which particular attention must be devoted. These will be the subject of succeeding sections in this chapter and may be summarised as follows :—

Excess heat in flue gas.
Excessive loss by radiation, etc., from outer furnace walls.
Loss by radiation through doors or other openings in the furnaces.
Losses by escape of furnace gases.

RECOVERY OF HEAT FROM FLUE GASES—RECUPERATION AND REGENERATION

A furnace may be considered as consisting of a combustion chamber and a chamber in which the stock is heated. These may be separate or the same chamber may be used for both purposes.

In continuous furnaces the hot gases generally pass in counter-current to the cold incoming stock and there is thus opportunity, if the furnaces be long enough, for efficient heat transfer from one to the other, the gases leaving the furnaces at a temperature so low that there is only just sufficient heat to operate the chimney and no further recovery is possible or economic.

In intermittent furnaces, on the other hand, the outlet temperature of the gases must depend upon the temperature to which the stock is required to be heated. In the earlier stages of heating the outlet temperature is lower than in the later stages, the exact temperature depending on the conditions in the furnace, as for example on whether the furnace is maintained at a high temperature continuously or whether, as in brick kilns, it is allowed to cool down between each operation.

For the purpose of this discussion it is assumed that the combustion gases leave the furnace at a high temperature as in glass furnaces, open-hearth furnaces, gas retorts or coke ovens. Under these conditions some use must be made of the surplus heat in the waste gases.

It can be calculated as for example in Fig. 165, Chapter XX, that if the products of combustion with normal amounts of excess air escape at 1,000° C. they carry away from the furnace nearly 6,000 B.Th.U. per lb. of fuel used in

TABLE 83. SENSIBLE HEAT IN PRODUCTS OF COMBUSTION FROM VARIOUS FUELS. PER CENT. OF GROSS POTENTIAL HEAT

Temperature	1,200° C.	600° C.
Coal 14,500 B.Th.U./lb.	48·3	21·7
12,750 B.Th.U./lb.	49·3	21·5
Fuel oil	46·8	21·5
Coke oven gas	48·1	20·9
Producer gas	65·6	29·8
Blast furnace gas	79·1	34·2

the producer, or about 45–50 per cent. of heat initially supplied to the process (see also Chapter V). Table 83 gives further data on this subject, and refers to sensible heat only, omitting the latent heat of water vapour contained in the gases.

Whether this heat can be recovered or not depends in the first instance upon the scale of the operations. Obviously heat recovery cannot be applied to a small rivet-heating furnace treating a matter of pounds per hour, whereas it could be applied to an open-hearth furnace dealing with an output measured in tons per hour. Plant is available now to treat the gases from quite small furnaces.

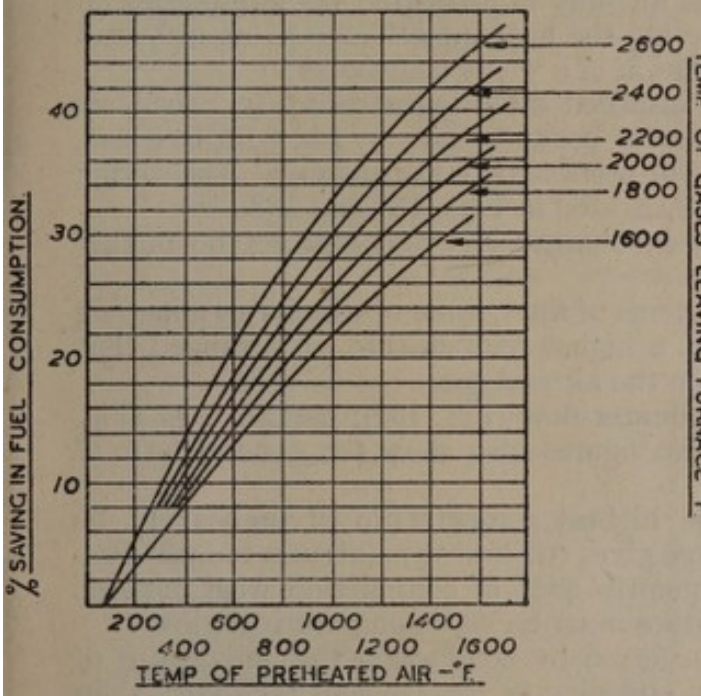

FIG. 137. Saving of fuel in powdered fuel furnaces gained by preheating the air for combustion.

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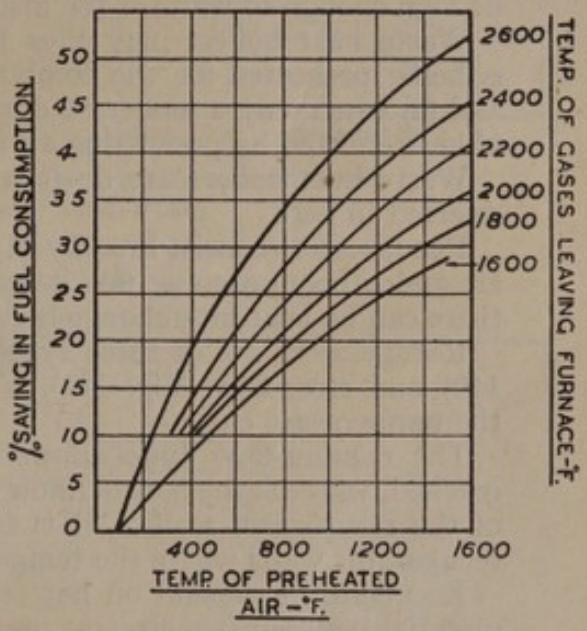

FIG. 138. Saving of fuel in producer gas fired furnaces gained by heating the air for combustion.

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Assuming that the scale of operations makes it economic, three methods of heat recovery from the flue gases are practised :—

(1) Recuperation.
(2) Regeneration.
(3) Waste heat boilers.

Recuperation and regeneration are methods by which the air used for combustion, and sometimes the gas also, are preheated by making use of the sensible heat of the flue gases leaving the operative part of the setting. Whether these methods can be applied depends on whether the process is one for which preheated air is desirable. For most heating operations in which a high temperature is required preheat is desirable. It has been pointed out in Chapter V that by preheating the air and gas the flame temperature is very considerably increased. If heat is to be transferred by direct radiation from the flame or from the hot gases, the flame temperature should obviously be as high as possible and air and gas should be preheated. On the other hand, if a long, lazy, low temperature flame is required preheated air and gas are undesirable.

It is not possible to preheat coal gas because it decomposes at a dull red heat with deposition of carbon. It is equally impossible to preheat solid fuels

(though as an exception to this, red-hot coke straight from the retorts is used in producers on gas works). In general producer gas and blast furnace gas are subjected to preheat.

Air for combustion can always be preheated if it is desirable to do so.

If there is efficient heat interchange, for solid fuel firing and for coal gas firing the quantities of incoming air and outgoing flue gases are sufficiently similar for the process of preheating the air to reduce the temperature of the flue gases to some 200° or 250° C., below which they are not cooled if they are used for operating a chimney. For producer gas and blast furnace gas, however, there is a surplus of heat in the flue gases over that which could be given to the incoming air and consequently if the air only is preheated (as in gas retorts) the temperature of the flue gases leaving the heat interchange plant may still be high enough to warrant the installation of a waste heat boiler.

Waste heat boilers may thus be installed after regenerators (*a*) when air is being preheated for the combustion of producer gas or blast furnace gas, and (*b*) when only a moderate degree of preheating is permissible. The saving of heat effected by preheating air is illustrated in Figs. 137 and 138.

Waste heat boilers are dealt with in Chapter XX and will not be further referred to here.

Recuperators consist broadly of systems of flues, some of which carry ingoing air and others outgoing flue gases at a higher temperature, so arranged that there can be heat interchange between the air and gas.

Recuperators are of three types, counter-flow (Fig. 139), parallel-flow (Fig. 140) and cross-flow (Fig. 141). These figures also show the general form of the temperature curves.

The counter-flow type allows the highest temperature of preheat to be reached, whereas the parallel-flow type gives the lowest maximum temperature of the recuperator walls. It is frequently used in conjunction with metallic recuperator walls where the temperature must be kept comparatively low.

Recuperators depend on heat transferred by conduction through a wall of material between two streams of gas, and may be less efficient thermally than regenerators especially when constructed in brick. They have, however, the advantage that they do not require periodic reversal because they are fixed passages through which air and gas flow continuously.

A regenerative furnace has been sketched diagrammatically in Figs. 49 and 135.

A regenerator is a heat exchanger constructed of refractory material. Reference to Fig. 135 will show that in its simplest form it comprises two chambers filled with chequer firebricks, the bricks being so stacked that the gases can flow freely between them and around them. If gas is burned as shown in Fig. 135 from the port on the left-hand side, the hot flue gases after doing their work in the furnace pass down the right-hand regenerator chamber and give up much of their sensible heat to the brickwork. The temperature of the brickwork decreases from top to bottom of the chamber. The air for combustion flows up the left-hand chamber. After some predetermined period, usually from 20 minutes to 30 minutes, a reversing mechanism changes the direction of flow of the gases. The gas is cut off from the left-hand ports and now admitted to the right-hand ports. The right-hand regenerator is disconnected from the chimney flue by closing a valve. The air is admitted to the base of the right-hand regenerator, and the flue gases pass down the left-hand regenerator. Thus the right-hand regenerator now gives up its stored heat to the incoming air, which is preheated, and the sensible heat of the gases is retained in the left-hand regenerator. After 20–30 minutes the flow is again reversed and the cycle is repeated.

An extension of the system permits alternate regenerator chambers to be

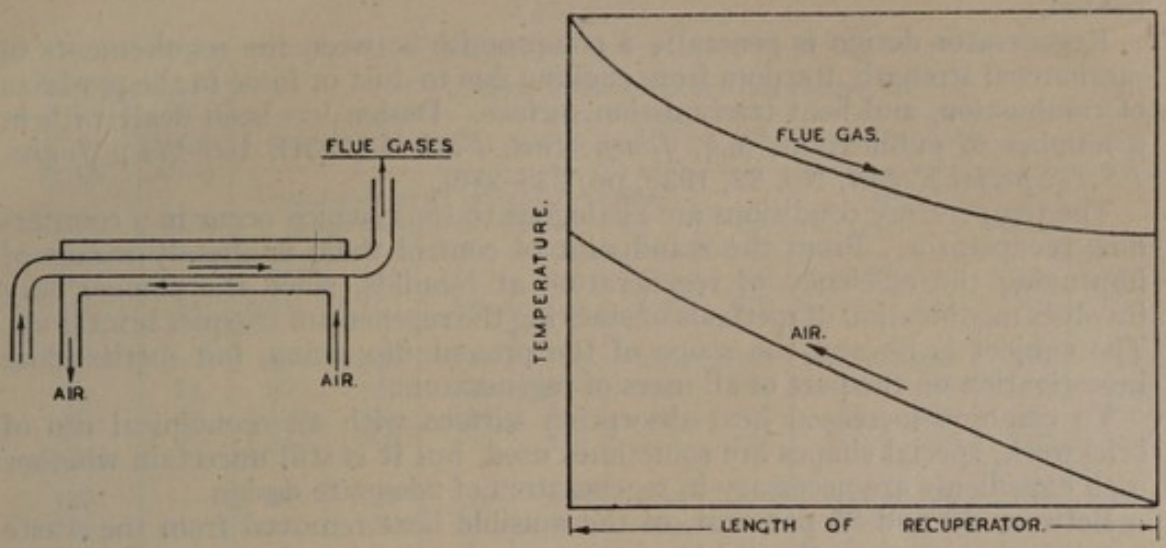

FIG. 139. Counter-flow recuperator and temperature distribution.
(Based on " Industrial Furnaces " by W. Trinks.)

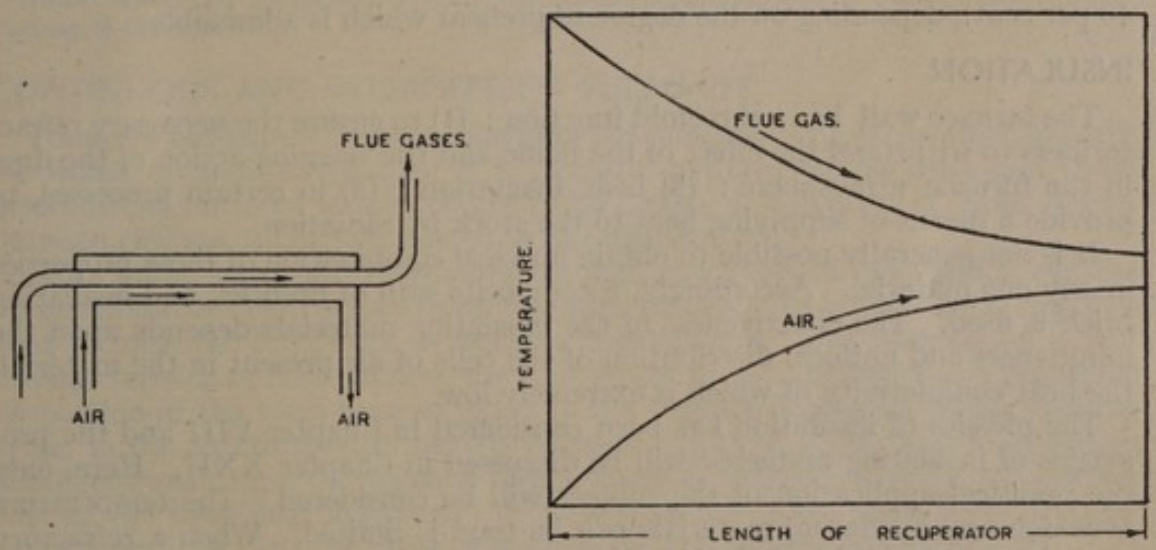

FIG. 140. Parallel-flow recuperator and temperature distribution.
(Based on " Industrial Furnaces " by W. Trinks.)

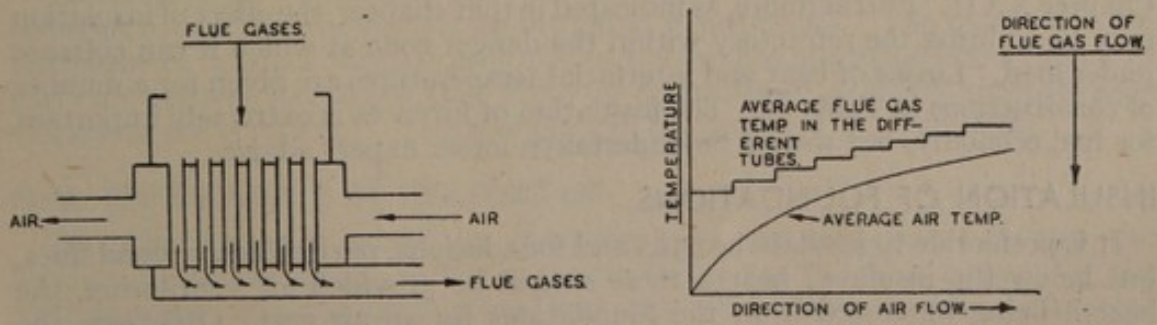

FIG. 141. Cross-flow recuperator and temperature distribution.
(Based on " Industrial Furnaces " by W. Trinks.)

used for preheating producer gas or blast furnace gas, the air being also preheated.

Regenerator design is generally a compromise between the requirements of mechanical strength, freedom from choking due to dust or fume in the products of combustion, and heat transmission surface. Design has been dealt with in a number of publications (e.g. *Journ. Inst. Fuel*, 4 (1931), 160–174; *Journ. I.S.I. Special Report*, No. 22, 1938, pp. 238–275).

The temperature conditions are analogous to those which occur in a counter-flow recuperator. From the standpoint of control there is already means of improving the efficiency of regenerators at rebuilds, since the change only involves modification of methods of stacking the regenerator chequer brickwork. The subject is beyond the scope of the present discussion, but merits close investigation on the part of all users of regenerators.

To combine increased heat-absorption surface with an economical use of brickwork, special shapes are sometimes used, but it is still uncertain whether such expedients are necessary in regenerators of adequate design.

Between 60 and 85 per cent. of the sensible heat removed from the waste gases during passage through a regenerator is transferred to the air and/or gas being preheated. Taking, however, the heat returned to the furnace by the regenerators via gas and air, as a percentage of the sensible heat in the gases reaching them, which constitutes the true thermal efficiency of the regenerative system exclusive of the flues, the maximum value realisable is of the order of 40 per cent., depending on the degree of preheat which is admissible.

INSULATION

The furnace wall has a threefold function: (1) to ensure the necessary refractoriness to withstand the effect of the flame and the slagging action of the dust in the furnace atmosphere: (2) heat insulation; (3) in certain processes, to provide a means of supplying heat to the stock by radiation.

It is not generally possible to obtain an ideal combination of these properties in any one material. Accordingly, a composite wall of firebrick and insulating brick is used. The effectiveness of the insulating materials depends upon the minuteness and uniform distribution of the cells of air present in the material, the heat conductivity of which is extremely low.

The physics of insulation has been considered in Chapter VIII and the properties of insulating materials will be discussed in Chapter XXII. Here, only the practical application of the subject will be considered. The temperature at which any insulating materials can be used is limited. When a refractory material forming a furnace lining is backed by insulation, the temperature of the outer surface of the refractory forming the interface between it and the insulation is raised, and so also is the average temperature of the brick. The interfacial temperature is the maximum temperature to which the insulation is heated and this must be less than the limiting temperature given in Chapter XXII. Furthermore, as indicated in that chapter, the effect of insulation may be to bring the refractory within the danger zone at which it can collapse under load. Losses of heat and interfacial temperatures are given for a number of constructions in Table 84. The insulation of furnaces is extremely important for fuel economy, but should be undertaken under expert advice.

INSULATION OF FOUNDATIONS

It is preferable to insulate hearths and foundations, particularly around flues, but below the insulated hearth there should be provided air ventilation, the hearth being separated from the foundations by an air gap. Otherwise the soakage of heat from large furnaces into the foundations may result in the concrete cracking from expansion.

TABLE 84

Thickness of		Temperature of the hot face—° C.									
Fire-brick in.	Insul-ation in.	1,400	1,200	1,000	800	600	1,400	1,200	1,000	800	600
		Temperature at interface between firebrick and insulation, ° C.					Heat loss B.Th.U. per sq. ft. per hr.				
4½	—	420	350	310	250	190	5,400	3,600	2,850	2,000	1,130
4½	4½	1,235	1,060	880	705	530	670	530	410	300	210
9	—	280	240	200	160	125	2,400	1,850	1,400	830	540
9	2½	990	850	710	570	430	790	620	460	330	230
9	4½	1,120	960	800	640	480	560	420	320	230	170
9	9	1,240	1,065	890	710	535	320	240	190	150	100
13½	—	260	220	190	150	120	2,150	1,600	1,300	740	530
13½	4½	1,020	880	735	590	440	480	360	270	210	160
18	—	180	150	130	110	90	1,030	750	600	460	340
18	4½	950	815	680	550	410	420	330	260	200	150
18	9	1,120	960	800	640	485	280	220	180	140	100

Note: Where no insulation is used the temperature recorded is that of the external surface of the wall.

CONTINUOUS AND INTERMITTENT FURNACES

There is a fundamental distinction between continuous furnaces which operate for weeks or months together at a virtually uniform temperature and those intermittent furnaces which require to be cooled to a greater or lesser degree between charges, and again heated to full temperature.

Both suffer losses by conduction through the walls of heat which escapes by radiation and convection from the outer surface. In both there is required heat to raise the brickwork to working temperature. But whereas, in a continuous furnace as here defined, the heat stored in the brickwork is a negligible proportion of the total heat supplied to the furnace during its whole working cycle, in the intermittently heated furnace this heat, which is at least partly dissipated each time the furnace temperature is reduced, may be the greater part of the total heat supplied. What proportion stored heat bears to total heat depends on the time cycle of the heating and cooling operation.

In continuous or long time-cycle furnaces the insulating problem is to prevent the dissipation of heat from external surfaces. In intermittent, short time-cycle, furnaces the chief problem is to reduce the heat storage loss, whilst not neglecting to mitigate the external surface loss.

The impression widely held that intermittent furnaces which must be cooled between operations should not be insulated is erroneous. The relatively new technique of hot-face insulation provides a complete means of saving much of the very serious fuel losses that occur in intermittent furnaces through heat storage in the brickwork. This subject will be discussed in more detail.

FUEL EXPENDITURE IN HEATING UP

When a furnace is heated up from the cold, the first stages involve the heating of the brickwork to the requisite temperature. It is frequently not realised what large amounts of heat are required for this purpose. The heat required over a period depends to a great extent upon the time cycle of the furnace. In a continuous furnace the proportion which the heat used for heating up bears to the total heat that is used in the furnace per annum is very small.

In a furnace with a weekly cycle (e.g. reheating furnaces), when the furnace is not used at the week-end it will be considerable. With a daily cycle, shutting down over-night (e.g. drop-forge furnaces and some heat-treatment furnaces) it will be very considerable. Finally, in furnaces operating on a heating-and-cooling cycle (e.g. annealing furnaces and intermittent brick kilns) in which the furnace is heated up with each charge of goods and cooled with the goods in, the heat consumption by the brickwork may assume quite surprising proportions in relation to the heat usefully consumed.

The heat stored in the brickwork in total B.Th.U. is given (Chapter V) by the product :—

weight of brickwork (lb.) × sp. ht. (per unit weight) × average rise in temperature (°F.).

The specific heat is within narrow limits the same for all siliceous refractories, including firebricks, silica bricks and insulating bricks. The rise in temperature depends on the conditions. The essential differences lie in the weight of the bricks. Firebricks and silica bricks weigh between 100 and 130 lb. per cubic foot in general, whereas insulating bricks are about one-quarter of these figures. Thus, for a given temperature range, the heat stored in a wall of insulating bricks is only one-quarter of that stored in a similar wall of acid refractory—an argument in favour of using as much insulating brick and as little refractory brick as possible in the walls of intermittent furnaces.

To give some illustration of the quantity of heat used for heating up the brickwork to the steady state, the conditions calculated in Table 84 have been used to calculate Table 85. Here the refractory is taken to weigh 120 lb. per cubic foot ; the insulation 30 lb. per cubic foot. The specific heat of both is regarded as 0·24. The temperature is calculated from a basis figure of 60° F.

TABLE 85

Composition of furnace wall		Mean temp. in wall				Heat stored in wall B.Th.U./sq. ft. of internal surface		
Refractory in.	Insulation in.	Refractory °F.	Refractory °C.	Insulation °F.	Insulation °C.	Refractory	Insulation	Total
9	nil	1,250	676	—	—	25,800	—	25,800
13½	nil	1,210	654	—	—	37,400	—	37,400
9	4½	1,829	998	945	507	38,200	2,400	40,600
13½	4½	1,765	963	877	470	56,200	2,220	58,420
9	9	1,902	1,040	988	531	40,000	5,020	45,020
13½	9	1,860	1,016	945	507	58,400	4,800	63,200

INTERMITTENT OPERATION

Clearly, the shorter the time cycle, the greater is the heat stored in the wall in comparison with the heat lost by conduction, etc., through the wall.

The first effect of external insulation is to increase the heat storage in the wall (Table 85) because the average temperature of the heavy refractory lining is increased. By putting 4½ inches of insulation on the exterior surface, the heat storage is increased by about 60 per cent. This is of no consequence in a continuous or long-cycle furnace, but becomes of great significance in a short-cycle intermittent furnace. The solution of the difficulty lies in reducing to a minimum the weight of refractory that must be heated and cooled each cycle. Until comparatively recently this solution was rendered impossible by the high

temperature of the insulation/refractory interface, insulating materials not being available to withstand this temperature.

The newer technique of " hot-face " insulation has become possible through the development of bricks which combine high refractoriness with good insulating properties. It is now possible to construct furnaces wholly of these bricks, or line the interior with these bricks. Tables 86 and 87 indicate from published figures the value of hot-face insulation for intermittent furnaces. Table 86 (*Trans. Cer. Soc.*, XXXV, 441) shows the very much smaller quantity of heat stored in a wall constructed of high temperature insulating material. Table 87 compares the performance of an uninsulated intermittent furnace with a hot-face insulated furnace with single and double insulation.

TABLE 86. HEAT STORAGE IN RAISING HOT FACE TO 1,300° C. AND MAINTAINING TO GIVE FULL SATURATION

Type of wall	Thickness		Heat stored, B.Th.U./sq. ft.			Relative proportions of heat stored
	Brick in.	Insulation in.	Brick	Insulation	Total	
Hot-face insulation ..	9	—	—	—	15,010	1
Firebrick backed with ..	18	4½	86,600	2,730	89,330	6
insulation. ..	13½	4½	67,720	3,025	70,745	4·7
..	9	4½	46,950	3,310	50,260	3·4

TABLE 87. FURNACE OPERATING AT APPROX. 1,000° C. OVER 120 HOURS CONTINUOUS CYCLE WITH A WEEK-END SHUT-DOWN OF 48 HOURS
(k = thermal conductivity in B.Th.U./sq. ft./hr./°F./in.)

Method of construction	9 in. firebrick k = 7·8	9 in. hot-face insulation k = 1·88	9 in. hot-face insulation, k = 1·88 ; backed by 2 in. insulation, k = 0·74
Loss by transmission through brickwork B.Th.U./sq. ft./week ..	146,400	39,300	25,200
Heat capacity loss, B.Th.U./sq. ft./week-end	20,090	6,540	6,810

Hot-face insulation can be used for a variety of purposes, one of which is for lining hot gas ducts. A producer gas main at a steel works which had been so lined gave the following results :—

	Before insulation.	After insulation.
Gas inlet	700° C.	730° C.
Gas outlet	460° C.	660° C.
Temperature drop along the main ..	240° C.	70° C.

The relatively low cold crushing strength of these bricks does not preclude their use for large furnaces, as is sometimes supposed. The weight of the structure is very much reduced by using the lighter bricks and the reduced average temperature in the brick for a given hot-face temperature leads to a smaller reduction in the crushing strength with rising hot-face temperature when compared with firebrick. Brick kilns up to 31 feet in diameter with a total

weight of crown of 14 tons have been in operation, using hot-face insulation, for the past eight years without major repairs; no important part of this structure is built of firebrick.

Figures obtained on a small gas-fired forge furnace are given in Table 88. No alterations were made in this furnace other than the substitution of $4\frac{1}{2}$ inches of insulating lining in place of the same thickness of firebrick.

TABLE 88. EFFECT OF HOT-FACE INSULATION ON A FORGE FURNACE

	Firebrick lined	Lined with insulating brick
Time to reach 1,150° C. from cold	55 min.	15 min.
Gas used in heating up	400 cu. ft.	90 cu. ft.
Rate of consumption	475 cu. ft./hr.	375 cu. ft./hr.
Gas rate for maintaining working temperature at 1,150° C.	350 cu. ft./hr.	175 cu. ft./hr.

Finally, Table 89 (*Trans. Cer. Soc.*, XXXV, 458) indicates results claimed on a gas-fired annealing furnace operating at 750° C., in which an older furnace of normal design was replaced by a new furnace built of insulating refractory, with a non-refractory insulating backing. Part of the improvement registered was due to the provision of instruments and controls.

TABLE 89

Heating up to 750° C.	Old furnace	New furnace
Time	4 hr.	1 hr.
Gas used per heat	4,500 cu. ft.	750 cu. ft.
Gas used at six heats/week	27,000 cu. ft.	4,500 cu. ft.
Saving		22,500 cu. ft. = 83·3%
Maintaining at 750° C.		
Gas used at 44 hr./week	33,000 cu. ft.	15,400 cu. ft.
Saving		17,600 cu. ft. = 53·3%

The quest for high output in intermittent furnaces has led to the practices of (*a*) constructing these furnaces of thin, uninsulated walls to reduce the thermal storage, and (*b*) regarding insulation as unwise; both practices are supposed to assist in rapid cooling and so to increase output.

These practices and beliefs are fundamentally unsound. Thin walls may be admissible for very short time cycles, but if the furnace is operated for longer than the limiting time corresponding to the thickness of the wall, these thin walls lead to a high external temperature and to great consequent waste of fuel.

Insulation—external or internal—is always an advantage even with intermittent furnaces which must be partly cooled internally between charges. The inside surface of the wall can be cooled rapidly by air (if suitable bricks are used) for the reception of cold stock, but the body of the brickwork remains hot and the furnace wall requires less fuel in the aggregate. The opinion that intermittent furnaces should not be insulated arises from ignorance of the true temperature gradients that remain in a furnace wall when the surface has been rapidly chilled.

Tables 88 and 89 indicate the much greater rate of heating up when hot-face insulation is used, as a direct result of the low heat capacity. For the same reason cooling is also accelerated, since the heat content of the brick is lower and the temperature gradient steeper.

When a furnace is closed down for a period daily or at the week-end, insulation will prevent heat losses and will enable the furnace to be started up again quickly and with little expenditure of fuel. In order that the maximum advantage may be derived from this effect, it is essential that the brickwork should be well pointed to avoid ingress of air, doors should fit tightly, and in particular the dampers must be tightly fitting so that no cold air is drawn through the furnace during the time it is closed down. Under good conditions it is possible to keep sealed-up furnaces hot for long periods. This is an alternative to banked fires or limited gas consumption that should be considered.

RADIATION LOSSES THROUGH OPENINGS

Fig. 142 indicates a furnace in which there is a hole in the furnace wall. An

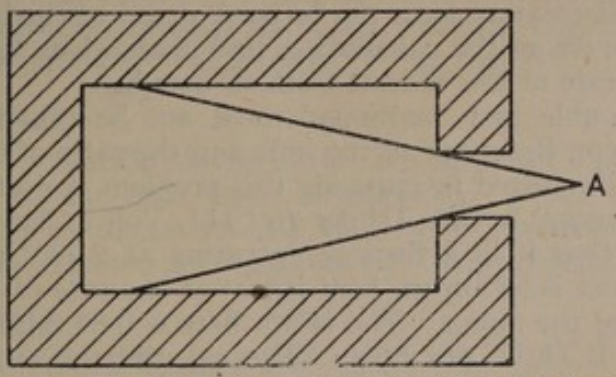

FIG. 142. Heat radiation from an opening in a furnace.

observer at A would see the interior of the furnace as if that interior were a plane surface situated at the opening and having the same area as the opening.

Since the interior of a furnace is generally taken to be equivalent to black-body radiation (E = 1) the rate of heat radiation per unit area of plane

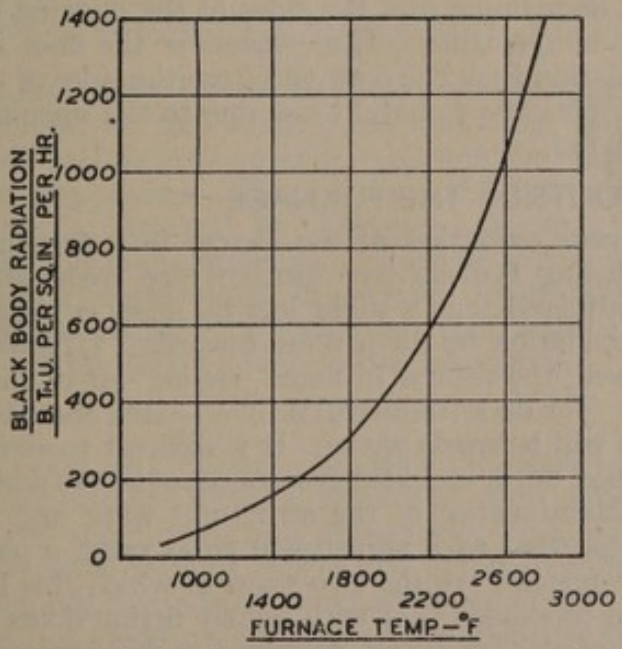

FIG. 143. Black body radiation from a furnace interior.

surface observed would be given by Fig. 143 for a furnace having walls of negligible thickness at the opening.

An example will illustrate the method of calculation. A furnace front is bricked up with a $4\frac{1}{2}$ inch wall, through which an observation opening (peep-hole) of $9 \times 2\frac{1}{2}$ inches is left open. It is required to calculate how much heat

escapes through the hole by radiation if the furnace temperature is 2,400° F. (1,315° C.).

Area of equivalent diaphragm = $2\frac{1}{2} \times 9$ inches = 22·5 square inches.

From Fig. 143 black-body radiation from 1 square inch of surface at 2,400° F. = 800 B.Th.U. per hour.

Heat loss from freely exposed diaphragm = $800 \times 22{\cdot}5 = 18{,}000$ B.Th.U. per hour.

This calculation is only correct if the furnace opening is constructed of very thin material. The result is too large because the thickness of wall obstructs the direct radiation to an extent depending on the ratio of the wall thickness to the width of opening; and it is too small to the extent to which the sides of the opening become heated by radiation received from the interior of the furnace, and re-radiate away from the furnace a part of the heat thus received. As a result of these errors the calculation just made gives figures of the order of double the real rate of loss of heat from an opening.

The figure of double just mentioned must not be taken as an invariable factor; it depends on the shape of the hole and the ratio of least width to wall thickness. Those interested in pursuing this problem are referred to the book "Industrial Furnaces" by W. Trinks (p. 111, Vol. I, 3rd Ed.). A similar calculation shows that if in a furnace operating at 2,200° F., a furnace door measuring 5 × 4 feet is left open, heat is lost at the rate of 1,728,000 B.Th.U. per hour, neglecting the factor. The factor here is 0·85, making a total loss of heat of 1,469,000 B.Th.U. per hour. The calculations are more specialised than the treatment here demands and are given not for instruction in the method of making them, but to direct attention to the importance of keeping all openings in furnaces closed as far as possible. All apertures through which the furnace can be observed should be covered with plates preferably backed by insulation.

In a furnace door which is opened periodically, continuous loss of heat does not occur, but the door lining and the sides of the opening become heated to practically furnace temperature. Thus whenever the door is opened there is the dual loss of radiation from the door and from the sides of the opening which are quickly cooled, plus the radiation loss due to the opening which has previously been calculated.

LOSS BY FLAME OUTSIDE THE FURNACE

In order to prevent oxidation of the charge in a furnace which must be operated in a reducing atmosphere, furnaces are frequently worked under pressure. It is considered that a slight loss by gases passing outwards is less detrimental than oxidation by air leaking inwards. If, however, the furnace door does not fit well, the volume of flame passing out of the furnace may be quite considerable. The door frame will be overheated and will become warped, and the conditions will be made worse. It is difficult to estimate the amount of loss from this cause, but it is evidently a source of loss to which close attention should be paid. Maintenance of the structural work and correct fitting of doors and covers, together with adjustment to as small a positive pressure as possible within the furnace are the two ways in which this loss may be minimised. It has been concluded that with tightly fitting doors this loss may not exceed 2 per cent. of the total heat delivered to the furnace, and should not average more than 1 per cent. With loosely fitting doors, and with a flame or jets of heated gases projecting against the doors, it may easily reach 8 per cent. and at times may even exceed that amount.

This loss is especially heavy in large furnaces with ingots projecting under the door into the open, because large, half-open doors cannot be made tight. Under these circumstances it is better to brick up the opening around the ingot.

A similar loss may occur through flames within the structure not being confined to the furnace chamber. It sometimes happens when combustion is badly adjusted that the flame passes through the furnace and continues to burn in the recuperators, regenerators, or waste gas flue.

If there are adequate arrangements for recovering the heat that is lost and this heat can be put to useful purpose, the result may not be very serious. If, however, the heat is not recovered the loss may be very great indeed.

Finally, in furnaces operating under pressure, open spy-holes or cracks in the furnace wall may cause a loss of gases by flame burning in the outside air.

It has been recorded, perhaps as an extreme example, that during the inspection of furnaces in the drive for fuel economy, instances have been noticed in which there appear to be nearly as much flame burning outside the furnace as inside. With injection burners in small furnaces this condition is often caused by excessive pressure cutting down the air supply.

SCIENTIFIC CONTROL

In any attempt at scientific control of furnace efficiency a primary knowledge of the distribution of the heat losses is essential. From the resulting heat balance it is possible to devise routine methods of control which will ensure that the controllable elements are regulated (*cf.* Chapter XXIX.).

THE HEAT BALANCE

The main elements of the balance sheet are :—

(1) *Useful Heat Transference to the Charge.* This is readily obtained by a knowledge of the temperature, weight and character of the charge from which the heat capacity can be obtained. It entails recording the necessary information of the weights charged, and accurate pyrometry, the importance of which in their bearing on economical furnace practice cannot be over-emphasised.

(2) *Combustion and Chimney Losses.* (*a*) Unburnt combustible, both in the products of combustion, as deduced from gas analysis, and in the ashes derived from solid fuels.

(*b*) *Sensible heat and latent heat in waste gases*, obtainable from determination of the CO_2 content of the waste gases and their temperature. Charts constructed for each fuel as in Chapter V. facilitate the evaluation of this loss.

(3) *Furnace Structure Losses.* Heat stored in brickwork and lost to surroundings is frequently obtained by difference. Its exact determination is difficult, but considerable information of value can be obtained by the determination of external surface temperatures. Excessive temperatures, besides indicating excessive surface losses due to insufficient thickness of refractory lining, point to breakdown of insulation and defective brickwork.

INSTRUMENTS

The instrument panel on a furnace should be the nerve centre of the installation. It should indicate at all times the rate of input of heat to the installation and whether the combustion and draughting conditions are correct. In the smaller types of plant draught gauges and pyrometers may have to suffice. The air consumption can be made an index of the heat input. Its measurement by means of an orifice gauge, a simple and inexpensive type of instrument (Chapter X.), offers a ready means of control on any type of furnace. Safeguards must be taken to avoid inleakage of cold air, which can vitiate the indications of the flow meter.

RECORDS OF PERFORMANCE

The first essential towards attaining control of fuel economy is to keep regular records of the data necessary for watching efficiency. These include routine

recording of the fuel used, the output, the hours worked, the variations of shop conditions, and of any special tests carried out on individual furnaces. Charts of the performance of furnaces should be drawn and kept under constant observation. If the rate of fuel consumption is plotted against the rate of output generally a straight line relationship is found, which when produced to cut the fuel consumption ordinate at zero output gives the rate of consumption for the empty furnace. This represents the quantity of fuel which is required per hour, say, to bring the furnace to temperature, and maintain it in operation under the requisite conditions. Alternatively, the weekly values of the fuel ratio in terms of fuel used per ton of output can be plotted against the weekly output. Such curves give datum lines from which any improvement in method of operation or constructional alteration can be gauged and optimum rates of output assessed. A set of values for a reheating furnace is given in Table 90, from which it can be determined that a rate of fuel consumption of approximately 0·9 cwt. per hour is necessary to keep the furnace at its operation temperature, and to make good the heat losses in the furnace structure and the waste gases. The values in the last column show how the fuel ratio becomes reduced as the rate of output is increased.

TABLE 90. FUEL CONSUMPTION—OUTPUT RELATIONS IN INTERMITTENT FURNACES

Rate of fuel consumption cwt./hr.	Rate of output cwt./hr.	Fuel ratio :— Tons fuel ÷ Tons output
1·24	4·5	0·275
1·25	5·0	0·25
1·40	7·0	0·20
1·52	8·0	0·19
1·76	9·0	0·195

PRACTICAL HINTS TO OPERATORS

The following notes comprise some brief practical hints to operators of various types of furnaces.

Coal-fired Furnaces

(1) With hand fired furnaces the depth of fire should be 15 inches. No fresh firing should exceed 1½–2 shovelfuls for each square foot of grate surface. Firing too heavily causes excessive smoke and waste of fuel.
(2) Intervals between firing should be from 15 to 20 minutes in reheating furnaces, and should never exceed 30 minutes, even in heat treatment furnaces.
(3) The grate should be kept covered, the fires level and free from holes, and the rake used when necessary.
(4) Firebars must be evenly spaced, and burning or warping should be rectified at once.
(5) Unnecessary wastage of cinders in the ashes is avoided by cleaning the fires carefully. After cleaning the fire is worked back gradually.
(6) The air supply is regulated to give bright clean fires.
(7) Inleakage of air should be avoided, since it removes heat from the furnace and transfers it to the chimney. All sources of air inleakage should be patched up.
(8) Dampers must be kept in good repair, and the draught in the furnace balanced to prevent either inleakage of air or blowing out of flame. In most furnaces a slight indraught below the door is consistent with balanced conditions of draught.

Coke-fired Furnaces

With coke firing, the larger the coke the thicker should be the fuel bed with a depth of 12–15 inches as a maximum. The conditions of firing differ from those of coal-fired furnaces, since no smoke can be produced from coke under any conditions, but guidance on all the other points is practically the same as for coal-fired furnaces.

Gas- and Pulverised Coal-fired Furnaces

(1) Burners should be kept clean and in good repair. All adjustable parts should move freely. Air controls on burner injectors often become jammed through dirt, corrosion or mechanical damage.
(2) Dampers should be free, in good working order and not subject to inleakage of air.
(3) The correct amount of air can be ascertained by inspection of the flame, though the indications will differ with various types of burner. More exact indication can be obtained by flue gas analysis, and it is of great help to furnace operators if a CO_2 recorder is installed for their guidance.
(4) In blast burners which normally work with a non-luminous flame, a long lazy flame indicates too much gas. Correct combustion is obtained by shortening the flame until the yellowish colour just disappears. This is done by cutting down the gas, or increasing the air supply. The roar of the burner is then generally at its loudest.
(5) In other types of burner, where mixing of gas and air is delayed, a bright yellow colour shows good combustion, the brighter the better.
(6) Once the burners are set to give proper combustion individual burners should be adjusted as little as possible. The gas supply to the whole furnace should be reduced or increased whenever necessary by using the main valve only. As with coal-fired furnaces, inleakage of air should be avoided whether from furnace doors, from open spy-holes or from faulty brickwork. Air inleakage in effect transfers heat from the furnace to the chimney and is the cause of much waste.
(7) In bogie furnaces the sand seals should be kept full.
(8) The damper should be used to keep a balanced draught in the furnace. The correct position for the damper can be found by burning a small flame or blowing smoke in a sight-hole and adjusting the damper till the flame just flickers or the smoke just fails to be drawn into the furnaces.
(9) Air slides and valves should be adjusted to correspond with the flow of gas. Whenever the amount of gas supplied to the furnace is altered the damper should be adjusted to re-establish the original pressure conditions within the combustion chamber or furnace.
(10) Furnaces should not be lighted up until they are required, due allowance being made for the time necessary to attain working temperature. When furnaces are not required gas and air should be shut off and the dampers closed. The fuel supplied to a furnace should be correlated with the amount of work required by the furnace.

Oil-fired Furnaces (cf. also Chapter XXVIII)

(1) The liquid fuel at the point of use must be of right viscosity if the burners are to operate properly.
(2) Burners should be dismantled and cleaned very frequently, preferably once a shift.
(3) Burners should be fitted with graduated valves that permit a visual indication of the amount of oil which is being used ; this permits ready adjustment of oil supply to load. Similarly, dampers should be graduated.

(4) When the furnace is shut down, the burner should be removed to protect it from the radiant heat or protected by a metal sheet. The burner should not be kept cool by leaving the steam supply on.
(5) Water should be separated from the fuel oil; it can delay combustion and produce heavy smoke.
(6) The sizes of apertures in burners should be checked periodically.
(7) The general principles of efficient operation that apply to other fuels also apply to oil-fired furnaces.

General Rules for the Economical Use of Fuel

(1) With existing equipment, heat treatment must be planned so that as far as possible the appliance is working at full load. Maintenance periods with no load and long periods of operation below capacity should be avoided. Major fuel economies can be achieved by careful load planning.
(2) Flame should be kept within the furnace; gas should not be burnt beyond the furnace in the flue.
(3) Doors should be reasonably well balanced and fit well; they should be readily operated. Flames outside the doors cause warping or damage to the door lifting gear; the damage and loss become progressively greater.
(4) Fuel should not be burned at full rate when heats have to be held.
(5) A record of fuel used and weight of goods heated should be kept.
(6) Leaky brickwork is one of the greatest sources of thermal loss. The furnace brickwork should be put into first-class repair so that infiltration of air cannot occur. It should also be painted with a suitable material, e.g. fireclay and cement or for lower temperatures a mixture of tar and fireclay, to prevent air leaking in through the pores of the bricks. Attention should be paid to the condition of the furnace doors, sight-holes, etc. Furnace doors should fit closely to the surrounding brickwork. The brickwork should be examined as a matter of routine at frequent intervals and in any event not less frequently than once a month.

CHAPTER XIX

SPECIAL FURNACES

Structural features—Auxiliary plant—Furnace testing—Melting furnaces : hearth, crucible, shaft and cupola furnaces—Sintering furnaces—Kilns and ovens : intermittent and continuous —Furnaces for the glass industry—Reheating furnaces : soaking pits—Heat treatment furnaces : single purpose furnaces ; conveyor furnaces ; " top-hat " furnaces ; lead pots and salt baths —Furnace control—Variable heat flow in furnaces.

INTRODUCTION

THE principles of furnace operation and design have so far only been dealt with from the standpoint of the combustion of the fuel, heat transmission and the flow of gases. The present chapter deals with the bearing of the structural features and the inferences from the heat balance on the problem of the efficient use of fuel in furnaces, and then with certain details of the operation of special types of furnaces.

These are classified on the following basis :—

I. Melting furnaces, including hearth, crucible and shaft types.
II. Sintering furnaces, represented as a typical example by the cement kiln.
III. Kilns and ovens.
IV. Furnaces used in the glass industry.
V. Reheating furnaces.
VI. Heat treatment furnaces.

STRUCTURAL FEATURES

STRENGTH AND DURABILITY OF FURNACES

The methods and materials used in furnace construction have an important bearing on fuel efficiency since breakdowns and delays may be a source of heat waste. Refractories and insulation have been given separate attention in Chapter XXII. The furnace framework is made either of cast-iron plates bolted together or steel structural work, iron castings and plates, selection being governed by mechanical considerations and the possibility of the part being subjected to heat and oxidation. None of these materials is satisfactory when exposed to high temperatures, and there must then be recourse to special materials such as heat resisting alloys. The development of these materials during recent years has resulted in considerable progress in the mechanisation of furnace practice, and the provision of structures and mechanisms capable of being exposed to the internal heat of the furnace. Such materials are used extensively up to temperatures of 1,000° C. and for special work at even higher temperatures.

They are required to be resistant to scaling, and to have sufficient hot strength to be capable of being used in many types of mechanisms such as conveyors. The factor on which selection of the material is based in regard to this last-named property is known as the creep strength, that is the load which the metal can carry for extended period at the temperature involved without perceptible creep. Other properties of significance are resistance to carburisation (carburising boxes), to the effects of repeated heating and cooling (chain conveyors, particularly those immersed in quenching media), heat diffusivity (glass furnaces), malleability (for bars and sheets), machinability and, in short, what is important is whether the materials can be cast or fabricated into the many forms to be found in the modern furnaces.

An extensive literature of the subject exists to which reference may be made.

The design of the structural framework of the furnace is based on the well-known principles of mechanical construction. A special point in its bearing on fuel efficiency is that robustness is essential since repeated heating and cooling of structural members together with the concomitant expansion and contraction of the refractory materials used can induce, in course of time, a remarkable degree of distortion of framework. Large furnaces of insufficient robustness have been known to be several feet out of alignment after extended use where sufficient strength and allowance for expansion have not been applied. Then cracks in brickwork more readily arise, resulting in air inleakage or flame emission, and a lowering of furnace efficiency.

THE FURNACE CHAMBER AND ROOFS

Provided the abutment of the arch, or skewback, is adequately supported by a strong framework with tie rods, the arch on heating must rise. In selecting brickwork the expansion characteristics of the refractory to be used must be known, also whether there is any after-contraction on firing to the temperature at which the roof is to be operated. If the brickwork were incompressible then with good shaped material, the bricks would be in contact over their full length near the arch support, but near the crown of the arch they would tend to touch only at the inner edge. Proposals have been made and roofs are constructed of specially shaped end bricks abutting on the skewback to allow for this variation to be taken up during heating. Actually, all firebricks are capable of crushing elastically to a degree compatible with a robust construction, and, in general, a rigidly bricked roof, provided ample rise in the arch is allowed (for example, 1½-inches to the foot), gives satisfactory service. Further, the softening of the bricks at furnace temperatures and the lower refractoriness resulting from the application of pressure, permit plastic compression of the bricks on the furnace side of the arch, and tend to equalise the load and compensate for irregularities of shape. Again, the line of thrust of a hot arch departs farther from the median line of the brickwork towards the hot surface as the centre of the span is approached, so that the greatest deformation is likely to arise where the initial separation of the brickwork is greatest.

Furnace roofs fail either by yielding of the skewbacks, or by spalling and wear. Spalling can weaken an arch, and it can then finally be reduced to a condition of collapse through normal slag attack and erosion by furnace gases and fume. It is remarkable how thin a roof can wear without collapse provided the abutments are rigid, but then the question of heat loss by conduction and from external surfaces possibly outweighs the loss of durability of the roof, unless as so frequently occurs in industrial practice, it is necessary to wait for a convenient period for repair. It may be urged with good reason that planned maintenance requiring repairs to be done at specified intervals is preferable to the policy of striving for the absolute limit of durability.

Under suitable conditions abutments can be made rigid and dependence placed on the rise of the arch ; alternatively, spring loaded or adjustable tie rods may be used, but these latter require skill in adjustment, and unskilled handling may result in failure of the arch.

Distances greater than 18 feet are seldom spanned by a single arch. They are either broken up into several smaller spans, supported on skewbacks, carried on longitudinal girder reinforcement, or slung from overhead beams. Flat roofs, built of refractory tiles suspended by hangers from overhead girders are also used. The relative merits of the sprung arch and the suspended roof depend entirely upon the conditions, the occurrence of movement, as in soaking pit lids, being a factor of significance operating against the success of the flat

roof. A complicated shape constructed in fired refractory material is always more vulnerable to the effects of variable stresses than the simple shapes capable of being used in an arched roof. On the other hand, under suitable conditions of use the suspended roof may be preferred since it eliminates the need for the skewback and gives freedom of design in the sidewalls for the provision of openings to the furnace chamber.

Roofs 9 inches thick are generally insulated with advantage, except in special cases over combustion chambers where the temperature conditions are too severe for the brickwork. Selection of a better-class refractory brick may, on the other hand, be here successful.

Roofs may collapse from mechanical injuries arising from explosions or impacts. The best of brickwork and design is of no avail against destructive treatment which may arise from errors in lighting up or in controlling charging operations.

In roof construction attention should be devoted to the choice of brick shapes to give uniform stresses, and a reasonable gas tightness to obviate the excessive leakage of hot gases which occurs in a badly built roof due to the buoyancy of the hot gas. In certain types of kilns it is considered good practice that the crown or arch should spring from a wall behind the kiln lining. The construction of crowns of certain types of kiln differs in important features of technique from that of other high temperature furnaces, and being a special field of practice is here excluded.

SIDE WALLS AND HEARTHS

If the side wall forms the support for the roof its correct construction is of primary importance both in relation to the thrust of the roof and of the hearth. The major fault to be avoided in the construction of the lower part of the furnace lining is a lack of adequate strength for the supporting girders of the hearth, which should be securely anchored to the buckstays, whilst side-wall support in high temperature furnaces should be provided by the use of strong cast-iron plates, particularly in situations not required to be made accessible for the purpose of brickwork repair. Such construction admits of the ready use of the highest grade insulating bricks in side walls.

Adequate insulation of hearths in modern furnace practice is imperative, and it is practicable to obtain material capable of withstanding the heaviest hearth loads. Recent research has shown that economical furnace operation depends to a considerable degree on the manner in which uniformity of temperature is rapidly attained. Since, except in under-fired furnaces, the hearth is the coldest part of the furnace, provision for its adequate heating is a prime factor in good furnace design. At the same time adequate refractoriness, stability and resistance to attrition must be provided.

An excessive amount of brickwork does not constitute good practice. There must be the correct combination of suitable refractories to maintain thermal insulation and mechanical stability. Stability is assisted by ventilation, i.e. the most satisfactory hearth is mechanically supported and air-cooled from below.

Slagging troubles or soft hearths can usually be overcome by a correct selection of refractories, for example by the use of a superior firebrick, or a basic refractory such as dolomite, chromite, or in special cases magnesite, and in the rarest sillimanite, carbon or carborundum. In siliceous hearths slag is run off in liquid form through a tap hole, and the condition of the hearth made good by fettling with a suitable refractory as, for example, sand or iron oxide cinder when the slags are ferruginous. Dry hearths should always be kept clean and free from scale and cinder by raking at convenient intervals of time.

FOUNDATIONS

The design of the foundations is not a feature that can be usefully discussed as a factor entering into the problem of correcting practice in normal operation, since it is a feature of the initial construction and can only be adequately met at the installation of the furnace. Overheated foundations may, however, be the source of uneconomical operation by permitting subsidence which gives rise to cracks in brickwork, and resultant wasteful inleakage of cold air. Thin, non-ventilated, brickwork hearths which are, however, capable of being readily altered, can cause overheating of concrete, dehydration of which commences at 260° C. and is completed at 480° C. Limiting thicknesses are D/6 for furnaces working at below 1,205° C. and D/8 for 870° C., D being the shortest dimension of the hearth in feet.

HEARTH SUPPORTS, SKIDS AND HANGERS

A wide range of mechanical devices is used for the purpose of hearth supports, skids and hangers, to enable stock to be moved readily through continuous furnaces. From the standpoint of fuel efficiency certain general principles should be observed in their construction and maintenance.

Apart from waste of fuel and output arising from interrupted campaigns and the preservation of general economy of operation, excessive heat losses may arise from water cooling, but its use under certain circumstances is inevitable. Skid pipes are usually of hydraulic section, and must be securely anchored in the furnace hearths. They may be fastened into the anchorage in such a manner as to admit of being turned to expose new and unworn surfaces for further service or, alternatively, welded strips may be used to take the wear. Safeguarding the water supply and preventing blockages in pipes are almost self-evident precautions.

The main objection to the use of water cooling in skids arises from cold patches in the stock, and finishing periods of heating must then be made by transfer to solid refractory hearths. The use of dry skids, hearth plates, and mechanical conveyors introduces a field of furnace practice of a specialised type. Where mechanical conveyors are used the amount of sensible heat lost from them may be an important item in the heat balance. The heat-resisting steels already mentioned are frequently used for these purposes.

JAMBS, DOORS AND OPENINGS IN SIDE WALLS

Openings in side walls can be a source of inefficiency if robust methods of construction are not applied. This applies particularly to furnaces operating at temperatures above 1,000° C. Open doors lose heat at a very high rate from direct radiation to the atmosphere. Distorted buckstays, piers and jamb reinforcement give rise to air inleakage, and flame loss, for proper fitting of doors is then no longer possible. Modern refractory practice in the provision of high temperature insulating materials, on the other hand, makes practicable more robust and better doors. A most important feature of furnace practice is attention to the sealing of the doors to prevent such leakages. The main factors in overcoming such disabilities are the initial robustness of the reinforcement and the use of well-fitting door frames, side plates and steady mechanical action during movement. Water-cooled doors may be essential, but the heat losses due to the cooling action should always be examined in relation to the general economy.

A badly fitting discharge door on a reheating furnace having a gap along its top edge 1 inch wide and 24 inches long will give a heat loss of approximately 200,000 B.Th.U. per hour due to escaping gases if the furnace is operating at

a temperature of 1,100° C. with an exhaust temperature of 700° C., and with a normal furnace pressure. This clearly shows the loss which can be obtained from a comparatively small opening.

AUXILIARY PLANT

In addition to the furnace proper there is often much equipment ancillary to the furnace which requires attention if the best results are to be obtained.

DAMPERS

The function of a damper is to regulate the draught, and accordingly it becomes the prime instrument of the furnace economy. The controls should always be situated in such a position as to be readily accessible, and preferably also permit of sighting the flame when adjustments are being made. The choice of the material from which the damper is made depends upon the conditions of operation. It must not crack or warp, nor must brickwork remain dislodged. It should move easily in its seating and be readily capable of fine adjustment since quite a small movement may mean a waste of many tons of fuel. Allowance should be made for expansion, and suitable covers or leakage seals provided to prevent the inflow of cold air to the flue. If the furnace is temporarily shut down the damper should always be closed. Closing the damper when shutting down the furnace is equivalent to turning off the fuel valves since it prevents cold air inleakage.

Water-cooled dampers of special design or dampers constructed of heat-resisting alloys are necessary for the highest temperatures. For gas-tightness they should be made to slide on inclined damper frames machined on the working face.

FANS

Controlled draught, whether "forced," "induced," or "balanced" (Chapter VI) assists the maintenance of the correct air-fuel ratio, and the adjustment of the rate of heat input to the most economical value. Mechanical draught is generally indispensable with recuperators, which require high velocities of air flow for adequate efficiency, with consequent increase of resistance.

Fan design and selection is a matter for the specialist. The principal consideration when fans are installed is to see that they are maintained efficiently by regular lubrication and cleaning. Particular care should be paid to the inlet side of the fan and suitable protection placed to prevent foreign matter being drawn into working parts. With induced draught fans disastrous results may arise from flame impingement on the blades or overheating from other causes. Valves and dampers in the fan circuit should be readily controllable. A fan gives a ready means of registering the air flow to the furnace either by pipe attachment with orifice gauge placed at the air inlet or by the use of an orifice gauge in the supply main (Chapter X). Care should be taken to ensure that the character of the flow in the main is suitable for accurate measurement if this method is used. Straightening grids may be used in difficult conditions to prevent swirl effects.

CONTROL VALVES

Control valves, as their name implies, regulate the amount of fuel entering the furnace. For every furnace there is an ideal rate of consumption of fuel which is required for the particular heating operation. Any departure from the correct setting of the valve results in loss of efficiency. In intermittent furnaces there is reached a moment at which the soaking stage is being approached, and the rate of heat input must be reduced. It is accordingly imperative that control valves should be provided with suitable indicators, for

example quadrants and pointers, to show the exact setting of the valve. All valves should be examined, cleaned and adjusted regularly to ensure that they are in good working order.

BURNERS

The accurate distribution of temperature is a matter of importance, frequently calling for the application of gaseous and liquid fuels of all types. The burner construction is accordingly a critical feature of the furnace design.

The main distinction of the various types of burners in common use lies in the method of aeration used. The intensity of heating and the form of the flame are governed by the nature of the fuel used, but for any given fuel the nature of the flame and the character of the heat release is controlled by the manner in which the combustible and the air are brought together. The flow characteristics of the two streams are dependent upon the laws of flow of fluids, but in the flame a complex system of changes may be brought about, and knowledge of the circumstances affecting the heat release is so far more or less empirical.

Natural Draught Burners. This type of burner is essentially an injector burner of the bunsen type, the gas being slightly above, and the air at, atmospheric pressure. A proportion of the air is entrained with the gas as primary air, the remainder being induced by the furnace draught as secondary air.

Preheated Air Burners. This type takes the form of one or more ports, and is supplied with preheated air from a heat recovery appliance, a recuperator or regenerator. It is usually used on large furnaces. Burner controls may consist of slide dampers built into ducts both in the air and gas supply systems. The gas is delivered at low pressure, and the air either by natural draught or at low pressure after passage through the preheater. The method of draughting the furnace and of controlling the waste gas dampers has a material influence on the nature of the flame.

Low Pressure Blast Burners. With these burners the gaseous fuel is supplied at pressures between 2 inches and 8 inches w.g., and with air pressures varying from a few inches to 5 or 6 lb. per square inch. They may be used with either cold or preheated air or both. They constitute the most simple and general type. The gas and air supplies are generally separately controlled, and it is advisable to ensure that the control valves are fitted with suitable quadrants and indicators to assist in the correct setting of the valves. They are normally fitted into a cone-shaped refractory burner block, which serves the multiple purpose of assisting the flow of gases to give the correct form to the flame, of assisting ignition at lighting up, and of maintaining combustion by the proximity of a hot radiating surface. With a refractory burner throat the relatively cold flame of a low grade fuel is not so readily blown off the burner. The maintenance of the correct gas-air ratio may be assisted by coupling the air and gas valves together through a proportioning mechanism, but it is essential to have good pressure regulation, as by a pressure governor on the fuel gas. Turbulence and good mixing are assisted by providing deflecting vanes in the annular throat carrying the air to give a vortex motion to the gases.

Crude gas is burnt in both of the above types of burners, but ample capacity must be allowed in the gas passages and valves, where tarry deposits may accumulate and cause stoppages. With hot gas, good insulation of gas supply mains is essential to keep tarry constituents as far as possible volatile, and provision should be made for the draining away of the tar without interruption of operation. Adequate cleaning gates, requiring the minimum of time for clearing deposits, are necessary. Bends should give good streamlining and mixing devices and nozzles should not be used inside the burner. Gas and air should have an ample mixing space. Disc-type valves are preferable for use with gases likely to contain tarry or dusty constituents.

This type of burner readily permits adjustability of flame length by admitting the air in separate supplies, an inner and an outer stream supplied in an axial direction.

The so-called diffusion burners fall into this class. In these the gas and air are made to diffuse slowly into one another, whereby mixing is gradual and the maximum length of flame attained. The ports may be annular, one within the other, when the flame is circular, or they may be long slots producing a flat stream of flame. They are particularly suitable for plate heating when operating on mixed gas.

It is frequently overlooked that air preheat changes the character of the flame, and that where the temperature of preheat is varying a specific setting of a flame cannot be maintained because the air is constantly changing in volume. Accordingly, burners utilising preheated air should be good mixing burners.

High Pressure Gas Burners. This type of burner may be operated in between one and three stages of injection according to the class of fuel and conditions of use. Gas pressures may be from 1 to 10 lb. per square inch. Alternatively, a similar type of burner may be operated by means of pressure air used as an injector to entrain gas supplied at low pressure.

With the high pressure burner the fans are not required, distribution mains are reduced in size, and steady pressures can generally be maintained by the use of a reducing valve as governor, though successful installations are operated without a governor. This system is convenient for use in converting existing furnaces to gas firing in locations in which space is limited, and only the simplest attachment to the furnace structure is permissible. With the elimination of air pipes and fans the risk of explosion is eliminated. The burner, however, lacks flexibility. The air/gas ratio is influenced by the pressure in the furnace and the output of the burner, and the more these factors vary the greater the departure from ideal combustion conditions. To avoid backfiring the lower limit of output is about ½ to ⅓ of full load, so that the burner is not suitable for furnaces requiring to be operated in a wide range of temperature—as, for example, some types of tempering furnace. The pressure of gas required to be available in the burner is less with the lower grade gases on account of the higher gas/air ratio, the required pressure usually being :—

	Calorific value (gross)	Pressure in. w.g.
For coke oven and town gas	500/550	118/196
Mixed gas (coke oven and blast furnace)	200/240	59/118
Producer gas	150	39/59
Blast furnace gas	95	20/39

Single-stage induction can be applied up to the point at which the air requirements of the gas do not exceed 2·3 times the gas quantity. The gases of lower calorific value have a lower gas pressure at which the flame flashes back in the burner, and the lower limit of capacity lies between 25 and 50 per cent. of the highest load according to the type of fuel gas and the furnace temperature. Since coke oven gas and town gas require more than 2·3 times their volume of air two-stage induction is generally applied when burning these gases.

Appliances for the burning of town gas and coke oven gas as a furnace fuel are described in Chapter XXVII.

Pre-mixing Burners. This is a popular type of burner which admits of the gas and air being delivered to the throat ready mixed. It has the advantage that the combustible mixture can be distributed by suitable mains to a number of different points in the furnace to produce the required temperature distribution. The aeration of the gas is effected in an injector mixer by the use of high pressure air, the gas pressure being stabilised by the use of a governor.

This burner is suitable for a very uniform furnace atmosphere, steady heating conditions and a well distributed supply of heat. Alternatively, high pressure gas can be used to induce air.

The velocity of the gas in the distributing mains must be above the velocity of flame propagation, and at the burner inlets to the furnace cooling fins are used to safeguard against backfiring. The arrangement is unsuitable for high degrees of preheat of gas or air. The safe limiting temperature of air preheat is 300° C., though it has been claimed to be practicable to 500° C.

With gases of low calorific value there is a greater tendency for the flame under normal operating conditions to be blown off the nozzle of the burner. These burners have the advantage of simplifying control from the standpoint of the operative. They are readily adaptable to automatic regulation of furnace temperature.

The principle of pre-mixing may also be combined with the use of secondary air at the burner nozzle, either supplied by means of a fan or induced by the draught of the furnace. The use of secondary or tertiary air can be applied in this manner to any type of injector burner. It is readily applied with high calorific value gas because the mixture ratios with such a fuel are appropriate and far above the upper explosive limit, but as each additional stage of air admission is made there follows the attendant loss of simplicity of control.

Combustion in Films and Surface Combustion Burners. Finally, there are the multi-jet burners in which the gas and air are subdivided into a number of small streams, producing in effect a series of films of flame. The flame volume is controlled, and burning of stock obviated. Excessive local temperature in the body of the flame is also avoided.

The modern burner is virtually a multi-holed brick, the refractory surface of which becomes highly heated and so forms an effective radiator. In the latest type the surface of the refractory is formed in the shape of a hemispherical cavity which serves as a concave reflector of radiation and accordingly is a very efficient heating appliance. The ultimate result of the combustion of gases in contact with refractory surfaces is the production of flameless incandescent surface combustion. Air preheating is practicable. The burner brickwork must be made of a high grade refractory ; the fashioning of the burner must be of a high standard of workmanship and accordingly the normal question of care in use and adequate maintenance introduces an operative factor which may be a disadvantage in certain applications.

Control of Furnace Atmosphere. In furnace operations the maintenance of a specific type of furnace atmosphere is frequently essential in order to avoid oxidation of the stock or to preserve a surface condition (see Chapter XVIII). In the more specialised operation such as bright annealing the material may be heated out of contact with the products of combustion, and controlled atmospheres provided by means of an auxiliary appliance. The type of atmosphere used depends upon the class of material heated and the operation involved. The matter is one for specialised study since in the reactions involved many complicated equilibria must be considered.

In fuel fired furnaces for the normal method of maintaining a constant furnace atmosphere with varying rates of fuel supply a constant fuel/air ratio is required. The commonest method of adjusting the furnace atmosphere is that of visual inspection. With some fuels the decision is relatively easy ; a so-called reducing atmosphere is smoky and a haze may be apparent ; a neutral or oxidising atmosphere is clear. With clean gaseous fuels, coke oven gas, water gas or clean producer gas these indications are absent. With certain types of burner the brightness of the flame is a guide, but generally in the absence of any positive means of controlling the fuel and air flow resort must be made to gas analysis.

With gaseous fuels the use of automatic proportioning equipment is the simplest and the most reliable method of control. Mechanical interconnection of gas and air supply is practised, but is not infallible, since with variations of gas and air pressure, fuel quality and furnace draughting the settings of the valves may readily be thrown entirely wrong. Other devices used are :—

(i) Proportional mixers, operated by a positive blower, a fan or a jet, working in conjunction with a gas governor to maintain constant pressure of gas supply. A rich mixture is induced through adjustable ports and secondary air is induced at the burner nozzle. In another type an inspirating Venturi tube acts both as pump and proportioning device. Either gas or air under pressure may serve as the power medium.

(ii) Gas/air controls, applied by means of flow orifices. The flow of gas and air is measured by means of orifice gauges. The pressure head across the gauge is proportional to the square of the velocity of flow for fluids of low viscosity. The pressure drop through the two orifices by acting on a balanced system of diaphragms or floats sets up a pressure differential, which operates a relay either one way or the other to operate the power unit. The power unit operates the control valve governing the variable flow. By setting the differential mechanism by a convenient device the gas/air ratio can be adjusted to any desired value.

Rules of Practice for Burner Operation. It is an important feature of all burner efficiency that certain simple rules of practice should be observed in their operation. For brevity they are summarised as follows :—

(i) Correct setting of the gas and air valves, and the adjustment of the damper to give balanced draught in the furnace is the first golden rule. These settings should be marked on appropriate quadrants.

(ii) Where several burners are operated from one rail the valves on each should be adjusted to give correct heat distribution and the main gas and air rates are controlled on main valves and dampers.

(iii) The installation must be operated at the correct pressures. Pressure gauges should be used.

(iv) Good maintenance comprises adequate cleaning, changing of corroded or damaged jets, free valves, concentric jets or burner throats, correct seating of the burner throat in the refractory port.

(v) Governors, proportioning apparatus and valves should not be so placed as to become overheated.

(vi) All instruments are to be kept clean and in good order. Makers' rules should be observed.

LABOUR-SAVING APPLIANCES

In modern furnaces many new types of mechanical appliances are now becoming an essential feature of the furnace operation, including charging machines, mechanical stokers and the driving mechanism for rotating hearths. Charging machines should be flexible, easily controlled, simple to adjust and quick in their action. In the use of conveyors, temperature must be controlled within specified limits to avoid destruction by overheating. With moving hearths sand seals must be always kept properly filled with suitable material ; otherwise damage to bogie carriage side plates and wheels or serious inleakages of cold air may occur. Foreign matter must not be allowed to drop into the seal to cause excessive friction or dislodgment of brickwork.

IMPROVEMENT OF EFFICIENCY AT FURNACE REBUILDS

Certain salient features have now been discussed relating to the bearing of the furnace construction on general efficiency. Matters of control, such as

adjusting combustion to give the correct mixture of fuel and air, and dampers to permit of the proper setting of the draught, or loading the furnace to the economical limit are all matters of day to day practice, which do not in any way affect the need for modification in the furnace structure. In course of investigations into conditions of operation it may become apparent that improvement can be effected by rebuilding or structural change in the furnace. Counsel to pay attention to possible changes in these respects must be tempered with the advice that since the science of furnace heating involves many complex factors, it is desirable to take the advice of a furnace expert on structural changes.

Attention should be paid to the maintenance of mechanism and furnace brickwork. The cleaning of flues, repair of dampers, maintaining the cleanliness of heat exchange surfaces as in recuperators, and the sealing of cracks in brickwork, all have an important influence on the amount of fuel consumed. An unsuitable size of grate, a furnace too large because it has been built for oversize material rarely handled, crowns too high or too low, flues of insufficient capacity or defective design, unnecessary burners, wrongly disposed and dimensioned ports, fans of an uneconomical capacity, unsuitably dimensioned combustion chambers, insufficient lagging of doors and furnace structure, are all capable of being corrected during the course of normal maintenace. Many of these defects are to be found even in modern furnaces because users are still disposed to look only at first cost. A robust, well-built and insulated furnace is always the cheapest in the long run.

INSTRUCTION BOOKS

Finally, in furnace or kiln operation, there should be compiled at the installation of the plant a handbook of guidance to the plant operators and engineers, in which should be specifically stated the essential "rules of practice" for operating the plant, and the points to be watched in maintaining the plant in good order. Such handbooks are now prepared by the best constructors and, if consistently used by successive operators, are most valuable instruments in the maintenance of efficient practice. They should be adequately illustrated by simple sketches, and expressed in brief and direct terms.

FURNACE TESTING

A number of British Standard Specifications have now been prepared for the determination of the performance of various types of furnace. These give guidance as to the technique to be employed for the measurement of fuel consumption, throughput and heat losses. The methods to be employed for the determination of the heat balance are indicated. The interpretation of the results, however, is quite another matter.

INFERENCES FROM THE HEAT BALANCE

The normal method of control of heating furnaces is based on the use of graphs of fuel consumption and rate of output. (See Chapter XXIX.) In order, however, to establish improved practice by modification of the furnace operation or design, detailed analysis of the heat balance is necessary.

Since—

heat supplied = useful heat + structure loss + exhaust gas loss,

it is apparent that for a constant fuel consumption and a constant structure loss an increase of useful heat can only take place by reduction of the exhaust gas loss. The value of a knowledge of the temperature and composition of the gases leaving the furnace accordingly becomes apparent.

In batch or intermittent furnaces, since the temperature of the furnace

passes through a cyclical change both the structure loss and the exhaust gas loss vary and there must be added also a knowledge of the storage and external heat loss.

STRUCTURE LOSS

The structure losses become more important the higher the furnace temperature, and heat protection is then all the more necessary. Whether inner linings can be replaced by hot-face insulation or first quality brickwork insulated by standard external insulation is mainly a question of temperature, and amount of wear and attrition imposed by the process on refractories. Furnace crowns over fireboxes operating reheating furnaces cannot be insulated unless high-grade refractories are used. Insulation may only result in uneconomical refractory wear since the mean temperature of the inner lining is raised by the use of the external insulation. In pulverised fuel firing the reaction of possibly molten ash with refractory material imposes a limitation on the method of building. Expedients to overcome any disabilities of this kind include the provision of patched linings of dolomite, magnesite or siliceous materials in combustion chambers, and the construction of flue off-takes of ample size and accessibility to remove accretions of ash and clinker.

In interpreting the characteristics of the refractory it should be borne in mind that the safe limit of under-load refractoriness is in general 100° C. lower than the value shown by the normal laboratory test, since at high temperatures refractories tend to behave as semi-viscous bodies, and with the passage of time tend to flow, so that the structure may collapse. Alternatively a greater thickness of refractory brick may be used which will reduce conduction loss and at the same time give greater stability to the structure. This technique applies in the use of both suspended and arched crowns.

Artificial cooling either by means of water as in jambs and partition walls or compressed air leads to substantial losses. Except in exceptional circumstances compressed air is a wasteful and expensive means of controlling the flow of heat. The temperature of the external surface of a $4\frac{1}{2}$ inches firebrick wall can be reduced from 400° C. to 40° C. by means of a jet of compressed air, and the temperature gradient through the wall when it is maintained at a temperature of 1,400° C. accordingly increased from 1,000° C. to 1,360° C., thereby causing an increase of 36 per cent. in the heat transfer.

The storage loss has been discussed in Chapters XVIII and XXII. This loss and the wall loss by conduction are complementary for the lower the storage loss brought about by reducing the thickness of the wall the greater the conduction loss. The relative merits of each case must depend upon the period of the heating cycle, whether the heat stored has time to penetrate the wall before the heating operation is completed. To determine when the loss by conduction overtakes the loss by storage it is noted that equilibrium of heat conduction begins to occur when $\frac{L^2}{4Dt}$ approaches 0·4,

where L = thickness of the wall (metres)
t = time since the beginning of the alteration in temperature, hours
D = thermal diffusivity, sq. metres/hour.

EXHAUST GAS LOSS

In all furnaces the use of the continuous principle is conducive to a low exhaust gas temperature, but in batch furnaces the gas must leave the furnace at a temperature above that of the stock. Particular care must accordingly be taken to obviate air inleakage. How far this temperature must be above the temperature of the furnace depends upon the character of the heat transfer

from the gases. Reference to Chapter VIII will show that in this type of furnace the governing factor is gas radiation, which outweighs the influence of convection. Gas radiation, apart from the composition of the gas, depends only on the thickness of the gas layer and the temperature. Since the temperature is fixed by the operating conditions it must be to the advantage of the furnace economy that there shall be within it large radiating masses of gas. This involves the use of relatively large furnace chamber capacity, providing that such provision is not at the same time outweighed by an increased structure and external surface loss, and that layering of the hot gas below the roof does not give a cold layer of air adjacent to the charge. Such a cold layer of gas forms an insulating layer since it is impenetrable to the dark heat rays of the hotter gas above. Accordingly the expedient of raising the crown of a furnace to give a better radiating gas layer and an increased surface of direct wall radiation will fail if care is not at the same time taken to ensure complete gas circulation and prevent air infiltration. A ribbed roof construction to create turbulence, as well as to increase heating surface, is a beneficial form of construction in this respect. The use of auxiliary burners and turbulent admission of secondary air are also expedients used to effect the same advantages.

Heat transmission coefficients in industrial furnaces of the continuous push type vary from 50 B.Th.U. per square foot per hour per °F. at 2,200° F. to 20 at 1,480° F.* for gas layers of about 1 foot 6 inches in thickness. The reduction of the heat transfer coefficients with almost the fourth power of the gas temperature produces an economic limit to the lengthening of the chamber of continuous furnaces. It may then be more economical to use the metallic recuperator to preheat the air for combustion, in which the advantages of convection transfer under conditions of forced circulation give a better utilisation of heating surface.

Summarising the implications of this discussion of the heat balance it may be concluded that fuel efficiency is promoted by the following :—

(i) The provision of adequate heating surface.
(ii) The largest practicable space for combustion and the heating gas layer.
(iii) The prevention of cold gas layers by means of auxiliary apparatus for creating turbulence.
(iv) Prevention of infiltration of air by making the furnace as gas tight as possible and by control of draught.
(v) A critical rate of throughput of material to be heated which is determined by the point at which the combined losses due to the structure absorption and the sensible heat in the exhaust gases are at a minimum value. This is consistent with a good performance in regard to the data discussed later.

EXHAUST GAS LOSS BY UNBURNT COMBUSTIBLE

It has already been indicated that unburnt combustible can result in serious loss of furnace efficiency. Thus in the combustion of blast furnace gas 5 per cent. CO + 1 per cent. H_2 in the waste gases is equivalent to a loss of 25 per cent. of the potential heat of the gas, and 1 per cent. CO is equivalent to an increase of about 90° C. in the flue gas temperature.

Combustion control can be assisted by the use of secondary and tertiary air introduced at positions in the flow of the gas stream, provided that the change is not endangered by excess air or a flame too strongly reducing (i.e. below a temperature of 900° C.). The same effect can be attained by auxiliary burners carrying the needed excess of air. Other effective measures are the use of air-gas controls.

* Metric equivalents approximately 250 Kg. cal./m³/h/° C. at 1,200° C. to 100 at 800° C.

For certain types of operation where low temperatures and good temperature uniformity are required long flames are used involving slow diffusion of volatile combustible and air. These are the most difficult to control and without late additional air admission may be subject to the disability of local overheating.

LOSSES UNACCOUNTED FOR

In the heat balance sheet there is always an item to cover the unascertained losses which are not readily measured, and the balance of error in making the determination of each specific loss. These include :—

(i) Loss in foundations.
(ii) Gas leakage.
(iii) Losses from open doors.

(i) *Losses in Foundations.* The loss in foundations is a somewhat special study, which has been taken up by J. D. Keller (*Amer. Soc. Mech. Engrs.*, 1927–8, 49–50. F.S.P. 50–57). The total flow of heat through hearths of a given shape was found to be proportional not to the area, but only to the diameter or width. The conception of an equivalent thickness of hearth was taken to be that thickness of a wall of the same material, through which the rate of flow of heat would be just the same as the actual rate through the hearth if one surface were held at the temperature of the interior of the furnace, and the other at the temperature of the surrounding air.

In unventilated hearths the equivalent thickness of the hearth at its centre was found to be slightly more than half the least width of the hearth. The average equivalent thickness referred to the whole area of the hearth was one-quarter the hearth diameter for circular furnaces, 22½ per cent. of the width for square hearths, and its value increased as the ratio of the lengths of the sides of rectangular furnaces, approaching a limiting value of 27 per cent. of the least width for very long hearths.

Formulæ have been proposed for determining the temperature penetration into the ground. From them can be deduced the continuous nature of the heat saturation, which can lead to failure of foundations, the risk of which is not entirely eradicated by insulation. That is to say, insulation only delays the heat penetration ; it does not stop it. Accordingly high temperature furnace hearths should always be ventilated.

(ii) *Gas Leakage, and* (iii) *Losses from Open Doors.* In reheating furnaces the gas pressure difference produced in a column of hot gas due to its buoyancy as against the same height of the outer atmosphere amounts to approximately 0·012 inch w.g. per foot (1 millimetre w.g. per metre) height. This buoyancy effect of hot gas must accordingly produce a positive pressure at the top of a furnace door, if the gas pressure at the sill level is to be zero, and no air be allowed to be drawn in. Alternatively, to balance the draught to give zero pressure at the top of the door is equivalent to carrying a negative pressure, that is an indraught, at the sill level. Thus, when the door is opened either an indraught of cold air or an outblast of hot gas results. Generally both occur.

Further reference to the magnitude of the radiation loss from hot surfaces (Chapter VIII) permits an estimate to be made of the direct heat loss derived from this source. Accordingly the heat losses due to the repeated opening of furnace doors may be considerable, and attention to this phase of the operation is imperative (Chapter XVIII). The importance of gas tightness and correct balancing of the draught is again apparent.

Having thus discussed the general principles of furnace construction and operation, attention will now be given to specific types of furnace.

I. MELTING FURNACES

Melting furnaces may be of hearth, crucible or shaft type. The technique of operation invariably depends upon special technical considerations, peculiar to each process, and frequently these have an over-riding influence on the manner in which the melting operation is conducted. It is therefore only possible here to review those features which are common to melting furnaces as a class.

Hearth and crucible furnaces for those processes requiring the highest temperatures, for instance steel making and glass manufacture, are frequently combined with regenerators or recuperators. In shaft furnaces the fuel is mixed with the charge, and the air for combustion blown in at tuyeres.

Apart from problems of mechanical design, economical performance depends upon three major features :—

(i) The design of the melting chamber, its shape and dimensions in relation to the rate of melting desired, and a critical rate of heat input up to the limit at which the refractories are able to maintain the mechanical stability of the furnace. Accordingly, the search for economy must be made by the study of the dimensions in relation to the rate of output required, and determinations of the critical rates of heating necessary for each set of conditions ruling. This means that importance must be attached to surveys of draught conditions and methods of firing.

(ii) Efficient combustion is of paramount importance since high rates of melting require the maintenance of certain flame temperatures, and whether conditions are satisfactory or not may depend upon a relatively small temperature gradient between the flame and the material heated. Accordingly, preheat of fuel and air becomes critical, and regenerative or recuperative methods must be efficiently applied. Further, suitable port or burner design and the correct disposition of the stream of hot gases in the working chamber may become equally essential.

(iii) Process factors, of particular importance in steel making and metallurgical melting generally, which may introduce the over-riding factors referred to above, are outside the present compass.

In general in hearth and crucible furnaces a fundamental relationship exists between the rate of fuel input and the rate of melting for any particular class of furnace. In Figs. 144 and 145 illustrations are taken from steel making practice in open hearth furnaces. The exact nature of this relationship depends upon a multitude of factors, including the furnace design and that of its auxiliaries as well as the many variables introduced by the process. The determination of this relationship for any furnace investigated is a first step in the study of its fuel efficiency.

The determination of the heat balance can be made according to the methods outlined in the specifications prepared by the British Standards Institution. From the heat balance an indication can be obtained of the directions in which improvements can be made to economise fuel. In hearth furnaces, for example, it can be learnt that apart from heat in the waste gases the greatest loss of heat occurs in the furnace structure, and the deduction is that the solution of the problem lies mainly in the refractories used. Heat losses have been further utilised by the application of the waste heat boiler to the hot gases leaving the furnace system. A typical series of heat balances is shown in Table 91. A line diagram of a typical fixed furnace is shown in Fig. 146.

Modern methods of furnace control involve the use of instruments. Instruments virtually maintain a constant indication of the critical rates of input and of draught and temperature conditions which amount to a watch over those features of the heat balance of most importance in determining efficient operation. The instruments used, in order of importance, are :—

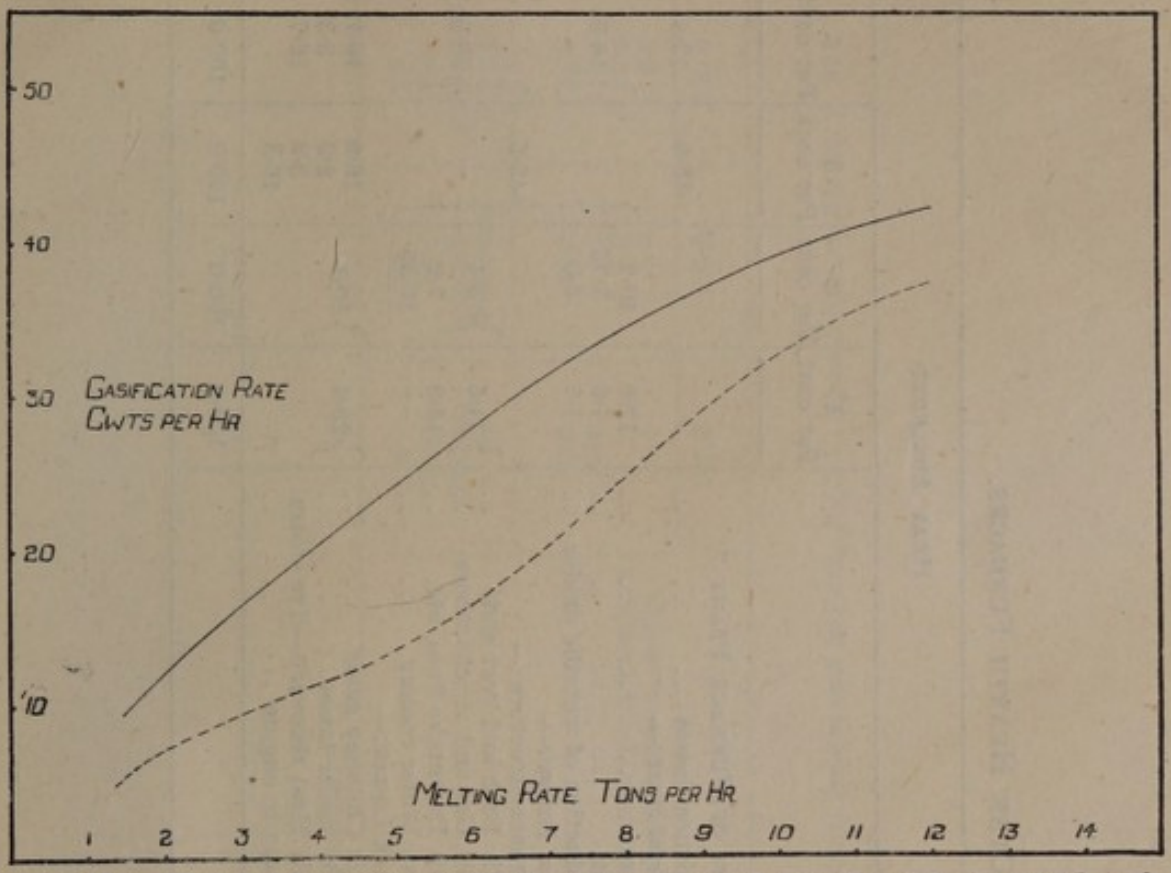

FIG. 144. Relation between rate of fuel consumption and rate of smelting in open hearth furnaces. The band between the curves shows the region in which results have been recorded.

(*J. Iron and Steel Inst.*, Special Report No. 22, Symposium on Steel Making, pp. 248, 539.)

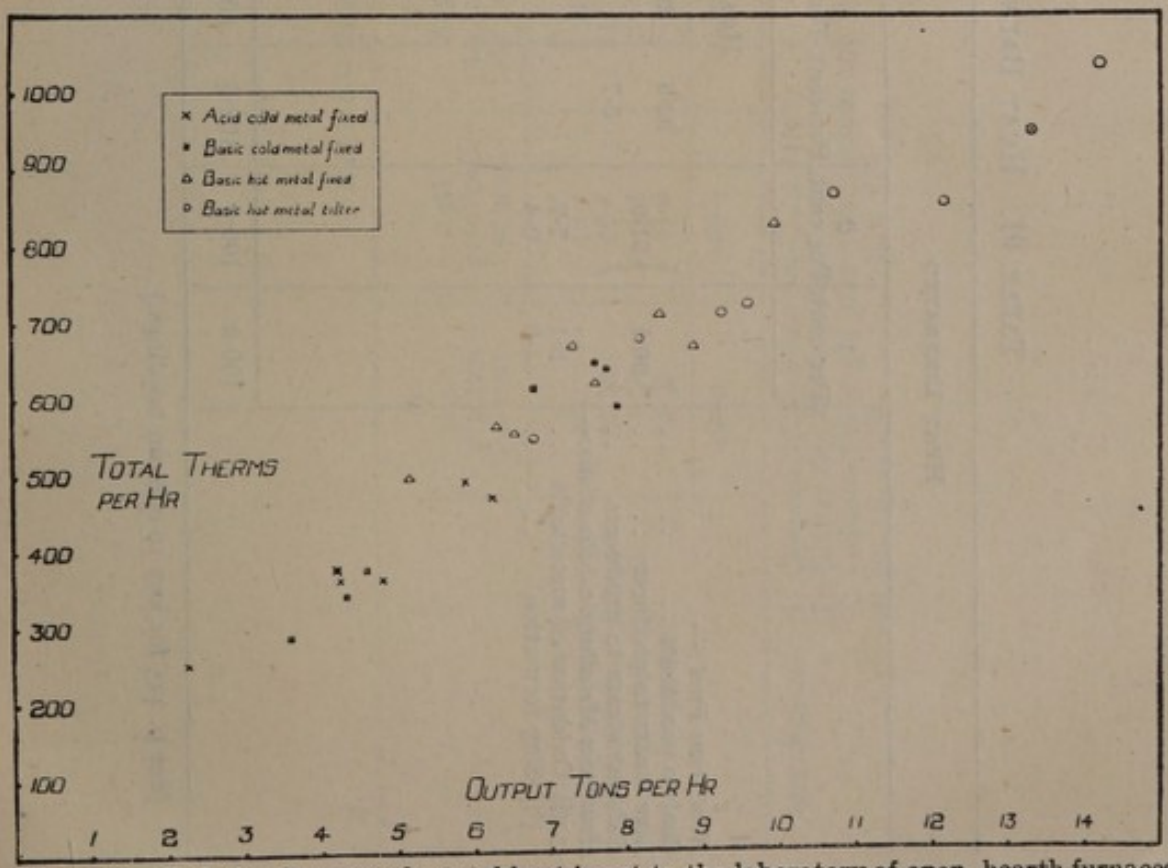

FIG. 145. Relation between the total heat input to the laboratory of open hearth furnaces and rate of output.

(*J. Iron and Steel Inst.*, Special Report No. 22, Symposium on Steel Making, "British Open Hearth Data and their Correlation," O. H. Committee, Iron and Steel Industrial Research Council.)

Table 91. Heat Balances of Open Hearth Furnaces.

Heat Liberated	1 Per cent.	2 Per cent.	3 Per cent.	4 Per cent.	Heat Absorbed	1 Per cent.	2 Per cent.	3 Per cent.	4 Per cent.
					Heat Balances of Furnace Plant				
Heat from Fuel :—					Producer losses			15·6	13·4
Coal to producers	90·1	91·9	94·3	94·5	*Useful Heat* .—				
Live steam to producers					Steel	12·8	10·2		
Exhaust steam to producers ..			5·7	5·5	Slag	7·3	7·75		14·9
Heat from Metallurgical Reactions:—					Reduction of metallic oxides ..	1·7	6·0		
(i) Oxidation of metalloids ..	8·1	7·7			*Heat Losses* :—				
(ii) Slag formation	1·8	0·4			Furnace structure :—			43·5	
					(i) Bath and port ends ..	34·5	3·4		33·6
					(ii) Gas and air chambers ..				
					(iii) Downtake flues, etc. ..	15·3	7·2		
					(iv) Water cooling	—	15·35		
					Stack Losses :—				
					(i) Chimney gases	28·4	50·1	16·6	16·1
					(ii) Boiler losses			2·2	3·3
					(iii) Heat recovery—live steam			5·8	18·7
					Steam to engine			16·3	
	100·0	100·0	100·0	100·0		100·0	100·0	100·0	100·0

[See p. 423 for key to column headings.]

HEAT BALANCES OF MOLTEN BATH

	5 Basic practice Per cent.	6 Acid practice I Per cent.	7 Acid practice II Per cent.		5 Basic practice Per cent.	6 Acid practice I Per cent.	7 Acid practice II Per cent.
Thermal efficiency based on heat input at furnace valve	*16·1*	*10·1*	*8·76*	Steel	80·2	91·2	81·5
Oxidation of metalloids ..	32·25	57·3	44·1	Slag	15·4	4·0	8·9
Slag formation	4·50	1·2	1·9	Decomposition of limestone		1·0	0·9
Balance from fuel ..	63·25	41·5	54·0	Reduction of iron oxides	4·4	3·8	8·7
	100·0	100·0	100·0		100·0	100·0	100·0

REFERENCES : (1) Clements, *J. Iron and Steel Inst.* (1922).
(2) Kinney and McDermott, South Works, Illinois Steel Co.
(3) Combination of gas producers, furnace and waste heat plant, Barnes and Sarjant, *J.I.S.I.*, Special Report No. 22, p. 271.
(4) *loc. cit.* p. 272. (5) p. 273. (6), (7) p. 274.

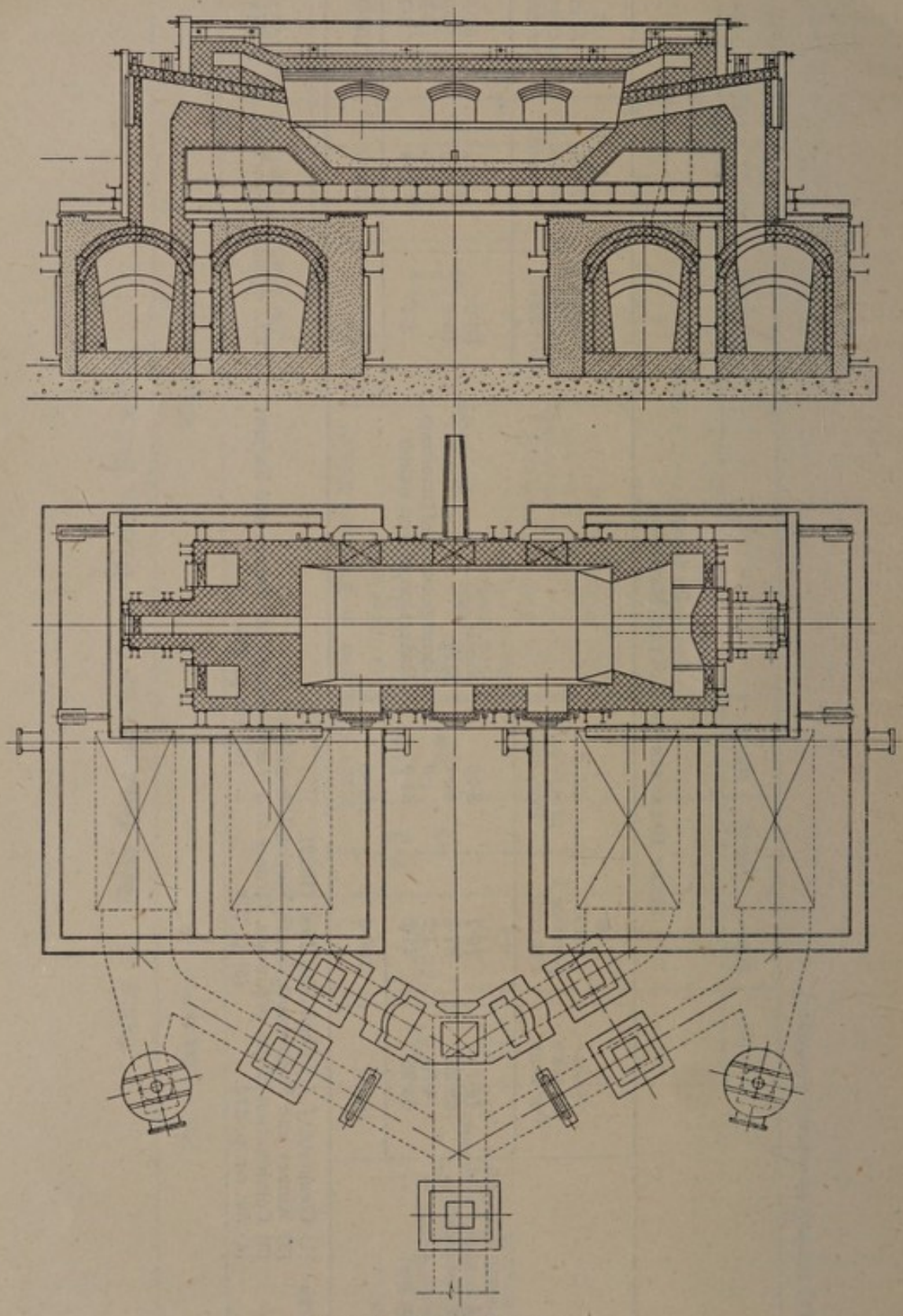

FIG. 146. Open hearth furnace.

(i) Fuel measuring appliances and pressure and draught gauges. These serve to assist the maintenance of the correct rates of firing and conditions of balanced draught necessary to avoid wasteful leakage of gas and air.

(ii) Temperature control of the molten bath or of the furnace lining by means of thermocouples, radiation and optical pyrometers which are of use from the aspect of process as well as fuel efficiency.

(iii) Indicating and recording pyrometers for observing the temperatures in the regenerative or recuperative system. Such instruments not only indicate unsatisfactory temperature conditions in degree, but also in distribution, and ensure the correct timing of the reversals in regenerative practice.

Automatic controls for correct admixture of fuel and air, for correct draughting and for reversal in regenerators, are coming gradually into use as the technical difficulties of applying delicate instruments to heavy industrial processes are being overcome.

FUEL SAVING IN OPEN HEARTH FURNACES

Notes here given were designed for practical guidance in the operation of hearth furnaces, in particular for steel making, but they have a general bearing on all types of regenerative hearth furnaces.

The problem of saving fuel in open hearth furnace practice is inseparable from problems of steel output and refractory consumption. It does not necessarily follow that using less fuel is saving fuel in the true sense. Eight tons of steel an hour from 40 cwt. of coal an hour may be more economical than 7 tons of steel from 38 cwt. of coal, and *is* more economical unless the furnace brickwork suffers in consequence. Apart, however, from alterations in furnace design and construction, which are not within the present scope, there are points to be noted in practice which tend to reduce the fuel consumption without adversely affecting output or brickwork and may benefit either or both. A number of these are set out below.

(1) Gas pressure should be kept at the lowest possible working limit.

(2) Culverts, valves and furnace brickwork from producers to gas ports should be examined regularly, and repaired when necessary. There should be little or no leakage at these places.

(3) Combustion in the furnace needs special consideration. Correct mixture of gas and air is essential to give the maximum heat to the bath. If the air and gas are not measured, or the ratio of air to gas not indicated on an instrument, the flame should be closely watched. It should be highly luminous and not too long. The tip should on no account reach the outgoing block. Too long a flame results in damage to the outgoing ports and tends to overheat slag pockets and chequers.

(4) A slight pressure but not an over-pressure should be kept in the furnace. A slight pressure prevents excessive cold air being drawn in, and saves the amount of fuel necessary to heat the air up to furnace temperature. An over-pressure forces too much flame through openings in the furnace structure, wasting heat outside the furnace and damaging structural steel and brickwork. Normally, a slight flicker of flame visible at the doors is used as an indication of correct pressure. Dampers are the means of controlling the pressure. They should, therefore, be arranged to work easily and should be adjusted according to the amount of gas being burnt. The more gas used the greater the quantity of waste gases formed, and consequently the wider the damper must be opened to give the extra draught required to clear the furnace.

(5) Gas ports and slopes should be fettled frequently in order to maintain correct alignment and flame direction. The importance of this is often not fully realised, but can easily be proved by trial.

(6) Reversals should normally be made at regular intervals, although this procedure should be modified if the chequer temperatures have become unbalanced. The object to be aimed at is even heating across the bath and this is much easier to attain if the corresponding chequers at each end of the furnace are approximately equal in temperature. Temperature recorders and controllers installed in the chequer system greatly assist the melter in balancing operations.

(7) As much information as possible relating to the draught in the furnace system should be obtained. If the pressures and draughts are known at various points when the furnace is going well, the knowledge can often be used to diagnose complaints when the furnace operating condition is unsatisfactory. For this purpose holes should be cut through the brickwork and fitted with easily removable plugs. Readings with a draught gauge may then be taken regularly. Suitable points for the test holes are :—

(*a*) In gas and air uptakes—say a foot or so above stage level.
(*b*) In gas and air slag pockets.
(*c*) Above gas and air chequers.
(*d*) In gas and air culverts as near base of chequers as practicable.
(*e*) In the stack flue near the damper and on the furnace side of the damper.
(*f*) Before and after the main gas valve, for obtaining drop in pressure across the valve.

As examples of information obtainable from such readings, if the difference between the pressures before and after the main gas valve is higher than usual, the flow of gas through the valve is being impeded and the valve housing needs inspection and clearing. If the difference between the waste gas draughts taken above and below a set of chequers is greater than usual, the chequers are becoming choked.

(8) CO_2 recorders are of doubtful value when applied to the waste gases of an open hearth furnace. But analyses should be taken to check correct combustion. If, about an hour after the furnace has been fully charged, Orsat samples are taken through a silica tube inserted into the gas downtake, reliable figures may be obtained. Normal CO_2 content at this point is 15–16 per cent. when operating on producer gas. Care must be taken that no air is drawn into the uptake around the silica tube.

(9) Instruments to record draughts and pressures continuously are very useful especially when simultaneous readings taken at two or more points are necessary for accurate comparison. But for a permanent installation it is probably better to have quite a small number of recorders, including one showing the gas pressure and another the effective stack draught, rather than a multitude which may confuse rather than inform.

It cannot be stressed too strongly that improvement in fuel consumption of open hearth furnaces depends as much on past as on present knowledge. Data concerning a particular furnace collected when it is going badly may be of little or no value unless they can be compared with similar data obtained when it was comparatively new and efficient. Someone should be responsible for the collection and collation of information. Fuel consumption, steel output, gas pressures and draughting conditions should be observed and set down weekly. This can all be done quite quickly once a routine is established, and the results, arranged as tables, curves or in any other way suitable for easy comparison, will soon be regarded as essential to the attainment of minimum fuel consumption.

CRUCIBLE FURNACES

Crucible furnaces are fired by coke, gas or liquid fuel, the main consideration in high temperature work being that the fuel must be capable of giving a high intensity of combustion since the quickest melting is usually the most economical from the standpoint of economy of both fuel and metal.

COKE FIRING

The size and quality of the coke is a factor of major importance. The manner of combustion of coke is related to its physical properties, the harder cokes in a limited supply of air burning with a higher ratio of carbon dioxide to carbon monoxide in the products of combustion, and so giving a higher intensity of combustion. Such cokes of low reactivity are the more suitable for use in high temperature melting operations, such as are carried out in crucibles, reverberatory furnaces and cupolas.

For a discussion of reactivity and combustibility of coke the reader is referred to the literature. It is a subject to which considerable investigation has been devoted. One of the best means of revealing a hot melting coke is the determination of the shatter test.

When coke is burnt on grate bars its ash content, its refractoriness, suitability of grading or size analysis, the draught available and the critical rate of combustion required to give quick melting must all be considered in relation to the work in hand. In the older type of natural draught crucible furnace or pothole, fired by coke, due to limitations of design the old beehive coke gave the most satisfactory and economical melting at the highest temperatures, but with adequate draught and a suitable size of coke, by-product oven coke is equally effective, provided a clean coke of low reactivity and good shatter resistance is used. The sulphur content of the fuel must be limited.

The quantity of coke charged at each feed and the draught must be adjusted to follow the course of the heat, so that fresh charges are not required just as the heat is nearly ready.

A suitable specification by way of guidance on general principles only for a coke of good melting quality is as follows :—

Moisture.	As low as possible.
Ash.	8 per cent.
Sulphur.	Under 1 per cent. (Purity and good shatter index permits a higher sulphur content to be carried.)
Shatter index.	1½ inch index as high as practicable and preferably over 90 per cent.
Size.	Less than 6 per cent. through 2 inches, the remainder being as close graded as practicable for the particular size desired.

GAS AND OIL FIRING

With oil- or gas-fired crucible furnaces the use of the correct rate of fuel consumption and fuel-air ratio is of prime importance. Fuel is wasted if too much or too little fuel or air is used. The optimum conditions are indicated from the shape and colour of the exhaust flame, and gauges indicating flow, pressure and draught are called for to assist control. A burner out of line will readily cause waste of fuel, as well as spoilt metal.

The size of crucible, and shape and dimensions of the combustion chamber and off-take flues are primarily a matter of initial design, but in many operations are amenable to alteration at rebuilds. Preheater tops, made from an old crucible are sometimes employed with the object of getting the whole of the charge into the furnace at the start of the heat. Worn linings should be constantly kept in repair, carbon deposits prevented from forming in combustion chambers and all leakages avoided.

SHAFT FURNACES

The most important shaft furnace is the blast furnace for the smelting of metals from their ores, but since this type of furnace is of specialised type, and its operation is bound up closely with a complicated metallurgical operation, it is not considered in a volume intended for general study.

Another important type of shaft furnace for melting purposes is the cupola furnace.

THE CUPOLA FURNACE AND FUEL ECONOMY

The cupola, when compared with other coke-fired melting furnaces, has a

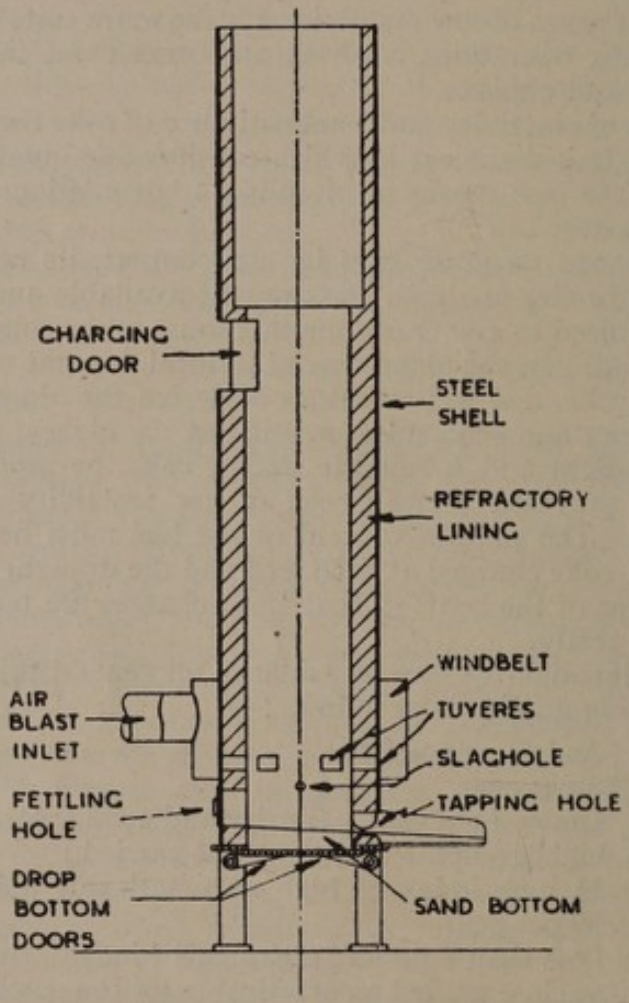

FIG. 147. Drop bottom cupola.

relatively high thermal efficiency and because it will give reasonably satisfactory results even when inefficiently operated, it is sometimes found that little attention has been paid to design and operation. It is intended here to show how design and operation can affect the thermal efficiency of cupolas and consequently their fuel consumption.

DESIGN

The cupola (Fig. 147) consists of a mild steel shell erected in a vertical position and lined with refractory. The shell is mounted either on a brick foundation or on steel columns. The steel column arrangement is used on most modern cupolas and the bottom of the shell is provided with drop-bottom doors, through which the debris, consisting of coke, slag, etc., can be discharged at the end of a melt.

In drop-bottom cupolas, the working bottom is made up with moulding sand which covers the drop doors. This bottom slopes towards the metal tap-hole at the front of the cupola, the tap-hole being at the lowest point. The sand bottom is put in through a hole in the shell at the back of the cupola opposite the metal tap-hole. This is known as the fettling hole or cleaning-out door, and in solid brick-bottom cupolas is also used for raking out the debris at the end of the melt.

Tuyeres for supplying the air blast are provided at a height of between 2 and 4 feet above the working bottom. The air is delivered to the tuyeres from a wind belt, which surrounds the shell. A high pressure fan, or a blower of the Roots type, supplies the air to the wind belt through a blast pipe. A valve is provided in the blast pipe so that the air supply may be controlled.

A slag hole is located at a level about 9 inches below the centres of the tuyeres. A charging hole is provided through which metal, coke and flux are fed into the furnace, and this is situated 10 to 20 feet above the tuyeres, according to the size of the furnace. The shell is usually continued for 15–20 feet above the charging hole to form a chimney.

OPERATION

In preparing the cupola for a heat, a fire is lighted on the working bottom and a bed of coke built up to a level just above the tuyeres. This coke bed must be thoroughly hot before it is finished off to its final height. The fire may be drawn up by natural draught, or a forced draught may be used, supplied by a small auxiliary blower or by means of kindling tuyeres from the wind belt. As soon as the fire is well alight, the fettling hole is closed up with sand and a cover plate fixed over it. The coke at the tuyeres must be well heated before charging begins. It is advantageous to finish off the bed with about 12 inches of cold coke to bring it to its correct level. This level should be checked by means of a measuring stick inserted at the charging hole. When the bed is completed, the cupola should be charged full to the sill of the charging hole before the blast is put on. Alternate charges of metal, coke and flux are used until the cupola is full. When the metal in the cupola starts melting, the rate of charging should be equal to the rate of melting, so that the furnace is kept full throughout the heat.

The metal should begin melting in five or six minutes after the blast is put on. The molten metal may be collected in the well of the furnace below the tuyeres and tapped off as required, or allowed to run from the furnace as it melts and collected in a ladle or receiver. When slag accumulates in the well, the slag hole is opened and the slag run off, preferably into a bogie for easy removal.

At the end of the melt, charging is stopped and the blast is kept on until all the metal has melted. A careful watch should be kept at the tuyeres and the blast should be shut off immediately it is seen that no more metal is melting. If this is not done, coke which could otherwise be recovered is wasted. By far the greater part of the damage to the lining takes place during this emptying or " blowing down " period, and unless the blast is shut off as soon as all the metal is melted, the wear on the lining will be increased.

As soon as the blast is shut off, the debris should be discharged from the cupola, either by dropping the bottom doors, or, in solid-bottom cupolas, by removing the fettling hole cover and raking out. The debris should be immediately quenched with water, using an ample supply. Care should be taken to see that it is completely black before the water is turned off. It is frequently found that the draught through the cupola will draw up any remaining embers and re-kindle the coke, which will then burn away during the night. When the debris is cold, usually the next morning, it should be hand picked and the

coke remaining returned for use in the cupola. This coke has not suffered in any way by being used in the furnace, tests having shown that it is not contaminated by any deleterious elements and is quite suitable to replace new coke on the charges.

TUYERES

The total area of the tuyeres should be one-fourth to one-seventh of the cross-sectional area of the cupola inside the lining at tuyere level. The shape of the tuyeres, whether round, rectangular, flared or parallel, has little influence on the efficiency of the furnace, so long as an adequate area is provided inside the lining. Some cupolas are provided with tuyeres in one row ; others have two or more rows. In the balanced blast and Poumay types, using multiple rows of tuyeres, the tuyeres in the upper rows are provided to admit air for the more complete combustion of the coke to carbon dioxide. In other types, the use of a double row of tuyeres means only that an additional quantity of coke is required for the bed each time the cupola is used. Unless a specially high pick-up of carbon is required, necessitating the use of a deep bed, this is wasteful and increases the fuel consumption of the furnace.

For the melting of high steel charges for the production of steel by the converter process, for example, a high carbon content is not required, and tuyeres arranged in a single row will give the best results.

INTERNAL DIAMETER AND HEIGHT

The internal diameter of the furnace at the tuyeres and in the melting zone controls the output obtainable from the cupola. This assumes, of course, that the air supply is adequate for the operating diameter. In normally operated cupolas, an output of 0·7–0·8 ton per hour is obtained per square foot of cross-sectional area in the melting zone. When melting high steel charges, however, the rate of melting is reduced, and it is also desirable to operate the furnace more slowly than is possible on pig and scrap-iron charges. The output when melting charges containing 80–100 per cent. steel is seldom more than 0·5 ton per square foot cross-sectional area per hour.

The shape of the lining is important and is governed by several factors. In cupolas which are tall between tuyeres and charging hole, a parallel lining from base plate to charging hole may be used, and this also holds for cupolas where a low output is required for the internal diameter. In short furnaces, and where a high rate of melting is required, a boshed lining is desirable. The holding capacity of the furnace for solid metal, coke and flux to a large extent controls the temperature at which the exhaust gases are discharged. The smaller the amount of stock held in the cupola, the hotter the exhaust gases and the greater the loss of heat from the fuel used. Except in very small cupolas, i.e. less than 24 inches internal diameter, where there is a danger that the stock will bridge and hang up in the furnace, the preheating zone above the coke bed should hold metal, coke and flux sufficient for one hour's melting.

Many cupolas are installed which are short between tuyeres and charging hole and owing to the short preheating space, the gases escape at a very high temperature and the thermal efficiency is relatively low. Conditions in these designs can be considerably improved by increasing the internal diameter from a level near the top of the bed coke and maintaining this increased diameter up to the charging hole. The thickness of the brickwork in the expanded portion must not be reduced below, say, 6 inches on cupolas of 5 tons capacity, or 9 inches on large cupolas. The area in the expanded portion may be 30–40 per cent greater than in the tuyere zone. The flare from the smaller to the larger diameter should be gradual and is usually made about 2 feet 6 inches deep.

Because of the loss of heat when the stock column is short, it is important that the cupola shall be kept charged full throughout a melt.

A boshed lining makes it a little more difficult to separate different mixtures when irons of a number of different compositions are melted in one heat. It is not desirable, also, when a cupola is completely emptied and re-charged several times a day. Because of the additional bed coke required and the severe wear caused at the lining, this latter method of operation should be avoided whenever possible.

An alternative to increasing the internal diameter in the preheating zone is to increase the height of the charging hole, but this usually necessitates major alterations to the charging platform and hoisting equipment. However, in some cupolas, operating for long periods, which are very short to the charging hole, the alterations necessary to enable the furnace to accommodate one hour's supply of metal in the preheating zone are well worth consideration.

REPAIRS TO LINING

During every melt in a cupola, part of the lining, especially that in the zone of maximum temperature immediately above the tuyeres, is slagged and eroded away. This material must be replaced after each melt by patching with ganister and/or firebricks, ramming round a former, etc. Whichever method of repair is adopted, the furnace should be brought back to the same internal dimensions each day. Slipshod repair work causes increased fuel consumption as well as short life of the lining and troubles in operation. If the repair work is poor and insufficiently dried out before the cupola is put to work, the new material will rapidly come away and heat must be consumed in converting it into slag.

A more important point is that if the internal dimensions are allowed gradually to become larger, the amount of coke used on the bed is increased. The efficiency of the furnace may also suffer, thus necessitating the use of more coke to produce metal of a suitable temperature. Whether the furnace is being patched to its correct dimensions can be found by occasionally checking the weight of coke required to bring the bed to its correct height.

HEIGHT OF COKE BED

The height to which the bed coke is built up above the tuyeres is an important factor in controlling the temperature of the metal from the first tap and also from subsequent taps. Other things being equal, a low bed will yield cooler metal than one which is high. A bed height of 2 feet above the tuyeres, that is, above the top row of tuyeres if two rows are used, is somewhere near the minimum height which may be employed. Where extremely hot metal is required, or where a high pick-up of carbon is desirable, a bed height of 4 feet may be necessary. In exceptional cases, as for example in producing metal for whiteheart malleable castings, an even higher bed may be required and it is, of course, false economy to jeopardise the temperature and quality of the metal by starting with a bed which is too low.

METAL AND COKE CHARGES

The weight of the metal charges has some influence on the efficiency of the furnace. A large number of light charges will, under certain conditions, give a lower thermal efficiency than a smaller number of heavy charges. No hard and fast rule can, however, be laid down regarding the size of charges for various sizes of cupolas as this depends largely on the type and quality of the metal being melted. The heaviest charge a cupola will melt successfully is about one-sixth of its hourly output, but it is often advisable to use charges one-tenth or less of the hourly output.

All metal charges should be weighed. The coke charges should also be weighed, but it is sometimes difficult to provide the necessary labour and other facilities. The coke should then be measured in a skip provided for the purpose. If the skip is of a suitable size, that is, just holding the correct weight of coke for a charge, this method of measuring will be found almost as accurate as weighing. Careful control of the weight of metal and coke is well worth while as it not only invariably results in a saving of fuel, but is also often found to yield an increased output and metal of a more uniform temperature throughout the heat. Charging coke by shovels or forksful, on the other hand, is unsatisfactory and usually leads either to a waste of fuel or to fluctuating metal temperature. The flux, usually limestone, should be approximately 25 per cent. by weight of the coke charge and may be weighed or measured in the same way as the coke.

COKE RATIOS

If all metal charges are weighed and the coke either weighed or measured accurately by volume, it is possible to assess the ratio between metal melted and coke charged. This ratio depends on a great number of factors, including the design of the furnace, amount of preheating space available and so forth ; the type of metal to be melted, size of individual pieces, amount of steel scrap included in the charge, etc. ; the quality of the metal, e.g. whether high or low phosphorus, or whether a high carbon pick-up is required. Generally speaking, the smaller the size of the pieces of metal charged, the lower the coke consumption in melting them. Large pieces should therefore be broken as far as this is possible. The higher the percentage of steel melted, the more coke is required to melt and carburise it.

Very small pieces of steel, such as punchings, oxidise very easily in the cupola and necessitate the use of additional coke if they are to be melted successfully. Owing to the difficulty of accomplishing this, even under the best conditions, their use should be avoided. Very heavy steel scrap requires a high coke consumption to ensure that pieces are completely melted before they descend to the oxidising zone near the tuyeres. If a high carbon pick-up is required, the coke consumption will generally be high, as carbon must then be supplied to be taken up by the metal. Cupolas operated intermittently with frequent shut-downs will require more coke than those run continuously.

Owing to the large number of factors involved, it is not possible to give definite recommendations for the ratio of metal melted to coke charged which can be achieved on different classes of work. The following table is intended only as a guide and shows good average practice in the industry .

Class of work	Metal to charge coke ratio
High-phosphoric iron for fairly heavy castings ..	12 : 1–15 : 1
High-phosphoric iron for light castings	11 : 1–14 : 1
Medium-phosphorus iron for engineering castings	10 : 1–12 : 1
Low-phosphorus iron for high duty and automobile castings	8 : 1–10 : 1
Charges containing 50–75 per cent. steel scrap ..	7 : 1– 9 : 1
Charges containing more than 75 per cent. steel scrap	6 : 1– 8 : 1

In examining the overall coke consumption in a cupola, the coke used on the bed must be taken into consideration. The coke recovered from the debris at the end of the melt should be deducted from the total weight of bed coke used. As the same amount of bed coke is required whether the furnace melts 5 or 50 charges, it will be seen that the overall coke consumption depends to

some extent on the length of the heat, and coke consumption per ton of metal melted will be less on long than on short heats. In very small foundries, therefore, where it can be arranged, it is better from the fuel economy point of view to have a long heat every other day than a very short heat each day.

In conclusion, nothing is so conducive to fuel economy and overall economy in foundries as the maintenance of a supply of metallurgical coke of high and uniform quality. If the quality fails, the consumption is increased, with its consequence of increased cost of operation and transport ; the metal quality and temperature suffer as a result of the increased sulphur and ash content of inferior fuels. Foundry coke is not only a fuel, but a technical material, the quality of which affects the quality of the metal melted. Hence fuel economy in the narrow sense, may from a wider point of view actually militate against economy and quality of product. All fuel economy proposals involving melting coke must be judged on this basis.

SINTERING FURNACES

The cement kiln may be regarded as the most typical example of the application of heat for sintering. It concerns also an important industry. The following statement outlines the methods of operation and maintaining efficiency in rotary cement kilns.

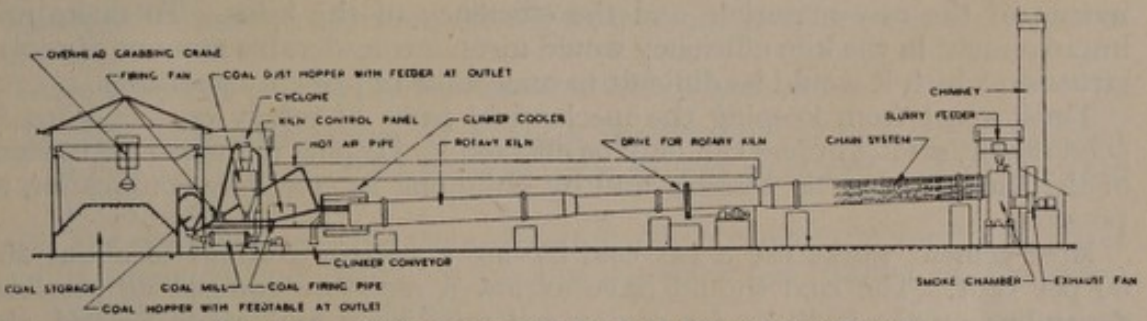

FIG. 148. Rotary cement kiln fired by pulverised coal.

ROTARY CEMENT KILNS

Rotary cement kilns as used in the cement industry, consist of cylindrical furnaces supported on rollers (Fig. 148). They vary in length from about 150 feet to something like 500 feet, and in diameter from 7 feet to 12 feet.

Nearly all cement works in this country are working on what is called the " wet process," which means that the raw materials are fed into one end of the kiln as a wet slurry and, owing to the rotation and inclination of the kiln, gradually gravitate towards the other end, where they emerge as clinker.

Pulverised coal is used in cement kilns as the fuel. Fuel oil or natural gas are also used sometimes in other countries.

The raw materials fed into the upper end of the kiln may contain between 36 and 43 per cent. of moisture. When entering the kiln, the slurry meets the hot combustion gases, the water is evaporated and later, when the material reaches a temperature of 1,650° F. (900° C.), the carbon dioxide is driven off ; finally, the material is heated to a temperature of about 2,650° F. (1,454° C.) in the hottest part of the kiln where the chemical reactions take place which cause the formation of the hydraulic product—cement clinker (Chapter XXIX.).

The clinker is generally cooled through a clinker cooler which either forms part of the kiln or is a separate unit. It is cooled by means of an air current and, therefore, a substantial part of the latent heat contained in the clinker is recuperated and utilised for preheating the secondary air required for burning the coal in the rotary kiln.

In order to produce the necessary chemical reaction and thereby give the required strength of the cement, it is essential to reach the above-mentioned temperature of about 2,650° F. (1,454° C.). This temperature is close to the fusion point of the refractories, and were it not that it is possible to build up a coating of cement clinker on the refractories, it would be difficult to get any lining material to withstand the heat and the abrasion.

Provided complete combustion is obtained with a minimum of excess air, and the moisture content of the slurry is kept as low as possible, fuel saving can only be looked for in the reduction of the temperature of the exit gases and of the cooled clinker.

To reduce the temperature of the exit gases, cement kilns are generally provided with a drying zone, which may have chains or other devices for the purpose of increasing the area opposed to the gases and external driers are also sometimes utilised. There is little which the operator can do to effect a saving of fuel in this part of the kiln other than to see that chains and other drying elements are kept in as efficient a state as possible. This also applies to the recovery of heat in the cooling of the clinker.

The coal consumption in rotary cement kilns operating on the wet process may vary from about 24 per cent. to about 32 per cent. when based on standard coal having a calorific value of 12,500 B.Th.U. per lb. The variation may be due to different factors, such as varying moisture content of the slurry, nature of the raw materials and the efficiency of the kilns. To make any improvement in the kiln efficiency would mean a considerable amount of reconstruction which it would be difficult to undertake in present conditions.

Thus, apart from keeping the mechanical parts of rotary cement kilns in good repair, saving in fuel can only be effected by keeping the moisture content of the slurry as low as possible and by obtaining as complete combustion as possible.

Most cement works use a gas coal having a volatile content of about 25–30 per cent. The coal should have as low a moisture content as possible, depending on the facilities for drying and grinding it and, when ground, the fineness should be about 15–20 per cent. residue on a 170-mesh sieve. Ash content should preferably be as low as possible, but it should always be of such a nature that it will assist in forming a protective coating on the refractory lining.

If coal having too high an ash content, or ash of an unsuitable nature, is used, it frequently leads to the formation of clinker rings, which not only tend to reduce the output of the rotary kilns, but also lead to inefficient combustion and incomplete burning of the clinker.

To ensure complete combustion, cement manufacturers generally reckon that it is advisable to have about 1 per cent. of oxygen in the exit gases. Frequent analysis of the exit gases should be made, and if an oxygen recorder is not already installed, it will be found a valuable instrument for the purpose of checking the combustion. It is very important to ascertain that samples of the exit gases are truly representative and, in this connection, it may be worth while noting that experience has shown that the most representative sample is extracted in the upper left-hand quarter (10.30 on the small hand of a watch) when looking at the kiln from the back and assuming it rotates in an anti-clockwise direction. Where two or more coals of widely varying qualities are used, it is essential for them to be well mixed so as to minimise variations in volatile contents, ash, and so forth ; otherwise unstable conditions are set up in the kiln, causing incomplete combustion. Provided really good mixing facilities are available, it is possible for cement works to use a proportion of anthracite having a volatile content as low as 6–8 per cent.

KILNS AND OVENS

Kilns and ovens are the structures used for the heating or firing of a number of important products, particularly those made by the pottery and heavy clay industries from clay or a mineral base. Differences in the type of kiln, together with great differences in the thermal requirements of the raw materials used to obtain the properties desired in the fired ware involve perhaps a greater number of problems specific to the process used than are associated with any other type of heating appliance. This is accentuated by the fact that the production of high yields of first quality ware is usually of major importance. Specifications for the testing of kilns used in the heavy clay industries have been prepared by the British Standards Institution (B.S.S. No. 1081, November, 1942, "Test Codes for Kilns for Heavy Clay Ware, Including Refractory Materials"). Similar specifications are being prepared for kilns used in the pottery industry. Reference to these standards will indicate those features which have to be watched in order to maintain or improve thermal efficiency. It is not proposed to deal here with the efficiency of such plant in a detailed way, but mainly to indicate certain outstanding factors affecting fuel economy.*

In the firing of all clay wares the need is recognised : for heating to a specific time-temperature schedule determined by the thermal requirements of the material ; for obtaining uniform heating and control of the process involved ; and for maintaining correct control over kiln atmosphere according to whether oxidising or reducing conditions are required to produce the desired changes in the ware.

Three stages of heating are almost invariably involved :—

(1) The water smoking period in which appreciable volumes of hot air must be passed through the setting in order to remove moisture until the ware is completely dry.
(2) The oxidation or preheating period in which chemically combined water is removed and oxidation of any carbonaceous matter in the green product is safely completed.
(3) The finishing period during which the required final temperature is attained and soaking time allowed at the finishing temperature for obtaining uniformity of heat treatment and developing the degree of vitrification or maturing properties required. Distinctive methods of pyrometry are used to determine when correct finishing conditions have been reached, e.g. seger cones or other heat recorders used to measure the heat work performed.

In each of these stages, the combustion conditions used are governed by special requirements for different classes of goods and the firing technique required usually differs markedly from that used in other types of heating plant. Firing methods also depend on whether intermittent or continuous kilns are used and on the applicability or availability of automatic firing mechanism. Intermittent kilns are mainly used to fire special products not amenable to continuous practice or to secure flexible treatment on a works manufacturing a variety of products where flexibility is of more importance than high thermal efficiency or large output of any one product. Unavoidable heat loss from the firing of these kilns is considerable. Continuous kilns are specially applicable to the firing of standard products where large throughput is desired. Recuperation of heat from cooling goods and from the kiln gases makes firing in this kind of kiln an operation of high potential thermal effi-

* The reader is referred to two publications by the British Pottery Research Association, viz. : (1) "Collected Talks Given at an Intensive Refresher Course in Fuel" ; (2) "Fuel Saving in the Refractories and Heavy Clay Industries : Part I.—The Firing of Intermittent Kilns ; Part II.—The Firing of Continuous Kilns of the Belgian Type."

ciency. Kilns of both classes are used in the heavy clay and in the pottery industries, but there are often differences in the types used in each industry. In general, coal firing is used on intermittent kilns, but graded or ungraded coal, pulverised coal, town gas, producer gas, oil, or electricity may be used for firing continuous kilns according to the quality and class of product made.

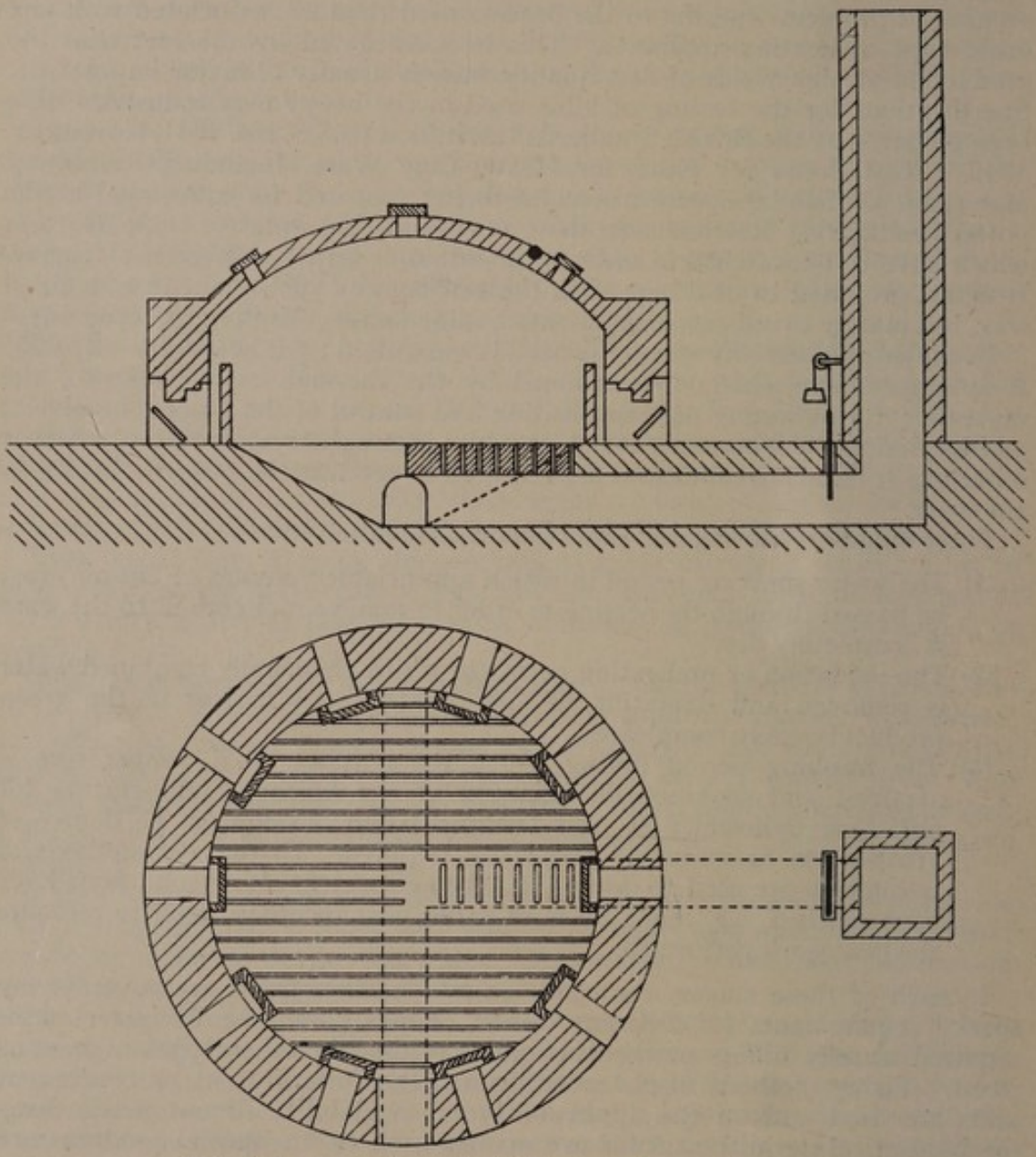

FIG. 149. Down-draught kiln for firing heavy clay ware.

In this country the use of mechanical stokers is largely confined to continuous kiln practice.

TYPES OF KILN AND OVEN

(*a*) *Intermittent Kilns.* Two main types of intermittent kiln are used in the heavy clay industry: the rectangular down-draught and the round down-draught. Both muffle and open-flame conditions are used with each type. In muffle firing the gases from the fires are not allowed to make contact with the goods being fired, heat transfer to the goods being obtained almost entirely

by radiation from the muffle walls. With open-flame firing, which is used to a much larger extent, all gases and flames from the fires pass through setting spaces among the ware before the combustion products are finally exhausted through the flue system. Apart from the shape and type differences generally recognised, intermittent kilns vary widely in size and in the arrangement or disposition of the sole flue system.

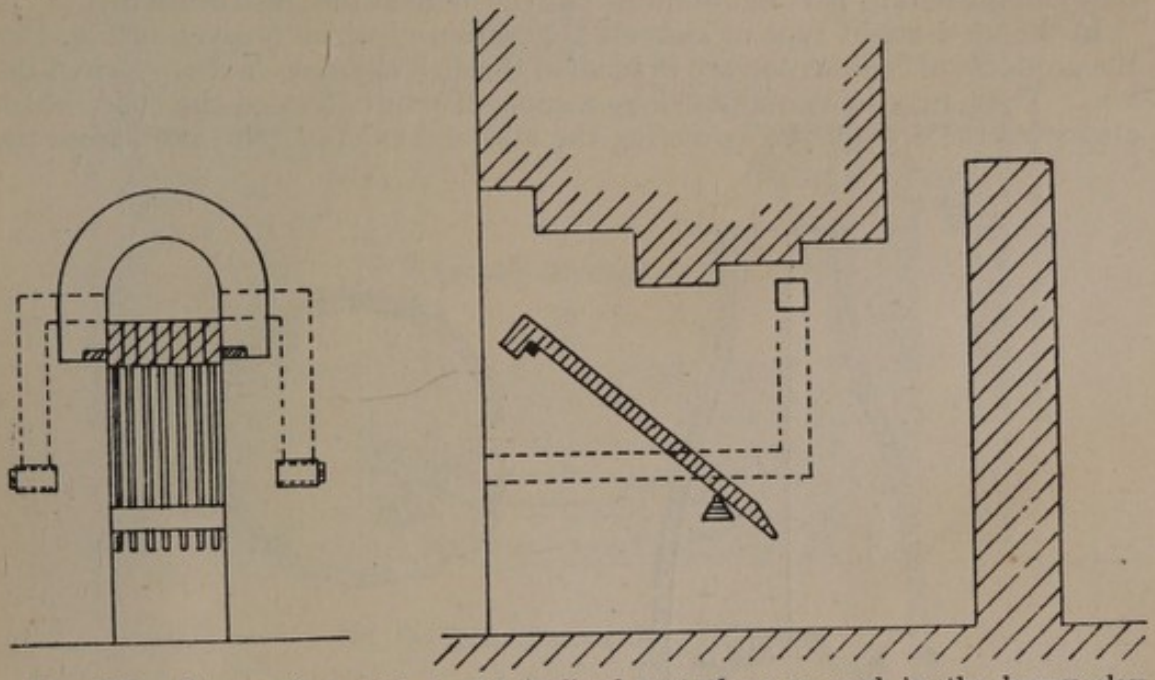

FIG. 150. Firemouth provided with inclined grate bars as used in the heavy clay industry.

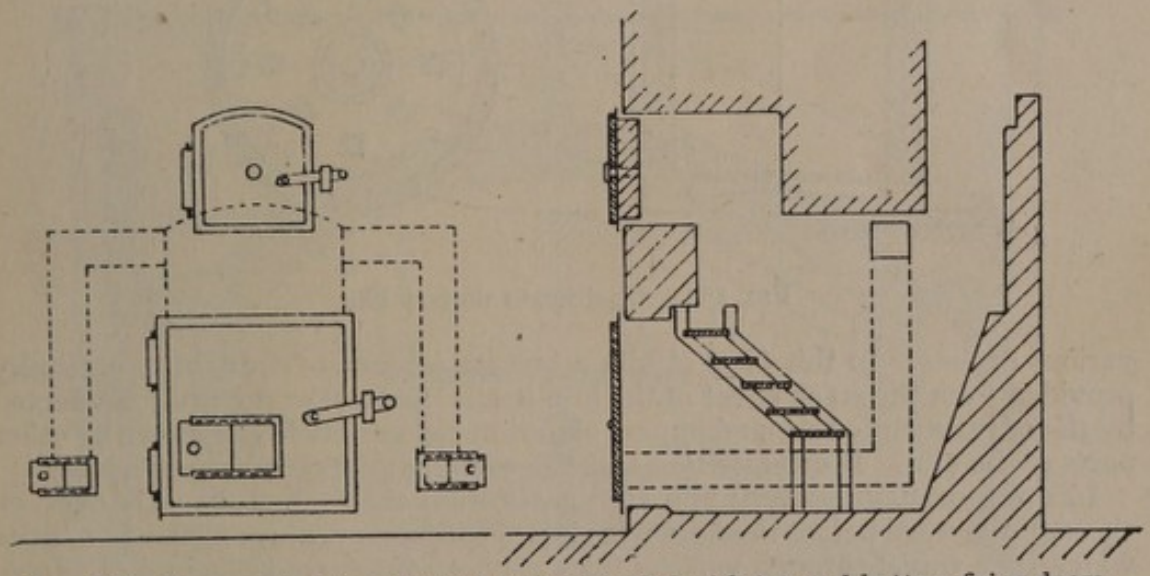

FIG. 151. Firemouth, stepped grate type, provided with top and bottom firing doors.

Each kiln is usually operated from a separate stack, generally external to the kiln, though a central internal stack is used sometimes. Draught on the kiln is controlled by means of a damper at the base of the stack. The number and disposition of the firemouths also varies. These may be of the solid bottom type or they may be equipped with horizontal, inclined or step grates. Combination grates are also used. Modern kilns are usually fitted with fire-doors permitting control of primary air, and slide-controlled secondary air flues

are being increasingly used. The designs of a typical round down-draught kiln and firemouths are shown in Figs. 149, 150 and 151.

In pottery practice two types of kiln or oven are distinguished—the up-draught and the down-draught. Each of these types may be fired on the muffle or open-flame system. Kilns are to be found in a wide variety of forms suited to the special needs of the product fired. The disposition of the firemouths plays an important part in obtaining correct temperature distribution.

In the up-draught type of kiln, an illustration of which is given in Fig. 152, the products of combustion are evacuated through openings in the crown of the kiln. These may be surmounted by a superstructure, termed the cone, which augments the draught for operating the kiln and exhausts the gases from the

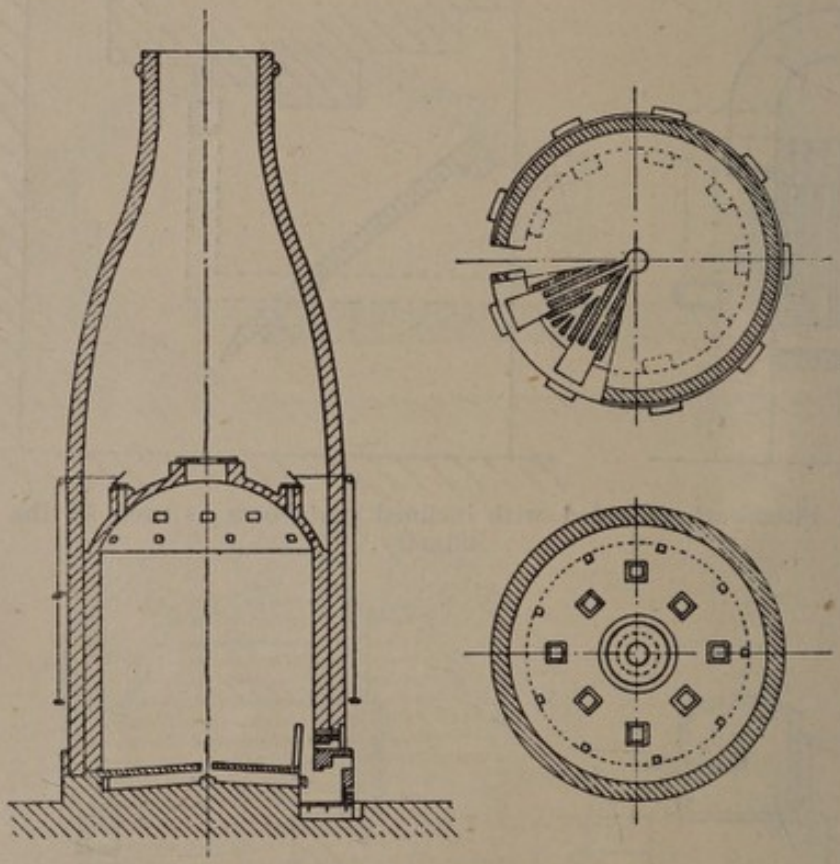

FIG. 152. Up-draught pottery kiln.

various outlets. In this type of kiln a certain amount of draught is naturally provided from the stack effect of the kiln itself. Control of the firing is effected by the adjustment of check dampers placed in the outlets in the crown or other parts of the kiln. The disposition and size of kiln outlets varies appreciably.

In down-draught pottery kilns the gases from the firemouths pass over or through the ware before they are withdrawn through openings in the floor from which they travel through vertical flues between the firemouths into the cone.

In the Wilkinson type of kiln part of the gases are exhausted through openings in the crown and part through openings in the floor. This arrangement involves a combination of up-draught and down-draught conditions. Muffle kilns used for pottery firing do not differ greatly in principle from those used in the heavy clay industry, but the cone system is generally used for exhausting the waste gases, and the kilns are generally of comparatively small size.

(*b*) *Continuous Kilns.* The economic advantages of mass production and high thermal efficiency are obtained from the use of continuous kilns (Fig. 153).

Because of the low fuel consumption and the refinement in control offered, advantages are sometimes obtained by firing with fuels other than coal.

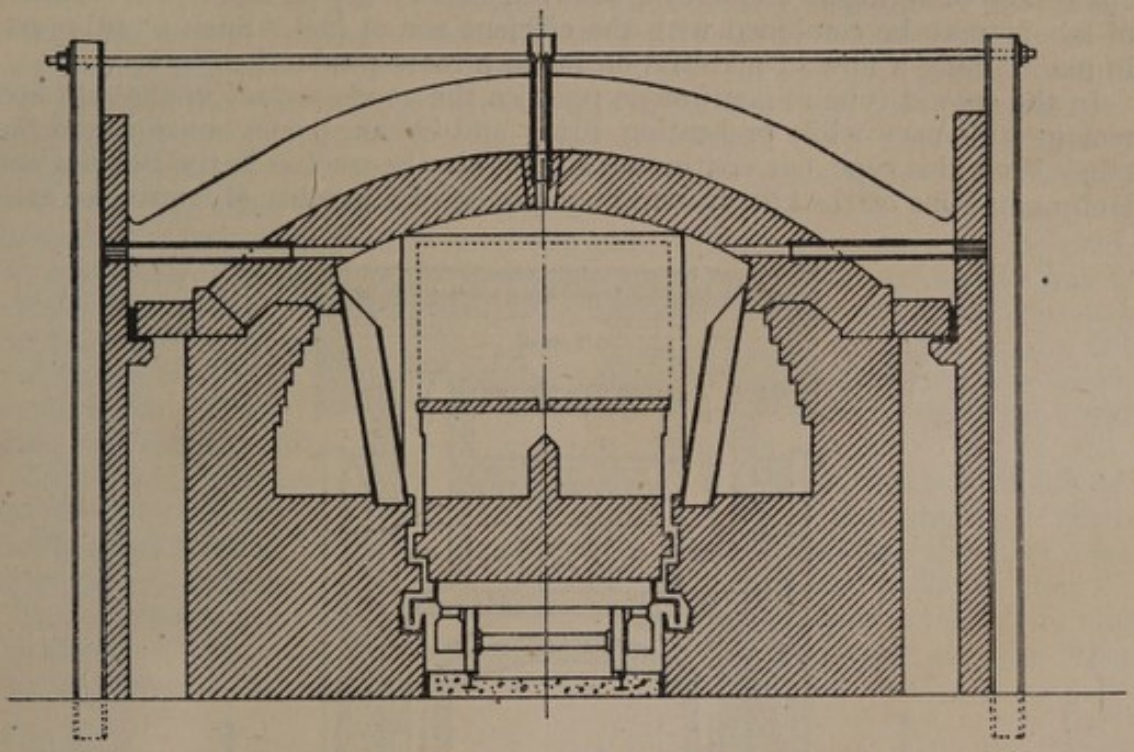

FIG. 153. Continuous kiln : (A) Thrift.

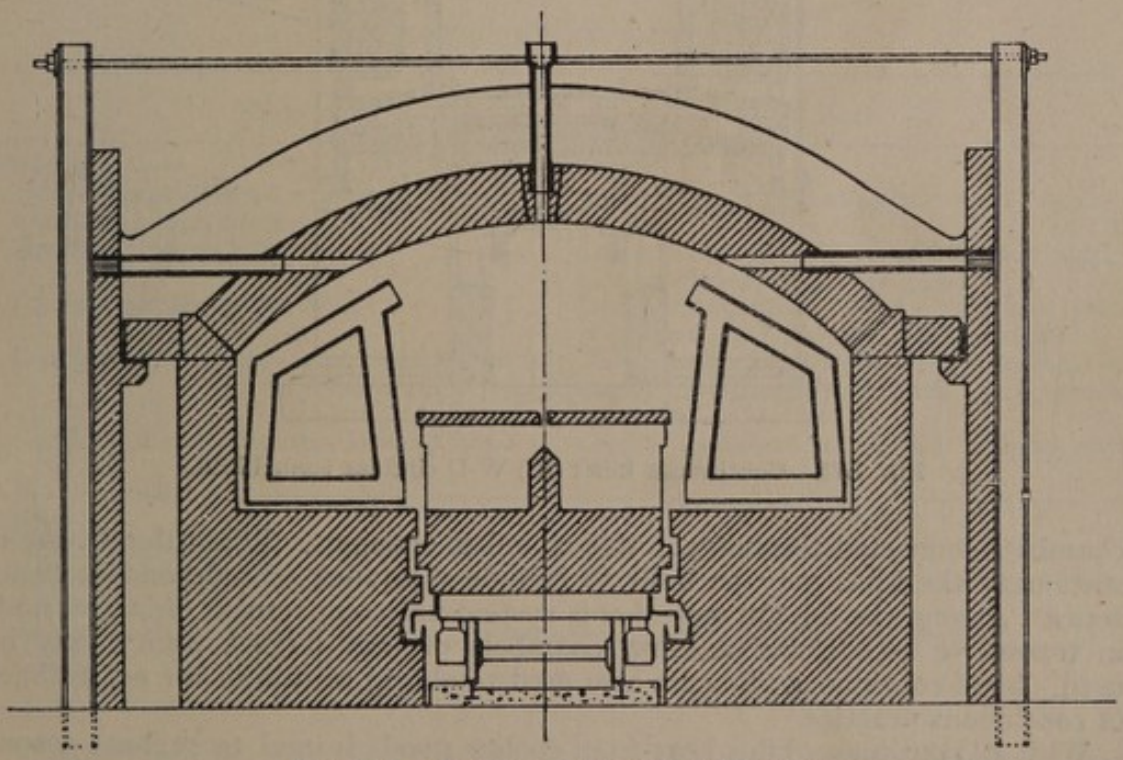

FIG. 153. Continuous kiln : (B) Dressler.

Two distinct firing principles are used in continuous practice. In car tunnel kilns the preheating, firing and cooling zones are fixed and the goods travel through these zones on cars or bogie carriages operated by an external pusher

mechanism. These tunnels may be either straight or annular, a moving hearth being used in the annular kiln instead of cars. The straight tunnel has the advantage of acting as a conveyor between factory departments and economy of labour may be combined with the efficient use of fuel. Such plant assists in maintaining a flow of material on modern mass production lines.

In the second type of continuous practice the goods are set in the kiln and remain stationary while preheating, firing and cooling zones move round the kiln. With this type, one continuous tunnel may be used as in the Belgian and Hoffmann kilns or the kiln system may consist of a number of transverse arch

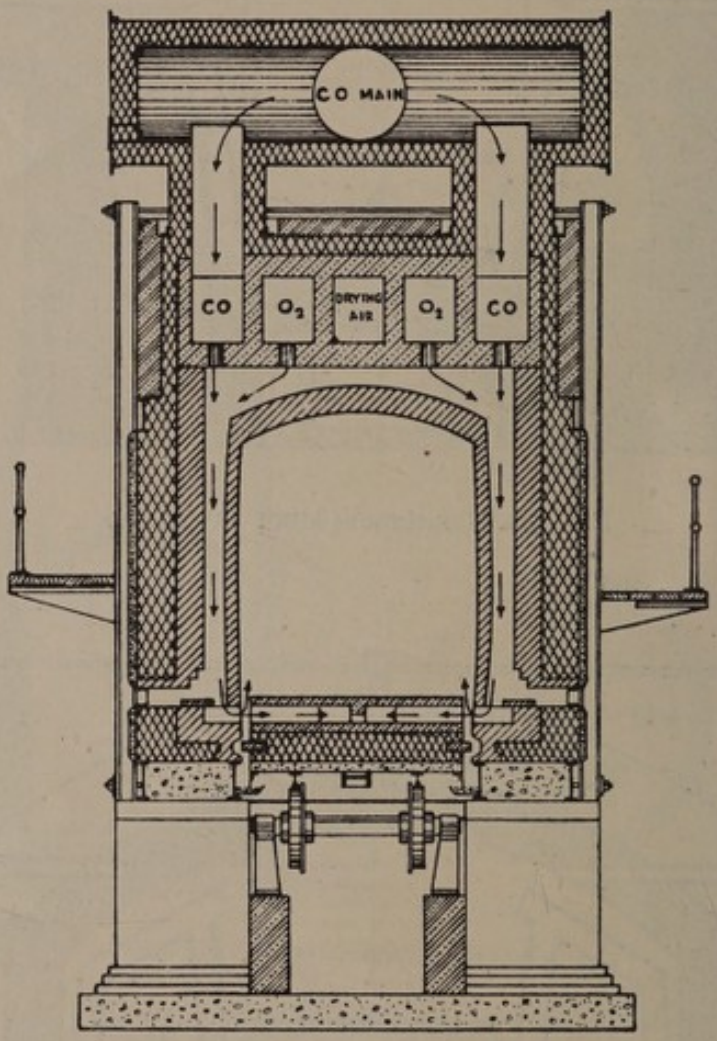

FIG. 153. Continuous kiln: (C) W-D circular tunnel kiln.

chambers connected through suitable chamber openings. When the setting is stationary the kilns may be built to include a number of independent firing zones. Among the advantages of this system are compactness of layout and, in transverse arch chamber kilns, chambers of very large capacity may be used. Semi-continuous kilns are also used to obtain certain of the advantages of continuous practice.

With all continuous kilns heat from cooling goods is used to preheat secondary air required for combustion and primary air may also be preheated in this way. Some of the heat recovered may be transferred to different parts of the kiln when more or less complicated systems of hot air circulation are used, or hot air may be transferred for extraneous use in dryers. The hot kiln gases are used to preheat the incoming or cooler ware so that the main heat losses

are confined to those due to radiation, convection and conduction from the outside walls.

In the heavy clay industry, straight car tunnel kilns, ring tunnel or annular kilns, such as the Hoffmann and Belgian, transverse arch chamber kilns, and semi-continuous kilns are used. Car tunnel kilns and Belgian kilns are mainly used in the refractories industry whereas Hoffmann and transverse arch chamber kilns are very widely used for firing building bricks. Open-flame methods of firing are generally used for heavy clay wares. Car tunnel kilns are invariably fired from the side. Mechanical stokers are usually used for coal firing, but these kilns may also be fired by producer gas. Belgian kilns are also side fired, the grate then extending across the width of each chamber and hand firing is generally adopted. The Hoffmann and transverse arch chamber kilns are top fired with or without the use of fenders or grates. The coal is usually fed to charging shafts left in the setting through a series of small holes in the crown about 4 feet apart. Certain transverse arch chamber kilns are side fired and firing by producer gas is also used for better quality ware.

Continuous kilns used in the pottery industry are mainly of the straight car tunnel type or annular type with moving hearth. Open-flame, muffle, or semi-muffle conditions of firing may be used. These kilns are generally fired with town gas or producer gas, but electrical heating is employed to some extent.

In addition to the above classification, kilns in the pottery industry are also classified according to the purpose for which they are used, e.g. for firing biscuit, glost or enamel ware.

FUELS

The type of fuel depends on the kiln used and the material fired. General requirements are shown in Table 92.

TABLE 92. KILNS AND FUELS

Kilns	Fuel
Heavy clay—	
Intermittent down-draught (open-flame, muffle and semi-muffle).	Coal : fired to side mouths.
Semi-continuous kilns	Coal : fired to side mouths or top fired ; fenders or grates may be used.
Continuous car tunnel kilns	Coal : mechanically fired ; producer gas ; pulverised coal and fuel oil also used.
Continuous transverse arch chamber kilns	Coal : top fired ; fenders or grates may be used. Producer gas used for special products.
Continuous ring tunnel or annular kilns..	Coal : Hoffmann top fired ; Belgian side fired on to grates.
Pottery—	
Intermittent, up-draught, down-draught, up- and down-draught, open-flame and muffle.	Coal : fired to side mouths.
Continuous car tunnel or annular moving hearth.	Town gas, producer gas, electricity.

Coal is the most usual fuel for kiln firing. For most purposes, size and size grading are important. Small coal is preferred for top firing and usually for use on mechanical stokers. For stoker firing, graded coal of small size is sometimes required. For intermittent kilns larger size or size grading is often specified, although " run of mine " coal is used if the proportion of fines is not too high. Correct size grading is of special importance during the finishing

period. Close grading is usually required for Belgian and fender fired kilns. Special grates may be used to suit the type of coal available.

Long flame coals with volatile matter content above 32 per cent. are required since radiation from long flames has considerable bearing on the uniformity of heating. For high thermal efficiency the ash content of the fuel should be as low as practicable. Low ash coals are specially desirable for firing to fenders and grates of the Belgian type where the ash may not be removed during the firing. Low ash content becomes more important with increased finishing temperature. Top fired kilns, such as Hoffmann and transverse arch chamber kilns, can usually deal with coals of higher ash content especially if the firing temperature does not exceed 1,000° C. Swelling or strongly coking coals are not generally desirable because they restrict the flow of primary air through the fuel bed. Where small coal is used for maintaining a thick fuel bed, however, some use may be made of coking properties if the bed is poked with sufficient frequency. Fusible ash is detrimental in general practice. Coals of low sulphur content are usually required for the firing of most pottery ware.

FIRING :—

(I) INTERMITTENT KILNS AND OVENS

The principles of combustion as outlined in earlier chapters apply in this type of plant, but must be considered also from special angles determined by the process involved.

Apart from the needs of the ware the main combustion requirements are governed by considerations of heat transfer. Until combustion can be maintained in the setting or in muffle flues, heat transfer from the fire gases takes place mainly by convection. Combustion of the coal should be completed in the firemouths with an excess of air. This applies to stages of the firing below about 700° C., since flame combustion cannot be completed in the kiln.

At higher kiln temperatures heat transfer is mainly by radiation, heat transfer from long flames having an important bearing on the uniformity of temperature distribution. During the higher temperature stages of the firing therefore it is desirable that most of the volatile matter from the coal should be burned among the ware so long as this can be done without detriment to the operation. In muffle kilns this combustion takes place in the kiln flues. The use of a deep fuel bed to provide carbon monoxide is also advantageous during this stage.

For each stage of the firing air control is very important and doors, covers or controllable secondary air flues are provided for this purpose. In exposed positions the provision of glut doors is an advantage to reduce wind effects and to allow the admission of primary air to be controlled.

Combustion conditions that require to be watched in order to attain economical use of fuel are as follows :—

(*a*) *Water Smoking Period.* Unless the goods have been pre-dried before setting in the kiln, very slow rates of combustion are required during this period because the maximum safe rate of drying may be very low. Slow burning is obtained by maintaining a shallow fuel bed, which remains at a comparatively low temperature because the kiln draught is low and only a small proportion of the air passes through the fuel. The drying is carried out by heated air and some 10–15 times the air necessary for combustion may have to be drawn through the kiln in the earliest stages. This may be decreased as drying proceeds till only about three times the volume of air needed for combustion is admitted when the temperature of the goods reaches 100°–150° C. The most satisfactory conditions are obtained when the difference between the temperature of the waste gases and their dew-point is about 6° C.

Artificially induced draught may be employed in the early stages to provide the required excess of air. The potter avoids conditions which might lead to condensation of moisture in the cooler parts of the setting whatever the type of oven, and this is controlled by the way in which the fires are gradually built up.

(*b*) *Rising Temperature Period.* In this period chemical changes in the ware become important. Chemically combined water is removed and if carbonaceous material is present it must be burnt out. Iron compounds must be fully oxidised. A specific oxidising atmosphere is thus needed during this stage of the firing and accurate control of secondary air is required.

Rates of firing depend on the time required to complete the oxidation before a temperature of 850° C. is reached in the kiln. The most efficient method of firing is that which avoids fluctuations in the temperature-time curve and undue excess of air. This can be assisted by giving the firer or the burner a heating schedule. If a recording pyrometer is used, it is recommended that the ideal heating curve .be drawn on the chart as a guide. Fluctuations may be unavoidable with intermittent firing and are not so serious under these conditions provided that black smoke does not issue from the oven over long periods.

In the early stages fires should be built up gradually by small and frequent additions of fuel. Secondary air should be controlled to give approximately 8–12 per cent. CO_2 in the waste gases. The carbonaceous matter in the ware, which is an additional source of heat, requires air for its combustion. The minimum content of excess air, subject as stated before, to the limiting conditions of the process, gives in this as in other heating processes the maximum economy of fuel. The advantage of preheating the secondary air is all the more marked when an oxidising condition is required to be maintained in the kiln or oven.

At a suitable stage in the firing, when the temperature has reached a point at which combustion of the volatile matter can be ensured by the temperature of the ware, and where secondary air flues are not provided, another method of firing may be applied. This consists in feeding alternate fires, whereby hot secondary air from the intermediate fires can burn the volatiles from the freshly charged fires. It is most important to observe correct timing of the air controls in relation to the time of firing, and also to set the air controls correctly. The appearance of the smoke haze from the chimney stack is a useful indication of combustion conditions where more refined methods are not available.

Only a small proportion of pottery ovens work with an external stack, but observation of the absence of smoke at the outlet of the hovel or cone is possible if the ovens are not enclosed in a building.

(*c*) *High Temperature Period.* Whilst generally high rates of heating may be employed at this stage and may be consistent with economy, again the technical features of the process determine the practice applied, and a soaking period may be necessary to ensure uniformity of temperature and the completion of the necessary chemical and physical changes in the ware, e.g. vitrification. With heavy firing, if oxidising conditions are required, local reduction may result, and ware be spoilt as well as fuel wasted. This is due to the difficulty of supplying sufficient air to burn the large volume of volatile products initially evolved. Small and frequent firing provides better conditions for the control of air.

In the normal method of firing, the glowing fuel is pushed down into the firemouth and the fresh charge of coal placed at the front and top of the fuel bed. With certain types of firing a steadier rate of evolution of volatile products may be obtained by using the coking method of firing. With this method the fresh charge is placed on a sloping refractory table above the main fire. When

the next charge is due the partially coked fuel is pushed from the table on to the fuel bed proper and the green charge again fed on to the table.

In pottery practice the rate of heating the ware is slowest at the top temperature although the rate of burning the fuel may be highest. Soaking is found to be necessary for most ceramic products in the first fire (biscuit) and is usual in the second or glost fire, but in the decorating kiln there is little or no soaking (i.e. deliberate holding at a constant temperature). In biscuit firing of pottery, fuel additions in the final period are usually heavy and it is generally necessary to maintain oxidising conditions if ware is not to be spoiled and fuel wasted.

Considering the oven or kiln as an isolated unit, i.e. in the absence of some form of recuperator, a high percentage loss of heat in the waste gases is the price which must inevitably be paid for the attainment of high temperatures in the setting, and the price is higher the larger the excess or deficit of air. Thus if the waste gases at the point of exit from the kiln are at a temperature of 1,000° C., they will carry away 43 per cent. of the heat in the fuel (assumed to be coal with a calorific value of 14,000 B.Th.U. per lb.) even if combustion is perfect, i.e. no excess air. With 50 per cent. excess air they will carry away 60 per cent. of the heat in the fuel.

The important influence of secondary air control can be illustrated in the following manner. If the combustion temperature of the fuel is in the neighbourhood of 1,400° C. and the finishing temperatures are in the range 1,000°–1,300° C., since the useful transfer of heat only occurs as long as the temperature of the heating medium is above that of the goods, a stage must be reached when the hot gases are imparting only a small proportion of their heat to the charge and the chimney loss must be necessarily high. If excess air is increased at this stage a serious loss of heat from the contents of the kiln or oven must occur, since air picks up sensible heat from the already hot mass, which generally has a large surface exposed to the gases.

If on the other hand the excess air is reduced beyond the safe limit consistent with good temperature distribution, a reducing atmosphere causing unnecessary damage to the contents of the kiln may result. Spoilage of material and waste of fuel are thus largely governed by the manner of the air admission to the firing points and the general control of the kiln draught.

Summarising the requirements of fuel efficiency, as in other types of heating appliance, importance must be attached to correct selection and application of the fuel, combustion efficiency, draught control, the maintenance of correct temperature schedules and the uniformity of temperature distribution.

(2) CONTINUOUS KILNS

Since the drying and preheating is performed by hot air from the cooling zone and hot kiln gases from the firing zone in continuous kilns, main considerations of the firing concern the conditions required in the firing zone. Thus once the required drying and preheating conditions have been determined, these stages of the heating are mainly controlled by fixing the lengths of the zones or parts of the circuit concerned. The drying and thermal requirements of the ware must be met as in intermittent firing, but because more control is usually obtainable much faster firing schedules are usually permissible.

Although most of the heat is recovered from the kiln gases and excess air does not involve the high losses found in intermittent practice, heat losses from these sources are not negligible and warrant some attention. Waste gases should not pass to the stack at temperatures above 100°–150° C. unless high draught is required and fans are not available. When fan draught is used waste gas temperatures need not exceed 60°–70° C. Heat losses due to excess air involve losses proportional to the waste gas temperature and the excess air

present. The influence of excess air is most marked in the firing zone. Heat distribution in the whole circuit is influenced by the firing conditions used and by the rate of fire or car travel, and conditions in forward zones may have to be altered when these factors are varied. These alterations are mainly effected by adjusting the draught balance.

The firing proper must be considered in relation to the type of kiln employed. Car tunnel kilns in particular are amenable to continuous mechanical firing of solid fuel, gas or oil or electric heating may be used. Firing conditions are easily controlled. Again, since the tunnel is of small cross section, uniformity of temperature distribution is fairly quickly established. The main requirements are: correct draught balance; avoidance of undue excess air, particularly in the firing zone; and adjustment of the rate of car travel, of setting, and of flame or firing conditions, to provide uniformly fired products. Typical continuous kilns are shown in Fig. 153.

With top fired Hoffmann and transverse arch kilns most efficient firing conditions are obtained when small dry coal is fed continuously so that much of the combustion may be completed before the coal reaches the kiln floor. Thus, when mechanical stokers are not used, there should be very frequent additions of fuel at not more than 15-minute intervals. Air control is again important and draught requires special attention. Other factors which influence the firing concern the type of setting employed, the number of cooling chambers and the number of forward chambers in the circuit.

The special requirements of grate and fender fired kilns, such as the Belgian and transverse arch chamber types, mainly concern control over combustion rates obtainable. The draught in the firing zone must be adequate to draw the maximum required amounts of primary air through the fuel bed without involving undue excess of secondary air in the chambers. For this reason the fuel bed must have a low resistance to air flow. Graded coals are therefore used and since the grates cannot be cleaned during the firing the coals should have low ash content. Generally coals of $\frac{1}{4}$–1 inch size are suitable for fender firing and coals of 2–3 inches size for the firing of Belgian kilns. Good forward heat should be aimed at so that the minimum amount of coal is needed to reach the finishing temperature. Frequent firing at 15–20-minute intervals is again desirable, and a thin level fuel bed should be maintained. Long flame coals are specially desirable because the whole of the chamber must be uniformly heated by coal fed to each grate, the distance between grates being 11–15 feet.

Different methods of firing car tunnel and annular tunnel kilns are used in the pottery industry. Contamination due to the effect of atmosphere is overcome either by muffle heating or firing in saggars, by the choice of suitable fuel or by electrical heating. Counterflow types of continuous kilns are also employed. With open-flame heating, producer gas, town gas or oil firing provides conditions which are readily controlled, and once the air requirements have been properly adjusted to give satisfactory combustion conditions only minor adjustments are required. With these fuels multiple burners may be used to increase the uniformity of heat distribution. Although costs are high, electrical heating offers advantages, and in particular thermostatic control over finishing conditions is readily applied. For muffle firing the air admitted to the firing zone need not much exceed theoretical requirements.

SETTING

The arrangement of the ware in intermittent kilns, in chambers of continuous kilns, or on the bogies or moving hearth of car tunnel or annular kilns, is equivalent in its effect on fuel consumption to the loading of other types of heating appliances. In kilns and ovens the material heated has thermal properties of the same order as the brickwork used in the construction of the

heating chamber. This means that rates of heating are necessarily low and accordingly the character of the setting is critical in permitting a suitable circulation of hot gases, and at the same time giving an optimum rate of heat penetration in keeping with the technical requirements of the process worked.

The hot products of combustion are capable of a rate of heat release to the contents of the heating chamber which is very much higher than the capacity of the material heated to absorb it. The character of the setting is therefore an important factor in the determination of efficiency. Specification of the optimum setting density for all materials fired in the different kilns is not possible here because of the variable requirements. In general, however, ware which is not pre-dried before setting in the kiln, or ware containing much carbonaceous matter must be set more openly than other green products.

In pottery firing a special problem concerns the relative efficiency of placing in saggars or firing in muffles, where contamination with sulphur gases must be precluded. Much empirical work has been carried out on problems such as these, on methods of setting, and on setting density, but further research is required to determine optimum conditions. The question is important because fuel consumption, throughput, uniformity of treatment and of heat distribution, may be greatly influenced by the setting.

FACTORS AFFECTING FUEL EFFICIENCY

The main general factors affecting fuel efficiency concern :—

(*a*) Control of air to meet combustion requirements and the special requirements of the ware.
(*b*) The suitability of the fuel or adaptation of firing methods to suit the fuel which is available.
(*c*) The maintenance of the shortest safe firing schedules and obtaining maximum throughput from the plant. In this connection as much of the production as possible should be fired in continuous kilns when they are available.
(*d*) The use of optimum setting density for the goods fired.
(*e*) Control over the firing. This includes methods of baiting, temperature and draught control.
(*f*) The use of insulation and attention to structural features.

The dimensions of firemouths, grates, fire arches, gluts and other pertinent structural features which may influence the flow of hot gases all fall into the class of controllable factors, which at rebuilds can be adjusted to improve performance. They are all dependent upon the process operated and local conditions ruling. Again no hard and fast rules of a general type can be laid down beyond an elaboration in detail of the principles which it has been the aim of this work to outline, in particular in the chapters on combustion, heat transmission, the flow of gases, and refractory materials. For details of structural technique reference should be made to specialist volumes and technical journals.

In heat insulation in kilns and ovens noteworthy contributions have been made to the more efficient use of fuel by the use of heat insulating materials both as external layers and in the internal structure. Thus the use of hot-face insulating material in intermittent kilns as an inner lining and for crowns, together with insulation beneath the kiln floor, has resulted in a considerable reduction of loss of heat in the structure, in producing greater uniformity of heating, and in reduced duration of firing for those wares which may be fired without harm at increased rates. A limitation arises in the practicability of using hot-face insulating materials for the highest temperatures. Saving in

fuel amounting to 10 per cent. and upwards of the original fuel consumption have been claimed as a result of using such materials.

The improvement of the structural features of large kilns and ovens, either of the tunnel or other unique design is a problem of a special type and its detailed discussion is a matter for a more specialist work.

Attention is directed to the value of measurements of temperature, draught and gas analyses, and to records of all factors concerned with firing. Measurements and records frequently enable fuel savings to be made because they emphasise the importance of economy to the firemen and provide objectives to be attained.

FURNACES AND FURNACE CONTROL IN THE GLASS INDUSTRY

In the manufacture of glass various types of furnaces and heating appliances are employed, designed each for its particular purpose, but by far the greatest proportion of the fuel used is that for heating the glass melting furnaces. Of these there are two general types known, respectively, as tank furnaces and pot furnaces. A relatively small amount of heat is used for the annealing of the glass or glassware.

MELTING FURNACES

Tank Furnaces. Tank furnaces consist essentially of a tank or trough, the bottom and lower sides of which are constructed of some suitable refractory material in the form of large blocks. Rising from the walls of the tank proper are the upper walls and covering the tank is an arch. The whole of this superstructure is usually made in silica brick.

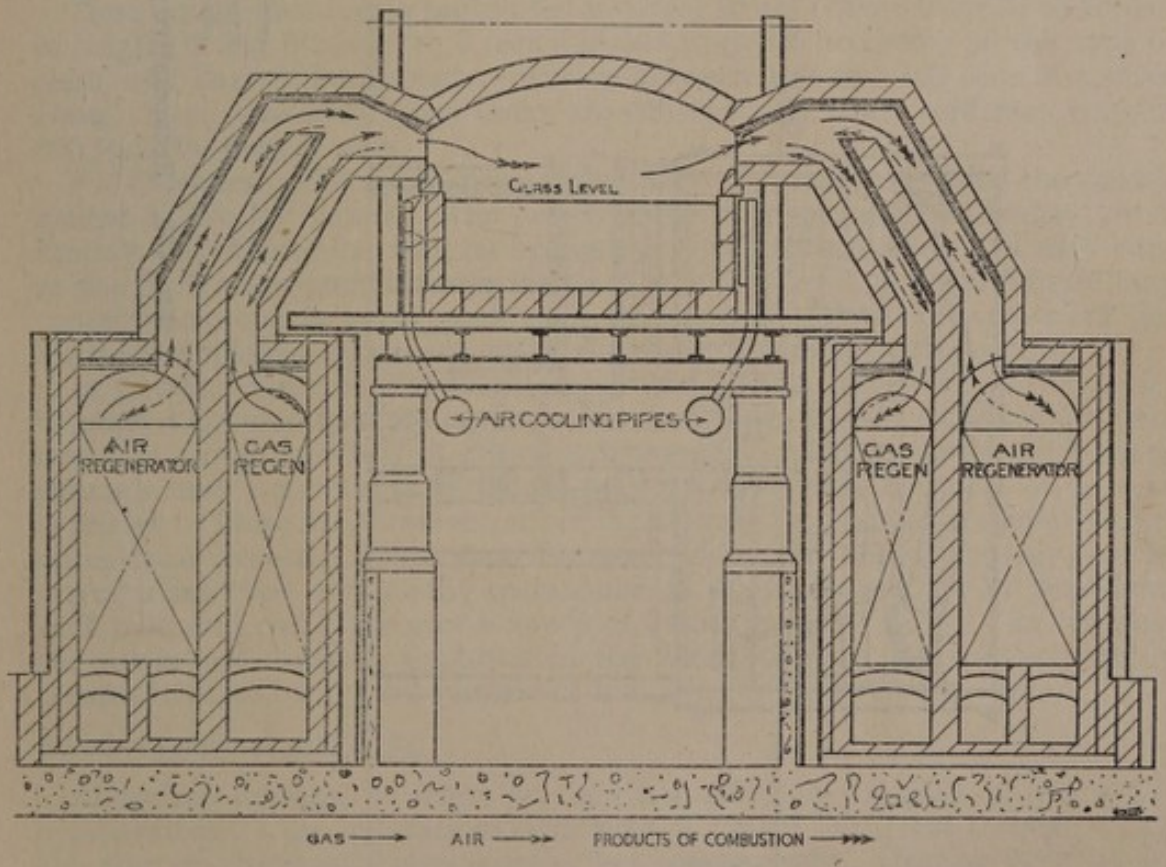

FIG. 154. Cross-fired regenerative glass melting tank, fired with producer gas.

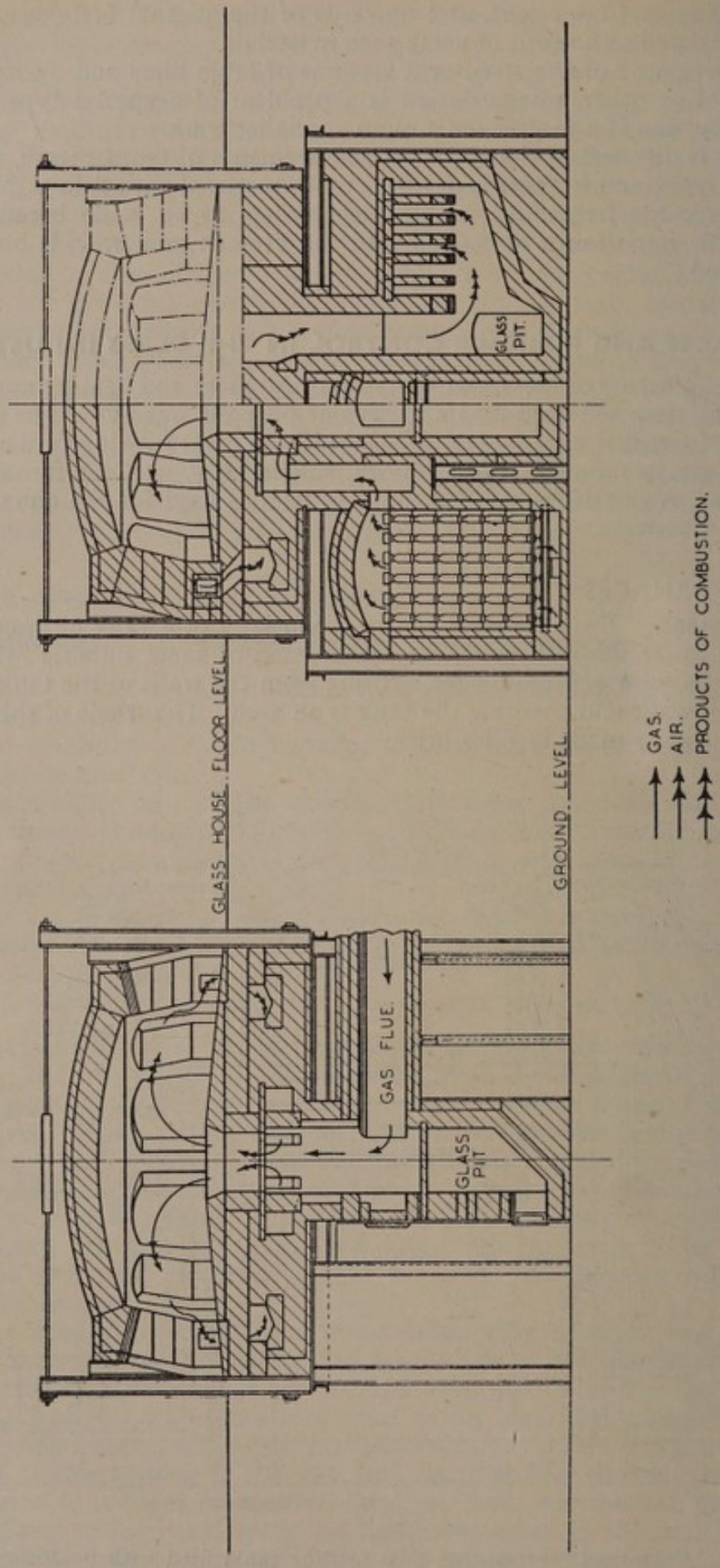

FIG. 155. Recuperative pot furnace for glass making.

A burner or burners are provided to supply flame for melting the glass, the flames being either projected down the length of the furnace (" end-firing ") or across the width (" cross-flame firing ") and flues are provided for the waste gases. These burners, including the means of supply of the requisite air, vary in design according to types of fuel and other circumstances. In all but very small furnaces, as much as possible of the heat in the waste furnace gases is recovered in recuperator or regenerator chambers, such recovered heat being employed to preheat the air needed for combustion ; and, in furnaces fired by producer gas, two regenerator chambers are usually employed, one for preheating the air, the other for the gas (cf. pp. 153 and 385).

Tank furnaces include day tanks and continuous tanks. Day tanks may be quite small, in which case recuperators or regenerators may be omitted, or may hold up to, or even more than 20 tons of glass. They usually contain no bridge and their operation resembles that of pot furnaces in that periods of melting and of working alternate.

The modern continuous tank furnace is almost invariably of the regenerative or recuperative type, recuperators being employed more generally for smaller furnaces. A typical cross-flame regenerative bottle glass tank is shown in Fig. 154. The tank is divided into two compartments by a double-walled bridge, the larger of the two compartments being termed the melting chamber and the smaller the working chamber. The depth of the glass is maintained constant by charging in raw materials at the melting end equivalent to the rate at which glass is withdrawn from the working end.

In tank furnaces for the continuous production of ordinary drawn sheet and rolled plate glass there may be three or even four compartments divided in most furnaces by floating bridges, and cross-flame firing is generally employed.

Auxiliary heating is applied to the discharge or working end of most tanks for controlling the glass temperature at the machines.

The average glass bottle tanks used in Great Britain range from 30 to 60 feet in length, 2 feet 6 inches to 3 feet 6 inches in depth according to the type of glass, and have a dead weight capacity between 100 and 400 tons of molten glass. Sheet and plate glass tanks are substantially larger and may contain 800 to 1,000 tons of glass.

Pot Furnaces. This is the most ancient type of furnace, in which the glass is melted in fireclay vessels. The pots themselves are usually of fireclay, or of fireclay and sillimanite, may be open or provided with a hood, and may vary in size from a few pounds up to 2 tons capacity. A so-called pot furnace may contain from one to twenty-four such fireclay pots. There are still a few pot furnaces employed which are either direct-fired or only semi-gas-fired.

In circular furnaces the flame, from whatever source it is derived, rises through a circular aperture or " eye " in the hearth or floor on which the pots stand in a ring, and is then drawn downwards from the dome of the furnace through flues constructed in pillars between the successive pairs of pots or crucibles. There are, however, other types of firing, and other shapes of furnaces than circular. Plate glass furnaces, mainly, if not exclusively, of the regenerative type, are usually rectangular in shape, have a row of pots down each side, and end ports give a sweep of flame down the length. In England the newer pot furnaces employed in the domestic glass industry are mainly recuperative and a typical furnace of that type is shown in Fig. 155.

ANNEALING FURNACES

Annealing furnaces or lehrs, as they are called in the glass industry, are comparatively long and narrow structures of tunnel formation through which the glass or glassware passes more or less continuously. Heating facilities are provided in all lehrs and are applied either directly or indirectly. The source

of heat may be any fuel—gaseous, liquid, solid or electricity. Many lehrs are built of brick with metal conveyors of various designs. The most modern bottle and press ware lehrs are constructed entirely or mostly of metal, are well insulated and employ the mattress type of conveyor. Some lehrs require no fuel or energy for a wide range of operating conditions, relying entirely on the heat in the glassware as it comes still very hot from the machine.

GENERAL METHODS OF PROCEDURE IN GLASS MELTING :—

TANK FURNACES

(1) *Charging the Furnace.* The mixed raw materials, comprising sand, limestone and soda ash and sometimes including a proportion of scrap glass or " cullet," known as " batch," may be charged into the furnace by :—

(*a*) Hand shovelling.
(*b*) Semi-automatic means.
(*c*) Fully automatic means.

In (*a*) and (*b*) the batch is introduced at frequent intervals in amounts consistent with the rate of withdrawal of glass from the furnace. In (*c*) the feed is similarly arranged, but is steady and continuous. The difference between methods (*b*) and (*c*) is that the automatic mechanism enables charges to be made at regular intervals of time. Completely automatic methods enable the furnace to be kept closed, thereby guarding against loss of heat and variation in temperature, and thus making for greater fuel efficiency.

Glass batch is " dusty " and avoidance of choking of regenerators is a matter for special consideration in the design of glass tank furnaces. Slag pockets, etc., are introduced and the modern tendency leans to the use of " briquetted " batch in order to curtail batch carry-over.

(2) *The Melting and Refining Processes.* Glass melting in tanks may be divided into three stages. First, the batch is pushed into the furnace and melted into a frothy bubbly glass. Second, this glass is refined, which means holding it at a temperature sufficiently high to get rid of the gas bubbles (partly by solution and partly by bringing them to the surface), and also to bring about the interdiffusion of the molten materials to make the resulting glass clear and homogeneous. Finally, the glass flows to where it is gathered by the glass makers on the machines and in so doing it is cooled to a temperature suitable for the particular process in use.

The first two stages just described are carried out in the melting chamber of the furnace and, as refining takes the longer time and requires the highest temperature, the major part of the space must be reserved for it. This means that in a glass tank all batch piles, other unmelted batch and scum must be confined to the smallest possible space at the batch entry end of the furnace. From 60–80 per cent. of the glass surface in the melting end should be left free for refining and this part of the furnace should be the hottest.

The best test of whether or not a furnace is producing good glass is the appearance of the glass as it is drawn from the working end. Consequently, providing there are no local conditions in the working end, feeders, forehearths, etc., which may be causing seedy glass, and providing other obvious precautions are observed, the appearance of seedy glass is an indication that the temperature is too low or that glass is being withdrawn from the working end too rapidly. The correct temperature is the lowest which will give glass of the required quality and in an amount sufficient for a reasonable production schedule.

(3) *Conversion of the Glass into Glassware.* The glass is supplied from the working end to one of several operating units, which may comprise workmen or

may be machines. According to circumstances, the operating units may consist entirely of workmen who carry out the shaping of the glass, or a combination of workmen and machines, or machines only. The machines in turn may all be of similar types and capacity or may vary. Although in practice such a distribution may be impracticable, it is an ideal to aim at to discharge the glass at rates as nearly the same as possible from all points round the working end where operation is proceeding.

Continuous tank furnaces are in operation throughout the twenty-four hours of the day and, in most works, for seven days in the week ; and they remain in operation for periods of ten months to two years, or even longer ; or, in general, until major repairs must be carried out.

POT FURNACES

(1) *Preheating the Pots.* The pots, which may have a bottom thickness from 3–5 inches and a corresponding wall thickness, are carefully preheated in an auxiliary furnace, known as a pot arch, over a period varying from five to fourteen days depending on the size of pot, the precision of the control exercised and the temperature to which they are fired before transfer, at their maximum temperature, to the melting furnace. Some modern pot arches are constructed with recuperators and can be raised to 1,300° or 1,400° C. In the simpler types the maximum temperature employed is 1,000°–1,100° C.

(2) *Charging the Furnace.* Each individual pot has to be charged. This is usually done by hand, but in plate glass pots from a charging machine attached to a truck moving on rails parallel to the furnace length. For most pot furnace work waggons holding some 8–10 cwt. of batch and cullet are employed and the pots are charged from shovels.

In furnaces in which melting by night and working by day is the system followed, the pots should all be emptied by a fixed time in the afternoon and after being brought up to melting temperature again, all will be charged to the same extent. With furnaces in which melting and working are proceeding side by side during the twenty-four hours, the quantity of material charged will depend on the working schedule. For example, some pots may be worked on small articles with the result that they are only half emptied over the normal working period, and whilst from one viewpoint it is desirable to work as much as possible, the prolongation of the work from one pot may interfere with the successful working of pots in the immediate vicinity which are to be worked on the next shift. A pot will then be filled once only, as this quantity of charge will fill the pot and make it ready for work at a time when adjacent pots are founding. When pots are worked low, two and sometimes three charges of material are necessary before they are full again. The period which elapses between the separate charges depends upon the size of pot, size of charge, type of glass, etc. Usually eight to ten hours is necessary between the individual charges.

(3) *Melting and Refining Process.* The pots may be either of the open type or the closed (that is, covered or hooded) type. Open pots are employed for melting plate glass and sometimes for electric light bulbs. For most other purposes in Great Britain covered pots are employed. The glass in open pots is heated both through the walls of the pot and by flame which passes over its surface. With a covered pot, the heat is supplied through the walls and by radiation from the crown. Melting is slower in closed than in open pots.

There are three stages in the complete preparation of the glass for working. In the first stage the materials must be completely melted ; in the second, time and usually higher temperature are needed to refine the glass ; whilst in the third stage the already refined glass is slowly cooled to an appropriate temperature level for gathering. The aims are similar to those in tank furnace

operation, but the necessary temperature adjustments have to be brought to bear on the individual pots, whereas in a tank furnace the appropriate temperatures are more or less fixed in the successive zones through which the glass flows.

(4) *Pot Furnace Operation when Producing Miscellaneous Ware.* In pot furnaces, because of the small numbers of any particular articles to be manufactured, and the intermittent operation of individual pots, the arrangement of the programme of work can greatly influence the quantity of fuel used in the furnace. In this connection points to consider are :—

(*a*) Wherever possible the furnace should be in production on a twenty-four hour basis.
(*b*) Furnaces should work to the greatest possible capacity.
(*c*) Furnaces should not work lop-sided ; i.e. during the working period the men should be evenly spaced around the furnace.
(*d*) Jobs should be allocated to those capable of doing them ; all " chairs " cannot make the same type of articles equally well.
(*e*) All mould holes are not necessarily alike ; some have special fittings, are extra deep, etc. When jobs are put out for these locations, work in the immediate vicinity may not be possible. Pots adjacent to those at which special jobs are proceeding should be charged up and usefully employed in melting glass ready (and not stewing) for the next teams of men on the following shift.
(*f*) Lehr capacities must be noted in accordance with jobs out on any shift, since some work takes up a lot of room, other work little. For economic working all lehrs should operate at reasonable capacity throughout all shifts. Transfer to lehrs from circular furnaces can be awkward, and some types of work must be got into the lehr quickly. Other gear may prevent certain jobs being put out at a specific time.
(*g*) Batch waggons must be brought up into position so that pots can be filled. Hold-ups can occur due to difficulties of getting waggons into position.

The control of satisfactory firing conditions in pot furnaces will in general be the same as for tank furnaces, but points applying particularly to pot furnaces are :—

(*h*) The wicket walls in front of the pots should always be kept in good condition and gas or air leakage should be avoided.
(*i*) Rings and stoppers should be made of first quality refractories. Stoppers must be burnt well before use, and rings should also be heated up in an auxiliary arch.
(*j*) Warming up moulds prior to use saves wastage of glass.
(*k*) Where recuperators are employed the tubes must be cleaned regularly and glass pockets must be kept clear of glass and slag.

ANNEALING FURNACES

Glass Annealing. The object of annealing is to avoid setting up in the glass objectionable strains which might cause fracture. This is achieved by passing the glass on a conveyor through a long tunnel called a lehr. In the first part of the lehr the glass is maintained at such uniform high temperature as is necessary to relieve the temporary stresses due to the prior processing. The glass is then cooled off so slowly that no new stresses are set up. This is continued to a temperature below which, due to the rigidity of the glass, no permanent deformation or stresses can occur and then the glass is cooled down to atmospheric temperature as quickly as necessary or as quickly as it will stand without fracture due to the temporary stresses.

The simplest procedures can be arranged and the highest fuel efficiency achieved, when the glass undergoing annealing is of simple and uniform dimensions, like sheet and plate glass, or consists of vast numbers of individual articles, like bottles of the same size and shape ; and when the sheet, plates or bottles pass continuously into the lehrs while still very hot from the machines which have automatically formed them. In modern lehrs for such articles, the expenditure of fuel per ton of glass is very small. The heat in the glass is sufficient to maintain the temperature required in the annealing zone, and only a little external heat needs to be applied for control purposes.

When articles of various shapes, sizes and thicknesses are simultaneously placed in the same lehr, compromise annealing schedules must be considered. The difficulty is greatest, and fuel efficiency lowest, when the articles are made from glass melted in pots and conveyed, already well chilled, to the lehr by hand. A thorough investigation into procedure, with a survey of fuel consumption, should be undertaken from time to time at works where such compromise schedules are necessary. The following suggestions may be given as applicable in general :—

(1) The ware should be put into the lehr as quickly as possible.
(2) The ware should be packed as tightly and uniformly as possible, but not so tightly that the articles touch. Square bottles or flat bottles should have a minimum distance of a quarter of an inch. The speed of the conveyor should be carefully adjusted to suit the required packing.
(3) Draughts are the curse of all lehrs. Doors must fit tightly and be kept closed as much as possible. The underside of all lehrs should be examined to see that no air can get in.
(4) In practically all lehrs, well-annealed bottoms are generally an infallible indication that the lehr is hot enough. If the sides or tops of articles are not so well annealed, the temperature should be lowered ; if this does not improve matters, draughts should be suspected.
(5) If bottoms are not well annealed, draughts should first be looked for, then the lehr should be examined for places where a sudden temperature drop might occur. Only if these measures fail should the temperature be raised.
(6) Instruments of all sorts must at all times function and read correctly.
(7) The lehr designer's instructions should not be forgotten or discarded.

FUELS USED IN THE GLASS INDUSTRY

General Review. Three types of fuel are used for heating the furnaces, viz. :—

(1) Producer gas.
(2) Coke oven gas or town gas.
(3) Oil.

In common with all large furnace operation the object of efficient fuel utilisation is to ensure that a *constant* quantity of fuel of *definite quality* shall be delivered to the furnace per *unit of time*, and that such fuel shall be supplied with the requisite quantity of highly preheated air to ensure complete combustion in the melting chamber.

The methods adopted to attain this end will be governed by :—

(1) Type of fuel : size and grading.
(2) Port design : { angle of inclination, point at which fuel and air meet.
(3) Combustion space : { suspended crowns { Time of sojourn of glass in furnace and efficient transfer of heat to charge. spring of crown

(4) Regenerators: { height and width, type and height of packing, reservoir between packing and top of chamber.

(5) Insulation.
(6) Correct draught.
(7) Minimisation of air inleakage.
(8) Suitable control instruments and intelligent interpretation of results.

The main concern here is items (1), (6) and (8).

INDIVIDUAL FUELS AND CONDITIONS OF USE

(1) *Producer Gas.* The gas should have as high a CO content as possible consistent with absence of difficulties in the gasification process. It should be of consistent quality and quantity, and be delivered at as uniform a pressure to the furnace as possible.

Regular gas analyses should be made, an experienced operator being able to determine the quality of the gas by a CO_2 determination only. For good quality gas this value will be 5 per cent. or less. Full analyses should be made at regular intervals apart from the CO_2 snap tests. The pressure in the producer main should be known and maintained. Pressure recorders of the contra flow type should be utilised where possible.

Flues should be kept gastight, dust pockets regularly cleaned, and flues burnt out with a minimum delay in the working programme. Many firms have eliminated periodic burn-outs by installing large gas mains and employing steam pokers during the normal operation of the plant. Automatic steam operated dust and tar removers are now available and are finding useful application in some plants.

(2) *Coke Oven Gas or Town Gas.* This fuel presents ample opportunities for the utilisation of control instruments and full advantage should be taken of such opportunities. The set-up and interpretation of results is as follows :—

(*a*) Recording pressure gauge on the mains.
(*b*) Governors fitted to each furnace to reduce main gas pressure to that desired, and to eliminate pressure fluctuations on the mains. This gives a steady gas supply.
(*c*) Orifice plates in individual furnaces coupled to recorders and integrators. Gas flow is thus continuously recorded and daily or hourly consumption is available.
(*d*) Recording calorimeter—if calorific value of the gas fluctuates the gas flow can be increased or decreased to meet it.
(*e*) With recuperative furnaces air meters can readily be used, and the required air-gas ratio can be maintained. On regenerative furnaces a duct can be fitted above the air valve and a large orifice employed to give the air flow. This orifice can be coupled up to a diaphragm on the gas supply and instruments employed such that a predetermined ratio of air to gas can be consistently and automatically maintained.
(*f*) Temperature recorders should be of open scale potentiometric type discussed later.
(*g*) Draught gauges should be of the recording type.
(*h*) Waste gas analyses should be consistently taken to ascertain that combustion has been completed in the furnace.

(3) *Oil Fuels.* The use of oil fuels is discussed in Chapters II and XXVIII. It is especially important in using creosote/pitch to ensure that the flame does not come into direct contact with the furnace refractories.

For efficiency and economy, high pressure atomising burners should in general be installed. The golden rule is to utilise a minimum of air for atomisation in

order that the maximum quantity of secondary air can be preheated to its maximum temperature prior to introduction into the furnace.

In pot furnaces where a large combustion space occurs relative to the space occupied by the pots, low pressure burners, whilst less efficient from a fuel efficiency viewpoint, possess advantages over the high pressure system.

In cross-flame furnaces, where periodic reversal occurs, arrangements are made whereby the burners are automatically removed from one set of ports as the burners on the opposite set swing into position. In some systems the burners are introduced in the sides of the ports near the furnace proper, as in this manner any undue wear on the ports is prevented and the full heat of the flame is developed where it is desired, i.e. in the furnace and not in the ports.

With oil firing, the deposition of carbon can be rapid if the oil flame, in its early stages of combustion, impinges on a cooler surface. This deposition may occur on port sides with incorrectly located burners, or may occur inside the tank proper. Poor atomisation and/or wrong location of the burner, is usually the chief cause of the trouble. The flaring of the port mouth may be necessary sometimes to avoid this difficulty.

In the operation of some of the larger glass-making machines, an auxiliary furnace, known as a rotary pot furnace, is contiguous with the main furnace, glass flowing from the main furnace to the auxiliary. Precise temperature control is essential, and it is worthy of record that these auxiliary furnaces are consistently maintained at a temperature of $\pm$ 3° C. by controlling pyrometers which, through a relay, regulate the oil supply to the combustion zone.

(4) *Air Supply and Control.* It has been shown (E. J. Gooding and M. W. Thring, *J. Soc. Glass Tech.*, Trans., 1941, **25,** 21–85) by a systematic examination of the air flow to glass tank furnaces that air or gas entering a regenerator do not necessarily distribute themselves equally to all ports, and thus sight holes for the easy examination of the flames from each port are an advantage. Grading of port dimensions may follow on the next tank repair as a result of such observations.

Some works have employed the difference in cross section of air or gas flues and the regenerator as a rough orifice plate, and by coupling up an inclined manometer to read the difference in pressure between air or gas flues and the regenerator have obtained useful control of air and gas flow through the regenerators—such methods are not accurate but by systematic observation they do give the furnace operator added data on his furnace and help towards efficient operation.

The air supply has also been controlled in some plants by using dampers in each individual port, but this practice is not generally found satisfactory due to mechanical difficulties which may arise during usage. Control is, therefore, best effected by careful adjustment of the air valve. Air shortage can be overcome by the installation of a fan on the air valve, in which event very close control is required.

The percentage of excess air should be a minimum consistent with correct length and type of flame desired in the melting chamber.

THE CONTROL OF GLASS FURNACE OPERATIONS

Both chemical and physical methods are widely used, the former embracing analysis of fuels and of waste gases, the latter including temperature and pressure measurements.

A. CHEMICAL METHODS

(1) *Analysis of Fuels*

(*a*) *Coals* are examined for their suitability as producer fuels.

(*b*) *Liquid fuels* are tested for viscosity and sulphur content, and also the temperature and pressure measurements.

(2) *Analysis of Waste Gases*

Air Supply. Systematic waste gas analyses are the usual method of checking the adequacy of the air supply, it being remembered that CO_2 is liberated from the melting batch and due allowance must be made for this factor. The following formula may be employed to determine the amount of CO_2 from the batch :—

$$x = \frac{100}{100-y}\left[A + C + \frac{y}{21}\left(B - \frac{C}{2}\right) - y\right]$$

where x = percentage CO_2 in waste gases due to batch.
y = ,, CO_2 in products of perfect combustion (i.e. maximum percentage CO_2).
A = ,, CO_2 in waste gases as analysed.
B = ,, O_2 ,, ,, ,, ,, ,,
C = ,, CO ,, ,, ,, ,, ,,

If C = zero, then

$$x = \frac{100}{100-y}\left[A + \frac{By}{21} - y\right]$$

and the true percentage CO_2 content of the waste gases is

$$\frac{A - x}{100 - x} \cdot 100.$$

The point of sampling the waste gases is of vital importance as inleakage of air is, in many furnace structures, sufficient to lull an unskilled operator into a false sense of security regarding the air supply. The sampling should be done through silica tubes, an auxiliary flowing stream of waste gas being tapped into the sampling apparatus, and the sampling point should be as *close to the melting chamber as possible.* A port close to the bridge should be employed in preference to one near the melting batch piles. This minimises the CO_2 from the batch piles.

Traces of CO are not inconsistent with high furnace efficiency, particularly where the sampling position is near to the furnace and the excess air is low, i.e. of the order of 7–10 per cent.

B. PHYSICAL METHODS

(1) *Temperature Control.* (*a*) *Temperature Control of Furnaces.* Pyrometers of the radiation type or rare metal thermocouples, correctly sheathed, and suitably positioned are employed for recording the furnace temperatures in the working and melting chambers of the furnace.

Open scale recorders of the potentiometric type are widely used. These instruments are capable of giving reasonably precise information of the fluctuations which may occur in the furnace temperature, and if calibrated so that the whole scale is covered by the short temperature interval, 1,200°–1,500° C. or 1,000°–1,500° C., give a much more accurate and useful indication of conditions than instruments whose scale covers 0°–1,400° C. Small fluctuations are not readily seen with the more extended scale and furnace control suffers accordingly.

It is usual to use, in addition, optical pyrometers of the disappearing filament type as a means of obtaining an indication of the true furnace temperature inside the furnace. Daily readings are taken by sighting these instruments, whilst the gas is off during a reversal, on the bridge wall or on the glass surface. Such readings act as a check on the normal recorders.

Radiation pyrometers are frequently placed in the back wall of the furnace, and are sighted on to closed refractory tubes. They must be kept in focus and be kept clean. Thermocouples usually occupy a central position in the crown in the melting and working ends of the furnace, and are mounted in special refractory blocks. All thermocouples should have compensating leads. Cold junction temperature compensation is automatic with modern instruments.

Whatever type of pyrometer is used it is important that it should be the slave and not the master of the operator. No pyrometer can indicate whether the furnaces are working satisfactorily or not. Its main uses are :—

(1) To indicate to the furnace operator whether the temperature is steady. tending to rise or tending to fall.
(2) To serve as guide if the necessity should arise for raising or lowering the temperature, this being judged by the condition of the furnace and the glass being withdrawn from the working end.
(3) To provide information for all concerned with the operation of the furnace.

(*b*) *Temperature Control of Regenerators.* In some plants thermocouples are located in the regenerators and reversals made at intervals corresponding to definite temperature conditions in these structures.

Normally, however, reversals occur each half hour. A steady rise and fall in the furnace temperature, between certain limits, occurs during each half-hourly reversal with regenerative furnaces. This fluctuation is, of course, not observed with recuperative furnaces.

(*c*) *Temperature Control of Lehrs.* It is important to have a thermocouple and recorder for the point in the lehr of maximum temperature. An indicator giving temperatures at various points along the lehr is useful. The most important of such points are :—

(*a*) Just inside the front of the lehr.
(*b*) The point where slow cooling is expected to commence.
(*c*) The point where fast cooling is expected to commence.
(*d*) Near the outlet end of the lehr.

If difficulty arises with a lehr, undoubtedly the best method of examining it is by the use of a travelling thermocouple, made up with iron and Eureka wire and insulated with asbestos tubing or refractory beads. Generally a length of 30 feet is sufficient. The thermocouple can be hooked to the conveyor or weighted in order that it may travel through the lehr with the ware.

(2) *Pressure Control.* Dampers should be of a type which gives easy and precise adjustment. Where remote control is not employed draught gauges should be so located that they can easily be read during damper adjustment.

Draught gauges should preferably be of the recording type and where overground metal cased flues are employed a piezometer ring should be utilised in connecting to the gauges. The draught tube should be placed between the chimney damper and the air valve, it being necessary to ensure that the air valve does not cause a large or variable drop, which, apart from other disadvantages, would introduce uncertainties into the draught readings.

The draught reading should always be as low as is consistent with satisfactory operation of the furnace, i.e. the adjustment should be such that a slight positive pressure is maintained in the furnace, and any " sting-out " should be sharp and clean.*

The following are maxims for getting good glass with fuel economy :—

(1) Keep the batch from floating up the furnace.
(2) Keep the melting end of the furnace at a temperature needed to make good glass but not hotter.

* " Sting-out " is the term for a jet of flame or hot gases.

(3) Maintain a steady temperature with the help of a pyrometer, and make regular checks of the furnace temperature by the use of a disappearing filament pyrometer sighted on a definite spot in the furnace, during gas reversals.
(4) Use the correct amount of air and avoid smoke.
(5) Avoid excess air and too short a flame.
(6) Keep long even flames wiping along close to the glass surface.
(7) Keep the stack damper as low as possible without causing excessive sting-out.
(8) Avoid leaving unnecessary holes open around the furnace.
(9) Stop up immediately any cracks appearing in the superstructure of the tank or regenerators.
(10) Use efficient regenerator system of adequate size and correctly packed.
(11) Use insulation wherever the condition of refractories permits.
(12) Use cooling only to maintain key refractory positions for an economically long life, endeavouring to reconcile long life with fuel efficiency.

ASSESSMENT OF THE EFFICIENCY OF TANK FURNACES

It is usual for costing purposes to base all costs on a ton of packed ware, but in order to keep a careful watch on the fuel efficiency of furnaces it is necessary to consider the weight of fuel used and the *total* output of the furnaces.

Unfortunately, for many years it has been a custom to express the fuel performance of a furnace in terms of the weight of fuel used per ton of glass melted. Unless the rate of output is absolutely constant this practice is misleading for the reason that the greater part of the fuel (65–75 per cent.) is not used for melting glass, but merely for the purpose of keeping the furnace hot (cf. Chapter XVIII).

The simplest way of overcoming this difficulty and a method which will allow correct comparisons to be made between different periods on the same furnace is as follows :—

A standard rate of output (representing a normal output) should be chosen for each furnace. Let this be W_s tons of glass per day (or week). For any period that the output is different from this the fuel consumption at the actual output can be adjusted to that for the standard output by adding or subtracting the fuel required for the additional or reduced amount of glass.

Let G = fuel used per day (or week).
W = output per day (or week).
G_e = equivalent fuel for standard output W_s.
W_s = standard output.
k = fuel required per ton of glass to correct fuel consumptions to standard output.

Then $$G_e = G + k\,(W_s - W).$$

The " equivalent performance " is :—

$$\frac{G_e}{W_s} \text{ fuel units per ton of glass melted.}$$

Values of k are given in the following table :—

Fuel		Value of k	
Coal	12,000 B.Th.U./lb.	0·16	tons coal per ton of glass
	13,000 ,,	0·15	
	14,000 ,,	0·14	
Fuel oil	.. ..	0·09	,, oil ,, ,, ,,
Creosote/pitch	.. ..	0·10	,, C.P. ,, ,, ,,
Coke oven gas	.. ..	7,000	cubic feet coke oven gas per ton of glass.

An example of the use of this formula is the following.

The average output of a certain furnace is 60 tons glass per week and the fuel consumption is 45 tons coal per week. The coal has a C.V. of 13,000 B.Th.U. per lb. Sixty tons glass per week is chosen as the standard (W_s) so that the usual efficiency is :—

$$\frac{45}{60} = 0{\cdot}75 \text{ tons coal per ton glass.}$$

During a particular period the output of the furnace was 70 tons of glass per week and the fuel consumption 48 tons of coal per week. In order to compare efficiencies :—

$$G = 48\,;\ W = 70\,;\ W_s = 60\,;\ k = 0{\cdot}15$$
$$G_e = 48 + 0{\cdot}15\,(60 - 70)$$
$$= 48 - 1{\cdot}5 = 46{\cdot}5.$$

Then the standard performance is $\frac{46{\cdot}5}{60} = 0{\cdot}775$.

Thus the efficiency of the furnace is not as good as normal (0·75). There are two points to note :—

(1) In many furnaces the fuel performance deteriorates with age. That is, the older the furnace the more fuel it requires. If, however, either the equivalent fuel consumption G_e or the standard performance G_e/W_s is plotted during the life of the furnace the curve obtained can be used for comparison with future " runs."

(2) The simple method just described does not permit rational comparisons between different furnaces. In general, larger furnaces will give better performance on the above basis.

Recently the Furnace Committee of the Society of Glass Technology has completed work on the development of a method of assessing the performance of tank furnaces of all sizes and working under greatly varying conditions. A complete description of this work has recently been published (*J. Soc. Glass Tech.*, Trans., 1943, **27**).

REHEATING FURNACES

Reheating furnaces are used for the heating of material for fashioning, and are accordingly confined generally to the metallurgical industries. They are found in rolling mill, forge, press shop, and other workshops where heating prior to shaping is required. Reheating may be effected in various stages, in each of which the function of the furnace may be different, and special types of furnaces for the specific stage of the heating may then be used ; e.g. pre-heating furnaces prior to transfer to another furnace for the main stage of heating, wash heating for quick reheating of hot blooms already partially fashioned, sheet and pair furnaces in connection with the rolling of sheets, and soaking pits for finishing the heating of already hot ingots for rolling.

They may be operated intermittently or as batch furnaces in which the heating of the material is carried out in successive charges, or on the continuous principle in which the charge is moved through the furnace by means of a suitable mechanism, such as a pusher, conveyor, bogie carriage, car bottom or rotating hearth.

All types of fuel are used, and the methods of their application vary widely according to requirements determined by the type of material involved and the local conditions prevailing as to availability of fuel and shop layout. Most of the factors relating to the problems of combustion and heat transmission have already been mentioned, and it remains to discuss special points relating to the various fuels.

COAL FIRING

(i) *Hand fired Furnaces.* Both natural and forced draught are used. The coal burns as it would on a boiler grate, with the difference that larger sizes of fuel and thicker fuel beds are customarily used. Rates of combustion on the grate fall within the range of 10–20 lb. per square foot of grate area per hour. Long flame, free-burning coals are preferred. All the air may be supplied as primary air through the grate. Secondary air may also be used with advantage, but its use requires attention to distribution and control, since the volatile content of the flame varies as each charge of green fuel burns off. Local jets of excess air may cause spoilt material.

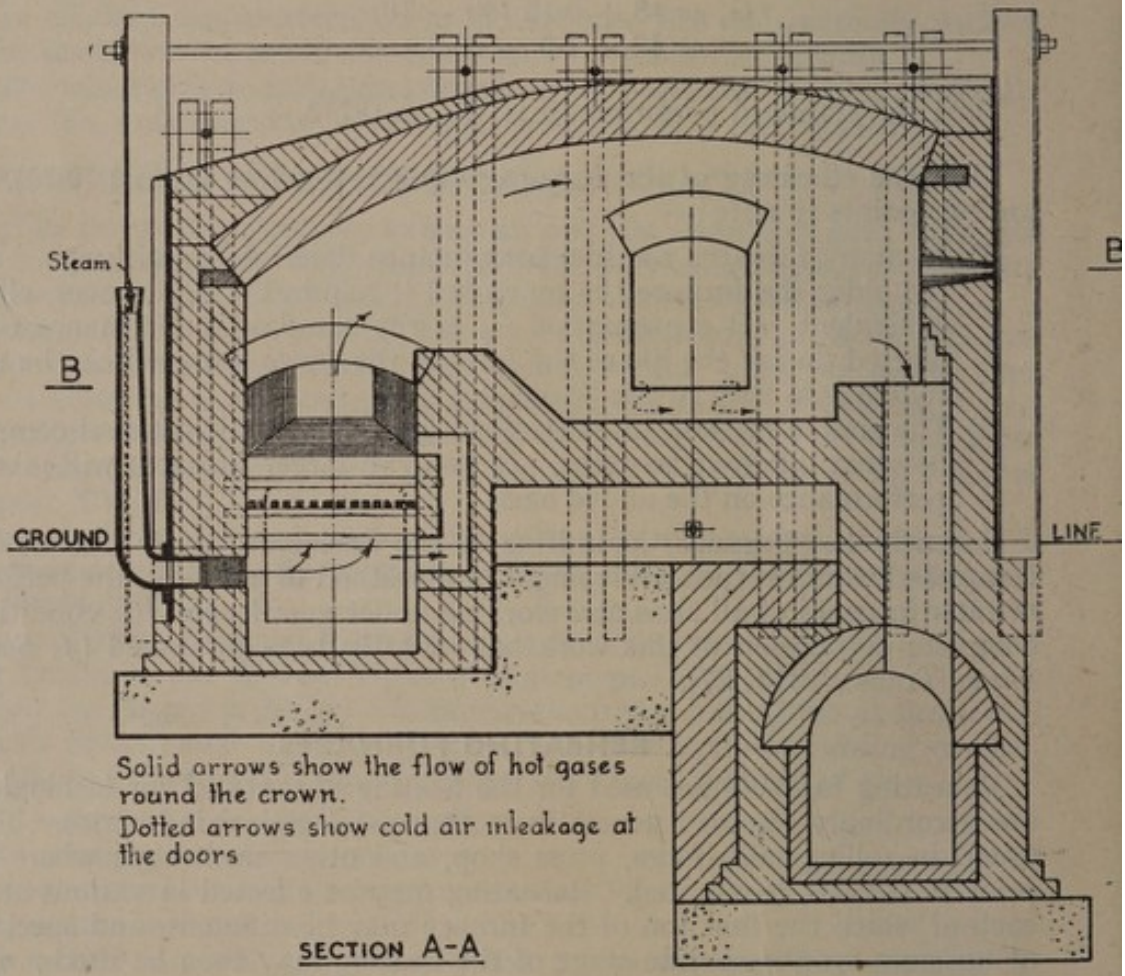

FIG. 156. Steam-blown coal-fired batch furnace.

With natural draught the main control of the furnace operation depends upon damper adjustment. Dampers should always be kept in good repair and be capable of fine adjustment since the difference between satisfactory and faulty operation may depend upon the possibility of working to a fraction of $\frac{1}{10}$ inch w.g. of furnace draught. The level of the grate in relation to that of the hearth and the bridge wall is important in assisting uniform heating. The lower the grate the higher the gas pressure at the bridge wall, and the easier it is to obviate air leakage at the hearth level.

The control of heating in the furnace is more effective when the fire-grate is enclosed and positive draught applied.

Fans are preferable to steam jet injection, which is illustrated in Fig. 156, since the steam consumption may be as high as 1 lb. of steam per lb of coal, and worn or wrongly centred jets are particularly wasteful. Firebar design

repays investigation. A device of using cast angle bars resting in grooves on a bearer sloped and admitting of the use of a trickle of water along the bars has been used with success in preventing clinkering. In large coal-fired furnaces, in order to obtain uniform temperature and adequate fuel bed control, it is often necessary to use several separated grates rather than one large one.

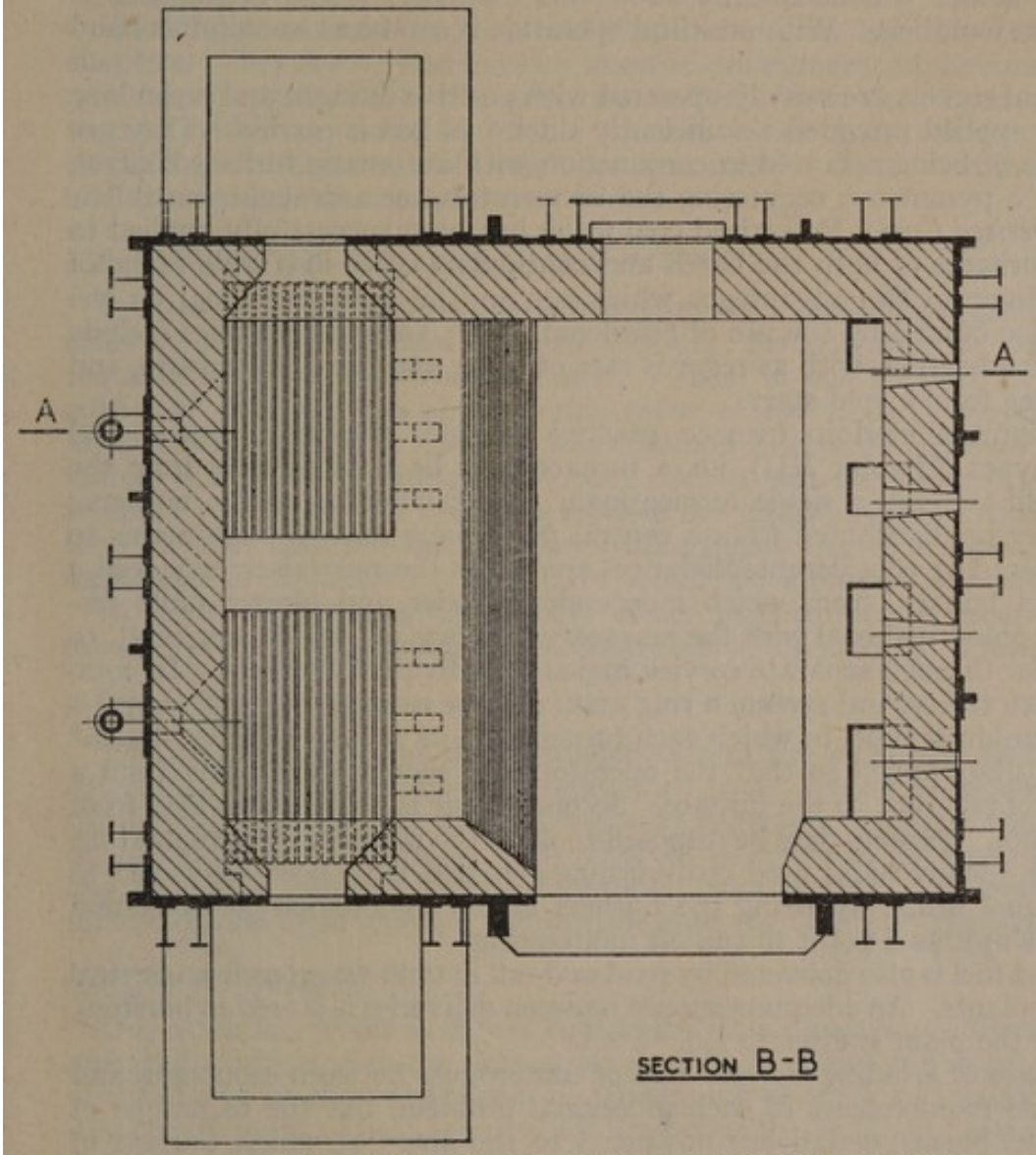

FIG. 156 (*cont.*). Steam-blown coal-fired batch furnace.

(ii) *Built-on Producers.* Self-contained producers built on to the furnace chamber vary in design from what is nothing more than a sloping grate fired through a door placed above the top of the grate to a static producer with the normal type of tuyere using steam-air blast and a charging hopper. Secondary air is necessary, and it may be preheated. Many special types of grate have been applied to assist in breaking up clinker and cleaning the fires. The choice of fuel used and the technique of operation resembles in essential elements the practice for normal producer operation (Chapter XVII).

(iii) *Mechanical Stokers.* Underfeed and sprinkler stokers are used for reheating furnace operation. It must be borne in mind that temperature conditions in the combustion chamber are necessarily high and the conditions may cause parts of underfeed stokers to be burnt out unless the proper precautions peculiar to the stoker are observed. For the same reason the back

plate and driving shaft of the sprinkler mechanism in the sprinkler type should be water-cooled.

It has been found possible to operate mechanical stokers in reheating furnace practice with almost ideal combustion conditions, and accordingly they can show an economy over hand firing. But it must not be presumed that a mechanical stoker will necessarily show this economy, unless intelligence is applied in its handling. With unskilful operation it can be as wasteful as hand firing.

Mechanical stokers are usually operated with positive draught and secondary air may be applied provided a sufficiently thick fuel bed is carried. They are also capable of being operated in conjunction with automatic furnace control, either from a pyrometer, registering the temperature, or a draught controller.

(iv) *Pulverised Coal.* Pulverised coal firing has been successfully applied to reheating furnaces of both the batch and continuous types in a wide range of practice. Some of the advantages which gas and oil have over coal on the grate are also offered by the use of powdered coal. These advantages include control of combustion, both as regards rate of firing and the fuel-air ratio, and quick heating from a cold start.

The equipment used in furnace practice includes the unit, central and combined types (Chapter XII), i.e. a furnace may be fired directly from the grinding mill through a single service main and distributors to the burners; the fuel may be distributed from a central pulverising unit through mains to furnaces placed at considerable distances apart; or the pulveriser may feed a sectionalised bunker, from which independent feeder and blower units distribute the pulverised coal with the necessary primary air, 30–40 per cent., to act as carrier through separate service mains to individual furnaces. In combination with the central system a ring main may be used.

In the combined type, in which each furnace has its separate feeder, distant control may be applied so that the operator may regulate the firing from a control panel adjacent to the furnace. Secondary air is usually provided from a separate fan. Burners may be disposed to meet the needs of the temperature distribution. Many hand fired coal-burning furnaces have been converted to pulverised fuel firing, by fixing the burners in the combustion chamber and using the ash-pit as a point to run off molten slag.

Pulverised fuel is also delivered by road and rail in tank waggons from central pulverising plants. An adequate supply between deliveries is stored in bunkers, from which the plant is directly fed.

The fineness of grinding is a question of compromise between economics and the technical requirements of each individual problem, but the technique of grinding and burner design has advanced to the stage at which the use of powdered fuel in small furnaces for the heating of billets for drop stamping is a practical and economic success.

In burner design it should be borne in mind that powdered coal will not burn in the open air, whereas well-atomised oil, with some difficulty, and gaseous fuels, readily, do so. Accordingly the flame should be surrounded by brickwork at its root. The design of the burner and combustion chamber is a specialist problem, but the simplest types of burner are usually the most successful.

The powdered coal flame is generally one of high calorific intensity, and therefore in general, with suitable safeguards against local overheating, applicable to reheating furnace practice.

The action of powdered coal on brickwork varies greatly with the composition of the ash and with the furnace temperature. Special attention must be devoted to selection of refractory materials and the design of flue offtakes and waste gas ducts. Ready accessibility of these for cleaning is essential.

In bunkers provision is necessary to safeguard against fires arising from

autogenous combustion. Carbon dioxide may be used as the smothering agent. Coal dust explosions are rare in modern pulverised coal plants.

Flame intensities and flame length can be varied with powdered coal in very much the same manner as with oil and gas.

GAS AND OIL FIRING

The characteristics of gaseous and liquid fuels have been discussed in other chapters. They are applied in every known form in reheating furnaces and their methods of application are too numerous to deal with in an adequate descriptive manner within the compass of this volume. Their main inherent advantage lies in the effectiveness with which the heat may be distributed in the furnace and the combustion conditions controlled.

AIR PREHEATING TO GIVE FLAME INTENSITY

Preheating of air and/or gas is necessary with certain low-grade fuels, for example producer gas and blast furnace gas, for attaining the temperatures required in reheating furnace practice. Thus, in one instance of the use of cold coke producer gas of a calorific value of 130 B.Th.U. per cubic foot a furnace temperature of 1,140° C. was obtained with an air preheat of approximately 300° C. at the recuperator, but since it was necessary that a furnace temperature of 1,200° C. should be reached this gas could not be used. It would have been necessary to use a burner with a higher degree of turbulence in the flame and an air preheat of 420° C. to attain the additional intensity required.

In reheating furnaces accordingly either regenerators or recuperators are used for preheating air, and/or gas where fuels of low calorific value are used.

FURNACE LOADING

The importance of the firing appliances and the structural features which determine the distribution of temperature have already been discussed. A third factor governing the fuel efficiency of a reheating furnace is the disposition of the charge in the furnace. This involves attention to the methods of handling, charging, discharging and, if required, transporting. The type of equipment depends upon whether the furnace is of the batch or continuous type.

BATCH FURNACES

Due attention to the design of equipment which may be readily and rapidly operated is reflected in the time saved in opening doors, and in proper placing of the charge on the hearth from the standpoint of quick and uniform heating. The hourly input of heat required by batch furnaces is largely governed by the character of the furnace chamber ; variations in the weight of material heated only affect the hourly rate to a limited degree. The point has been illustrated in Chapter XVIII, Table 90. The appliances which are used for handling are therefore of considerable importance. Lifting forks and peels, slung from runways or attached to a wheeled carriage, are used and should be capable of convenient and speedy operation. Porter bars are employed for heavier articles, such as ingots and blooms. Charging machines of every degree of fashion from simple carriage types to power-operated and highly mechanical appliances are also called into service. The heaviest charges are heated on bogie carriages, or car bottoms, which are provided with sand seals. Alternatively, the furnace roof may be moved and the charge handled by overhead cranes.

In car type furnaces, particular attention should be devoted to hearth insulation and air inleakages at sand seals, and particularly at the sides below the door. Cold bottoms can be due to air leakage at these points. The condition of bearings, adequate anchorage and robustness of rails or other type of bogie

track are of particular importance in reheating furnace practice where temperatures are likely to be high.

The batch furnace is favoured for forge, press and drop stamping work. The furnace dimensions, method of heating and general mode of operation are determined by the size and character of the material heated. For small blooms and billets rapid heating may be required in order to reduce scaling and, when heating steel, decarburisation of the surface.

A critical factor in the operation of such furnaces is the rate of heat input to the furnace. The most rapid heating is the most economical from the stand-

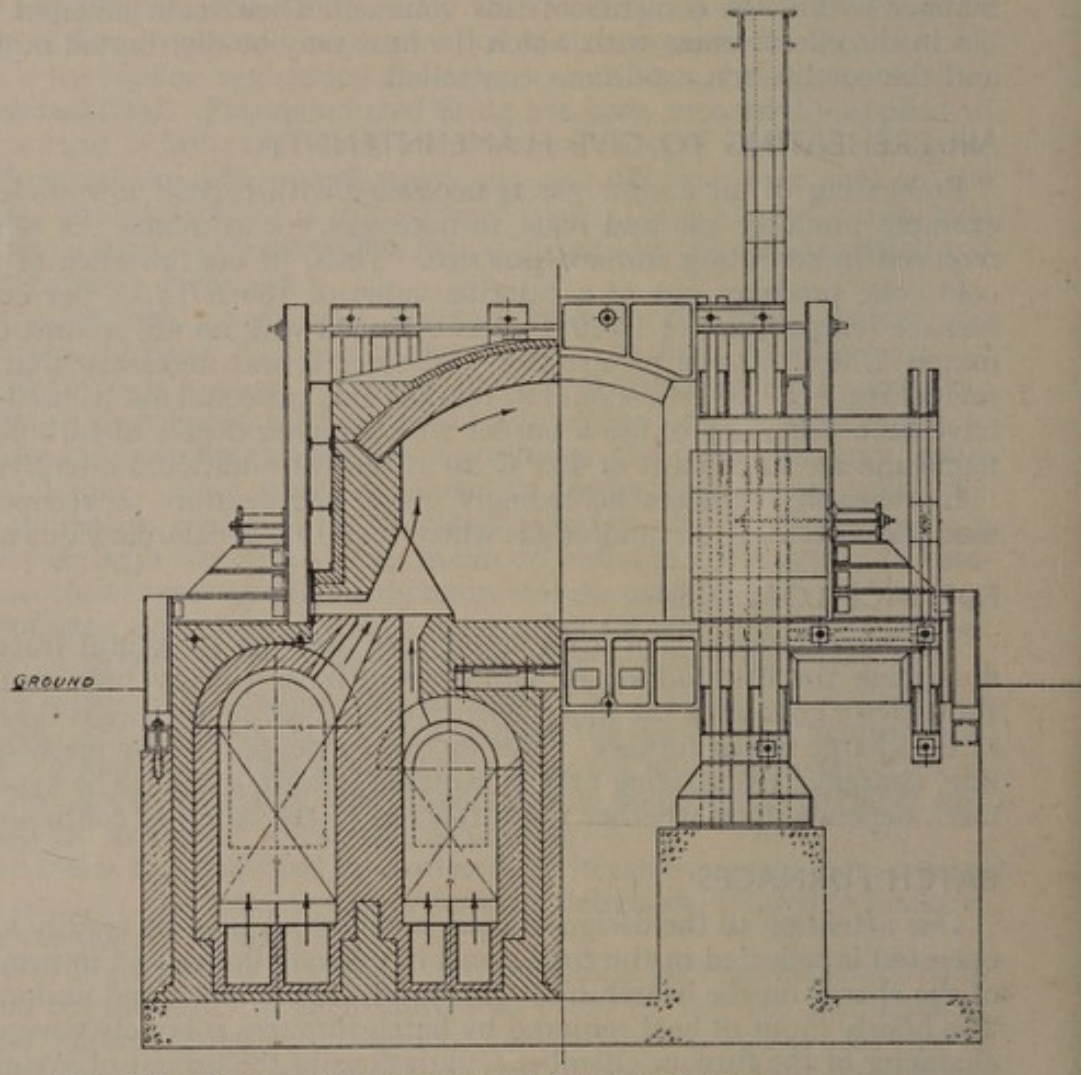

FIG. 157. Regenerative reversing producer gas furnace.

point of fuel economy, but materials may require slow heating for metallurgical reasons. It is a problem distinct from that of the continuous furnace, and cognisance must be taken of the heat absorbed by the furnace structure. The main controllable factors in operation are the rate of firing, the air supply and the balancing of the draught. In coal-fired furnaces much depends upon the skill of the operative as to whether the fire is kept clean and even, of the most suitable thickness, whether fuel of the correct size is fired, and the dampers and air slides correctly operated.

A regenerative reversing solid bottom type, fired by producer gas is shown in Fig. 157. The air regenerators are placed outside the gas regenerators and the ports are located in recessed combustion chambers at hearth level. The disposition of the gas and air reversing valves are shown in the longitudinal section of the furnace. Such a furnace is used for heating large forgings.

CONTINUOUS FURNACES

A great variety of design is used for continuous furnaces for reheating, an effective and popular design being the pusher type. Considerable investigation has been made into the fuel efficiency of this class of furnace, and a voluminous literature may be consulted.

The continuous furnace is favoured because it offers a relatively simple method of conveying, and at the same time heating, material, and hearths may be constructed capable of withstanding high temperatures and the action of the scale falling from the charge. The chamber is built with either side or end

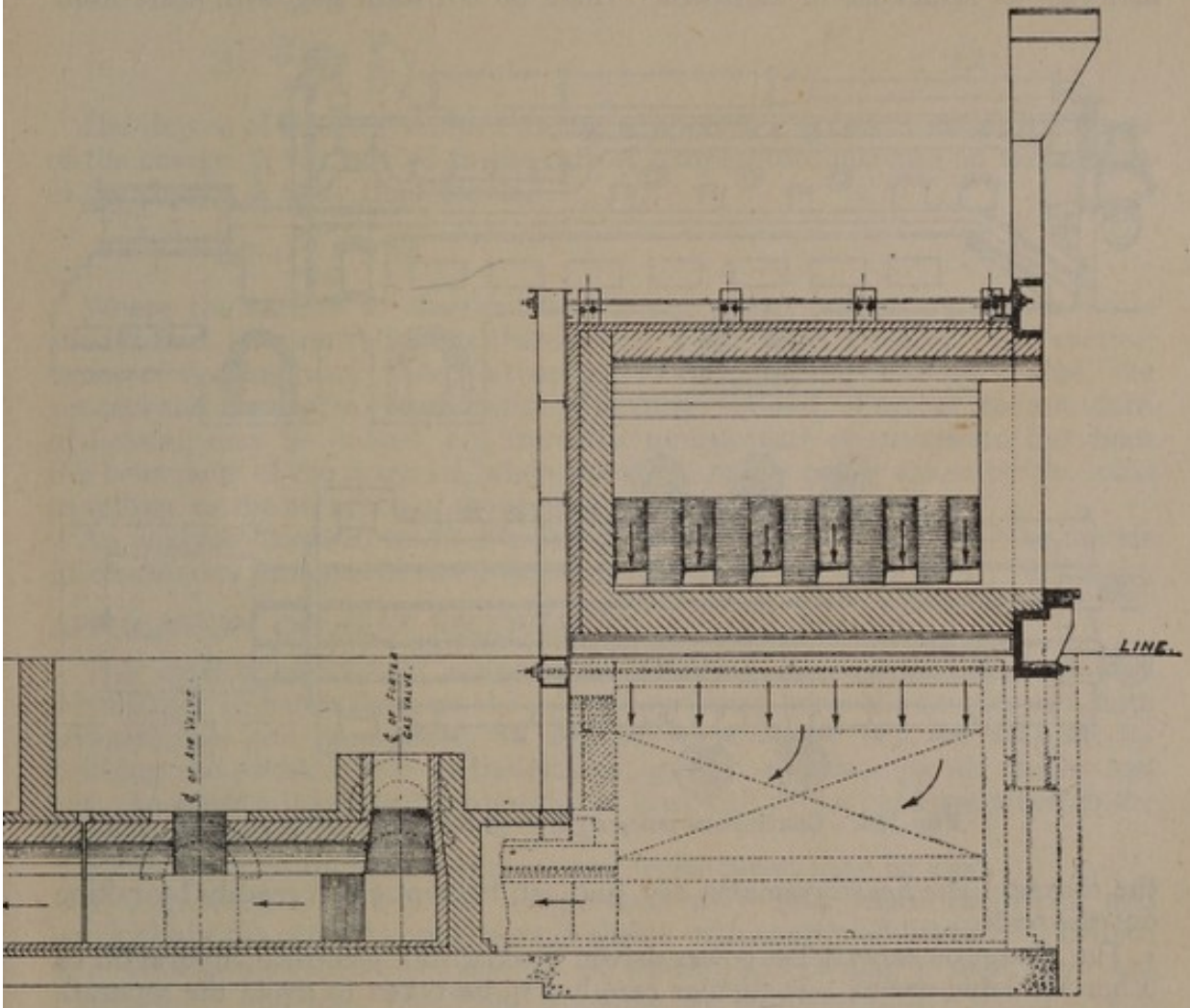

FIG. 157 (*cont.*). Regenerative reversing producer gas furnace.

discharge according to the conditions which obtain for the transfer of the heated material to the forming plant. Furnaces of the side-discharge type must be used when long bars of comparatively small section have to be heated for rolling in continuous mills. In order to avoid excessive speeds at the finishing pass, such bars must enter the first stands of the mill slowly. Accordingly, the rear end of the bar must be kept in the furnace as the leading end enters the first pass.

An essential technical requirement is that the furnace must be capable of providing the necessary uniform final temperature throughout the ingot, bloom or slab. Where a solid hearth is used there is an inevitable chill due to the absorption of heat by the material of the hearth, or by the skid bars over which the charge is pushed. Provision with heavier sections may accordingly be made towards the discharging or soaking end to turn over the material by

convenient means, so that the cold side may be exposed to the hot radiating surfaces of the furnace chamber. Ingots or blooms up to 8 inches diameter may be allowed to drop over the edge of the skids on to a refractory forehearth. Heavier blooms can be operated in this way if the forehearth is reinforced with heavy metal masses, and the temperature not too high to give adequate durability of the hearth.

Ingot manipulation may be used for heavy masses up to 23 inches diameter. The forehearth may be constructed of basic refractory materials, comprising magnesite, chrome or spinel bricks, stabilised dolomite or other suitable material, as brickwork or monolith. These do not form slag with scale from

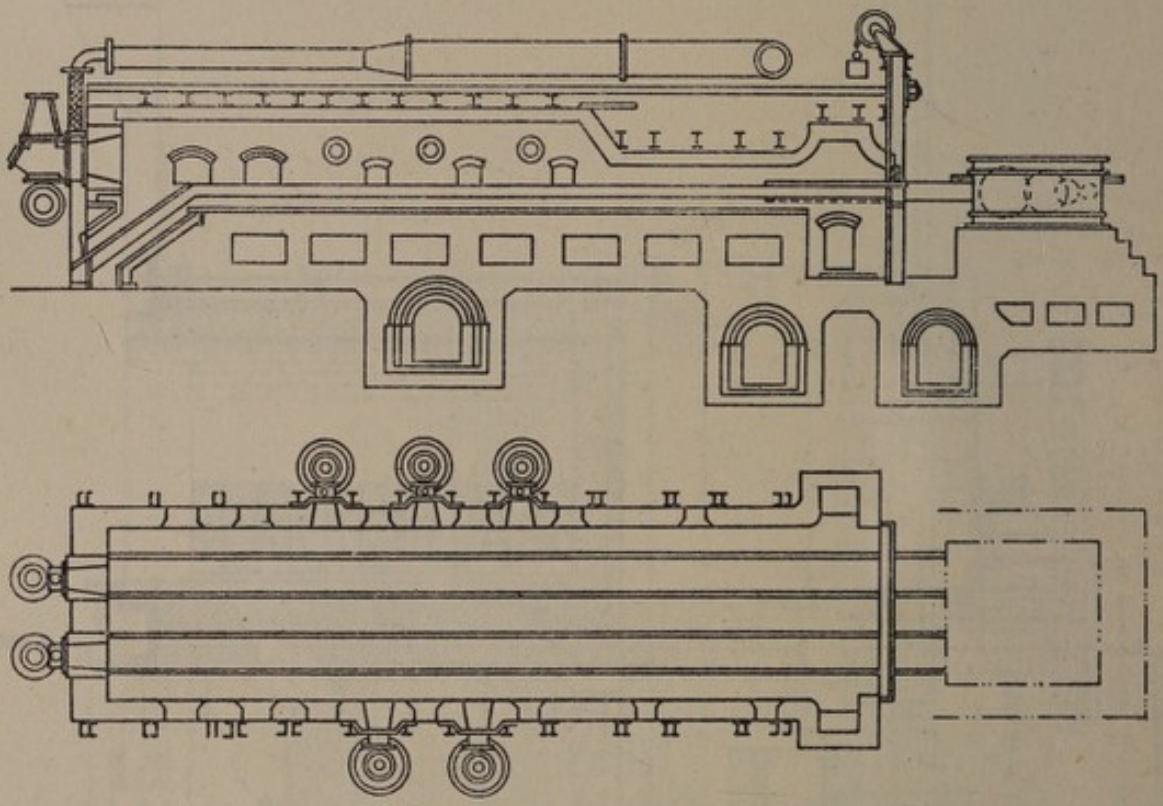

FIG. 158. Continuous pusher gas-fired reheating furnace.

the charge. The hearth remains dry, and can be kept clean readily by raking out the scale.

The charge is moved by power-driven mechanism, electrical, hydraulic or other available means. Particular care has to be taken to avoid the separate pieces riding over one another or fouling the side walls of the furnace. Accordingly, doors placed at suitable intervals may be provided to give accessibility. A continuous pusher type of furnace is shown in Fig. 158. It is equipped with end and side burners for gaseous fuel, and end discharge from a horizontal hearth.

In order to assist the attainment of uniformity and to raise the output of the furnace, the skid bars may be raised either for the whole or a section of the furnace chamber to permit of underfiring. The problem of supporting, anchoring and so safeguarding the rigidity of the bars introduces maintenance difficulties of no mean order, so that in many mills separate batch soaking furnaces are frequently employed in order to ensure the required standard of heating.

TIMES OF HEATING

In the heating of steel, a formula has been given by Helweg for deriving the final character of the soaking at the discharge end of the furnace.

If t = heating time in the furnace, minutes.
ΔL = corresponding length of the hearth, metres.
V = velocity of the charge over the hearth, metres per hour.
D = diffusivity of the charge, square metres per hour.
l = vertical thickness of the material of the charge,

a specific minimum value for the heating time, or alternatively for the corresponding length of hearth required is given by

$$t \geq \frac{18l^2}{D} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

$$\Delta L \geq 0{\cdot}3\ \frac{l^2}{D}\ V \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

The degree of soaking, defined as the temperature gradient within the mass of the charge, $\Delta\ \theta$ is related to the rate of temperature increase on the surface of the charge, $\Delta\ \theta'$ by the formula,

$$\Delta\theta = \frac{l^2}{2D} \cdot \Delta\theta' \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

Where the surface is observed by the use of an optical pyrometer, scale should first be removed from the surface of the heated metal. In practice, times of heating vary widely according to the class of material heated, the process and the degree of uniformity of heating required. Further, the standard of heating may be judged, not from the temperature observations, but from the behaviour of the material, when worked, e.g. the power taken by the mills in rolling, or the accuracy of movement of the tool in piercing.

An analysis has been made of existing data on the times of reheating metals in continuous furnaces in the *Journal of the Institute of Fuel* (1937, X, 355).

APPLICATION OF HEAT RECOVERY

The application of heat recovery to reheating furnaces is practised with advantage. In batch furnaces the exhaust gas must leave the furnace at a high temperature, and accordingly an appreciable economy may be attained by utilising the sensible heat of the exhaust gases for preheating air and/or fuel gas. In general it is possible to utilise at least half of the heat contained in the exhaust gas as air preheat.

Firebrick or metallic recuperators and regenerators may be used.

(i) *Recuperators.* Firebrick recuperators may leak, since firebricks, and particularly the cements used, tend to shrink with continued heating. Further, any refractory structure is subjected on heating and cooling to expansion and contraction, and the repetition of these movements ultimately produces disturbance of the refractory at the joints, which become a serious source of air leakage. Such difficulties are likely to be aggravated by ground vibration such as arises from the use of heavy mechanically-operated hammers or presses. On the other hand, with suitable ground conditions and adequate means of accessibility and maintenance, firebrick recuperators have their uses.

Metallic recuperators are becoming more widely used, since metals have been developed which are suitable for the purpose. They have a low heat capacity, and the construction is readily made air-tight. Accordingly high air velocities may be carried, and the degree of utilisation of the available heating surface is increased, since the coefficients of heat transfer are higher. These are increased by the better heat conductivity of the material of the tube walls as well as by the increased velocities. The limiting condition governing the abstraction of heat from the waste gases is determined by the pressure drop permissible. With induced draught any desired velocities may be obtained, but where this provision is impracticable the use of refractory surfaces in con-

junction with the metallic surfaces effecting the heat exchange is an advantage. The coefficient of heat transfer by convection in slowly-moving gases is relatively low, and this may be compensated for by the radiation from the hot refractory surfaces (Chapter VIII).

The upper limit of attainable preheat of the air in metallic recuperators is 850° C., being fixed by the durability of the material of the recuperator tubes. Above this temperature regenerators must be used. In furnaces fired with gas or pulverised fuel there does not appear to be an upper limit for the useable hot air temperature as far as the durability of the furnace is concerned, because by reducing the amount of fuel burnt per hour, and with a suitable distribution of burners, any desired low working temperature can be maintained. On the other hand, in grate firing the temperature of the primary air is confined to about 300° C., since burning of the grate and clinkering troubles may arise beyond this limit.

(ii) *Regenerators.* Reversing regenerators are most effective in applications in which the temperatures of the exhaust gases are not subjected to large cycles of variation. They have a large capacity for heat storage, and require much ground space for the regenerator chambers, accompanying flues, and reversing valves. Where applicable they are the most efficient form of heat exchanger, and require the least attention for maintenance and renewals.

Where regenerators are used burners must be of simple design, since gas and air pressures are relatively low, and the burner throat becomes on reversal the outlet flue.

(iii) *Preheating Chambers.* The continuous furnace is a combination of heating chamber and preheater, since at the charging end the sensible heat of the products of combustion serves to preheat the incoming charge. In batch furnaces it is occasionally the practice to lead the waste products of combustion from one chamber to preheat the charge in another, but a difficulty may be encountered in overcoming the loss of draught resulting from the resistances due to connecting flues, and the frequent opening of charging doors.

(iv) *Waste Heat Boilers.* The waste heat boiler is another means of recovering sensible heat in the products of combustion leaving the working chamber of a batch furnace. Its advantage depends largely on the economics of each individual case, whether the value of the steam raised is adequate compensation for the costs of installation and operation of the boiler. Cyclical variations of temperature of the waste gases, the avoidance of excessive inleakage of cold air, and whether the resulting variations of output of steam may be economically dealt with have to be considered. It has been in many older installations the practice to apply auxiliary firing to the boiler to maintain the steam output when the furnace output of waste heat falls off, but this practice does not always provide the most economical combustion. The subject is discussed in Chapter XX.

FACTORS DETERMINING THE PERFORMANCE OF REHEATING FURNACES

By far the most important factor controlling the fuel efficiency of a reheating furnace is that of load in relation to the dimensions of the furnace chamber. The quantity of heat absorbed by the structure in bringing the furnace to a required condition is mainly determined by the design and structural features of the furnace, but the rate at which heat may be developed in the chamber and the manner at which the heat absorbing surfaces of the charge are exposed to the heat are controllable.

For similar types of furnaces a relationship exists between the ratio of the useful heat absorbed by the charge to the total heat input, usually defined as the " thermal efficiency " and the rate at which the useful heat is absorbed in the charge per unit of " heating surface." By " heating surface " is here meant

the total heat-absorbing surface. When the hearth is covered with stock, then strictly the total heat absorbing surface should include the crown, side walls, hearth and the surface area of the stock exposed to radiation less the area of hearth covered. In practice the " heating surface " constitutes the flame-swept surfaces, and is roughly approximate in many reheating furnaces to the area of the refractory surfaces. In simple terms it amounts to relating the fuel used per ton to the rate of throughput on a comparative basis of furnace size.

Using the above basis of comparison the relationship has been shown in Fig. 159. Most of the examples illustrated, the full record of which may be obtained from the paper quoted (*J. Inst. Fuel*, X, 355), refer to continuous furnaces.

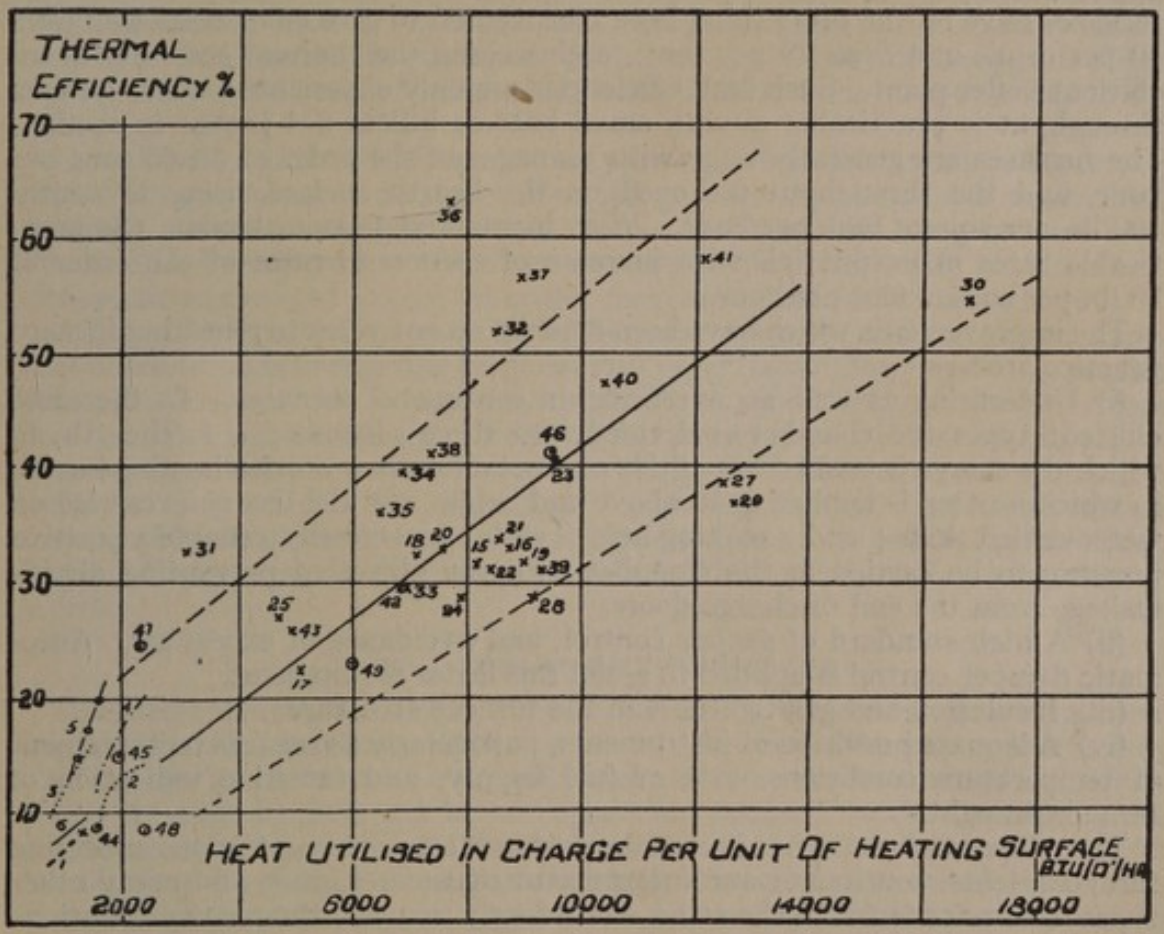

FIG. 159. Relation between thermal efficiency and rate of utilisation of heating surface for furnaces of related type.

The curves for furnaces, shown in 1 to 6, Fig. 159, are derived from plant in which accurate knowledge of the fuel consumption and output was available. They refer to conditions of slow heating in which a high standard of uniformity of temperature was required.

Furnaces 15–23 were of the continuous pusher type fired by coke oven gas and heating long billets covering 50–75 per cent. of the hearth area.

Nos. 24, 25 refer to furnaces of the same size and similar class of charge and output. No. 24 was fired by producer gas, No. 25 by mixed gas of calorific value 210 B.Th.U. per cubic foot.

Nos. 26–30 show in a marked manner the general relationship between thermal efficiency and rate of utilisation of heat per unit of heating surface in spite of some variations in the proportion of waste heat appearing in the products of combustion.

Each of the groups 31, 32 and 33–35 are comparable amongst themselves.

Nos. 36–41 refer to a series of tests by W. P. Chandler (*Am. I.S.I.*, October 27th, 1922) in which the performance of continuous pusher furnaces equipped with recuperators contrasted with that of non-continuous regenerative reversing furnaces. Nos. 42–49 again are all furnaces of varied type, showing the general relationship. The improvement of No. 45 over No. 44 was ascribed to the application of air recuperation in alloy steel tubes and insulation of the furnace.

Other points emerging from such comparisons are that thermal efficiencies of the order of 40 per cent. have been obtained from non-continuous regenerative furnaces, but that at the same time structure losses arising from the maintenance of the whole of the chamber at the working temperature accounted for approximately 35 per cent. of the heat of the fuel. Pusher-type continuous furnaces have on the other hand been constructed to give efficiencies well over 40 per cent., and even 70 per cent., approaching the thermal economy of an efficient boiler plant. Such high efficiencies are only experienced where a rapid throughput is practicable as with small bars or billets 2–4 inches in section. The furnaces are generally large with tonnages of the order of 30–60 tons per hour, and the throughput reckoned on the hearth surface rising to nearly 100 lb. per square foot per hour. With ingots and heavier blooms the practicable rates of output fall with increase of section to rates of the order of 30 lb. per square foot per hour.

The improvements which have been made in recent years to reheating furnace practice are :—

(i) Underfiring as well as overfiring in continuous furnaces. In the most efficient types the chamber is divided into three sections ; a forehearth, in which the charge is pushed over skids on a solid hearth ; a main heating hearth in which heating is applied from above and below, and the charge is carried on water-cooled skids ; and a soaking hearth with a low crown to enable a positive pressure to be carried in the chamber with the object of preventing air inleakage from the end discharge doors.

(ii) A high standard of gas/air control, and avoidance of excess air. Automatic damper control is applied to assist this latter requirement.

(iii) Insulation and gas-tightness in the furnace structure.

(iv) Adequate provision of instruments, particularly as regards measurement of temperature conditions, rate of fuel supply, and sensitive indication of furnace draught.

The reheating furnace has its counterpart in all types of continuous annealing furnaces, lehrs, continuous and intermittent ovens and kilns, and many other types of furnace found in the arts. Accordingly many of the recommendations contained in the discussion of these varied types in various parts of this work may be equally applicable to other industries.

SOAKING PITS

The combination of the large open hearth furnace and the continuous rolling mill involves the provision of means whereby hot ingots may be fed to the mill without being allowed to go cold. The plant used is known as a soaking pit. It is virtually a vertical reheating furnace into which hot ingots are fed for heating for rolling, and the construction in the earlier forms was a series of pits sunk in the ground taking one or more ingots. The pits were put into communication with one another along the lower section for the passage of the heating gases, leaving arches to support the lean of the ingots. The cell pits are constructed in batteries and fired by solid fuel, producer gas with regenerative heat recovery, or coke oven, blast furnace or mixed gas, employed with recuperators, regenerators or waste heat recovery for the raising of steam. Chamber pits are also used having either arched or suspended crowns, the

movable lids being operated in sections. The pits may be fired by separate burners or in batteries.

In modern practice, particularly in the heating of special steels, it has been recognised that accurate control of temperature and of the atmosphere of the pits is essential. Systematic removal of slag is desired on the grounds of durability of the refractory linings. For this purpose the hearth is inclined towards the slag notches from which the slag is removed in small slag bogies, or pans. In certain types of practice a layer of coke breeze is used on the hearth to absorb the slag, which remains in a powdery form and is removed at regular intervals.

The heat requirement may vary from 4·5 to 20 therms per ton of material heated, according to class of charge, plant and conditions of operation. Scaling losses are of the order of 1½–3 per cent. Outputs based on the area of hearth may be as high as 150 lb. per square foot per hour, though it is suggested that a more reasonable basis of comparison is the area of flame-swept surface in the pit.

Gas-air ratio control instruments may be used for distributed fuel gas, e.g. coke oven gas, blast furnace gas or mixed gas, for the purpose of combustion control. Alternatively gas and air meters and recorders may be preferred. Pressure and temperature recorders on regenerators and waste gas exhausts, and sensitive draught gauges on outlet flues are other useful instruments.

Alternative types of plant to serve the same purpose comprise the combination of continuous preheating furnaces and separate soaking furnaces, operated either as "in-and-out" furnaces, or as continuous furnaces with a rotary hearth. The last-named do not act as the counterpart of the soaking pit, as they are confined to bloom and billet heating. Continuous reheating furnaces are now constructed of the bogie type for heating ingots. These may replace soaking pits, since the separate heats can be charged whilst hot on to the waiting bogies. The newest developments provide for the heating of the charge in two layers.

HEAT TREATMENT FURNACES

Heat treatment operations are carried out for the purpose of producing some desired physical condition in the material heated. The treatment usually requires the application of a specific cycle of temperature change in a given time. It may be followed by an operation external to the furnace, which imposes the incorporation in the design of special mechanical appliances for transferring the charge to the plant in which the external treatment is applied. For example, stock may be required to be conveyed through a hardening furnace and then quenched, say in oil, or it may be necessary to apply a specific type of cooling treatment.

Practically all the general principles so far discussed apply in some degree to heat treatment furnaces, but the latter have certain special features as a class, which amount in effect to the need for a high standard of temperature control and uniformity.

CHARACTERISTICS OF HEAT TREATMENT FURNACES

Since, in general, heat treatment operations follow a number of other manufacturing operations, the article treated is much enhanced in value, and greater control must be exercised to avoid loss due to spoilage than in earlier operations. Machining may have been completed, and accordingly the condition of tne surface may require consideration. Warpage may be of significance, and, most important of all, there may be quite a narrow tolerance in the limits of the permissible range of temperature of treatment in order to obtain the desired physical properties in the material heated. Not only must the material be

heated more uniformly, but the temperature to which it is heated must be more accurately controlled than in other furnaces. This implies a greater degree of refinement in the firing, and the need for a greater application of control instruments. Accordingly it becomes advantageous to use fuels and furnaces which are capable of the requisite standard of scientific control.

Further, temperatures are relatively low, in comparison with the types of plant already discussed. Accordingly a greater degree of mechanisation becomes practicable, and in this respect a remarkable development of conveying and other types of auxiliary appliance have been called into service, constructed of suitable heat-resisting materials.

GENERAL UTILITY FURNACES

There are many types of straightforward heating operations—annealing, hardening, tempering, carburising—requiring the application of temperatures below 1,000° C., which are carried out in a simple type of batch furnace. The furnace then becomes in effect a general purpose furnace, and the sole problem of the designer is to apply the necessary flexibility of heating, uniformity of temperature within the limits required, and thermal economy. Floor space available may determine the selection of fuel, and then gas or oil may be used.

In such furnaces the following features of design and operation are important in the control of efficiency :—

(i) Location of burners and flue offtakes to give quick and accurate attainment of uniformity of temperature. In small furnaces bottom heat should be provided, either by the location of heating flues in the hearth or by arranging for the waste gas flues to traverse the brickwork of the hearth.

(ii) Recuperation of waste heat and preheating of the air for combustion by the incorporation of the recuperator elements in the main structure of the furnace.

(iii) Use of refractories of low thermal capacity, since general purpose furnaces are generally used intermittently and the amount of fuel used for lighting-up becomes an item of importance. Hearths are called upon to withstand wear and attrition, and the working surface must be made of a hard firebrick.

(iv) High grade external insulation.

(v) Robust framework, well-fitting, well-lagged doors, readily-controlled dampers and freedom from leakage.

(vi) Pressure gauges or simple orifice flow meters on the fuel, if oil or gas.

Pyrometric control is essential in most heat treatment furnaces. Recording instruments, of the multiple reading type, provide the most effective means of controlling the performance of a battery of these furnaces. Frequent calibration of both couple and instrument are necessary.

A detrimental feature of the small general purpose furnace, which is a frequent source of waste in that indraught of cold air readily occurs round doors, pyrometer holes, lighting-up ports and sight holes, is the provision of a chimney which is too high, and prone to give an excessive and unnecessary draught. Where positive draught is applied the chimney draught required is accordingly reduced. For venting the waste products outside the shop, cowls should be provided into which the necessary length of shortened chimney is allowed to discharge the furnace gases.

In the larger types of general utility furnaces the choice of fuel may be extended to include fuels other than those most readily controlled. Raw producer gas, pulverised fuel, coal fired by underfeed stoker, coke and, in normal times, oil may be applied efficiently. Many are operated on coal, hand fired, but the trend in practice is towards the replacement of hand firing by more controllable forms of heating.

In all types of heat treatment furnaces both large and small, electricity is applicable.

SINGLE PURPOSE FURNACES

It would require a separate treatise to describe adequately the many forms of heat treatment furnaces which are devoted to specific operations, and to enumerate the special points to be watched in maintaining their efficiency. They are usually constructed by expert furnace builders, who have developed their special features as the result of long experience. It is a good practice to enlist the assistance of the constructor at the outset in drawing up simple explanations of the method of operation, with clear sketches, to assist the operative staff to make the best use of the furnace. The essential points to be watched in maintenance as well as in operation should be indicated.

The following survey is restricted to a discussion of selected types, with the object of giving an introduction to a complex problem.

In the construction of single purpose furnaces for heat treatment in addition to the major principles which have already been discussed both in Chapter XVIII and parts of the present chapter, a number of special points have to be given consideration. These are :—

(i) Space uniformity of temperature.
(ii) Temperature regulation with regard to time.
(iii) Protection of material.
(iv) Handling of material.

(I) SPACE UNIFORMITY OF TEMPERATURE

Ideally, to obtain rapid and uniform heating a metal should be exposed to a heating medium which imparts its heat to the charge equally on all sides. The lead bath or the salt bath meet this requirement, but they cannot be applied universally for obvious reasons. An electric furnace with resistors in crown and hearth as well as side walls is the next most suitable agent for the purpose. In all heating a temperature difference must be set up between the heating medium and the surface of the charge. This difference is large in the early stages of heating and diminishes as the soaking stage is reached.

In batch furnaces the temperature differences must be controlled by the firing rate solely ; in continuous furnaces by the control of throughput as well as that of rate of firing. Low rates of heating are in general conducive to a high degree of uniformity, but the thermal efficiency of the operation may thereby be impaired. The space uniformity at any one moment depends mainly on the design of the furnace and the disposition of the charge. Various expedients are used to assist in securing the object in view.

The most satisfactory expedient lies in the disposition of the burners or ports through which the heating gases enter the working chamber. Others are :—

(*a*) The use of muffles or perforated walls.
(*b*) The use of a " lazy " flame, or in electric furnaces, of controlled temperature resistors.
(*c*) Forced circulation of hot gases.
(*d*) Disposition of the heating surfaces at a distance from the charge.

(*a*) *Muffles.* Muffles retard heat transfer and accordingly assist temperature equalisation in the charge. They also permit of the use of controlled atmospheres and therefore serve for the protection of the material heated. An inverse of the ordinary muffle method is to allow the heat to be developed in a muffle with the flame inside. It is developed to its most advanced degree by the use of heat-resisting metal tubes inside of which gas is burnt. The tubes are suitably disposed in the furnace chamber to heat the charge by radiation. These are used in the so-called radiant tube furnaces.

In fireclay muffles the limiting capacity in rate of heating is of the order of 5,000 B.Th.U. per square foot per hour for muffles of not more than 1 foot in the smallest dimension, and 3,000 B.Th.U. per square foot per hour up to 3 feet.

Greater thermal efficiency is obtained where it is practicable to admit the products of combustion to the interior of the muffle. This is generally referred to as semi-muffle heating.

Avoidance of cracking in monolithic muffles and care of muffle brickwork in built-up muffles are essential elements of efficient practice.

Carborundum is used to increase conduction of heat. Heat resisting metals are also applied to muffle practice.

In principle, the use of perforated walls is similar to method (i) (*d*), because the charge is exposed to radiation from small bright spots located at a distance from the charge. Perforated walls are not an invariably successful expedient in attaining uniformity of temperature, for where the gases are moving slowly the hot gases tend to be discharged at the highest ports and they may serve to promote circulation of gases in such a manner that cold layers of gas may collect more readily at the hearth level. Further, they are not necessarily of good stability, and being thin and adjacent to the combustion chamber are subject to failure.

(*b*) *The "Lazy" Flame.* The use of the "diffusion" burner has already been mentioned. In this appliance the fuel gas and air are discharged at separate and adjacent ports, and are allowed slowly to diffuse into one another giving a long flame. The use of producer gas, discharged at ports lying adjacent to relatively large air ports in such a manner that both port velocities are low, gives a soft lazy flame which is particularly effective in giving space uniformity of temperature.

Low temperature electric lamps or gas-heated surfaces emitting a high proportion of infra-red radiation and disposed in large numbers to give uniform heating have been applied to the drying of enamelled ware.

(*c*) *Forced Circulation.* For low temperature heating below 750° C. there is no form of heating so effective in attaining uniformity of temperature as forced circulation of the products of combustion. Reference to Chapter VIII, Heat Transmission, will indicate the influence of velocity on the coefficients of heat transfer at low temperatures. Further, cognisance must be taken of the gas radiation from the products of combustion, from which it must be apparent that by circulation of the gases at velocities which will overcome those of the natural convection currents, improvement in the uniformity of heating must follow. A minimum desirable velocity is of the order of 20 feet per second, but higher velocities of the order of 60 feet per second and upwards are still more desirable. The optimum value must depend upon standards of uniformity required and the gas-tightness of the furnace structure. Many furnaces are in operation in which all temperature difference from a practical standpoint has been eliminated.

With this type of furnace the heating is generally indirect, combustion taking place in a separate chamber. Alternatively, burners may be placed in side walls in separate combustion chambers and the products drawn by induction into the main circulating stream. The hot gases are continually recirculated through the furnace by means of a fan, in any desired direction. A reversing valve may be provided to give alternate direction of flow. A temperature difference of only 2° C. at 600° C. between top and bottom of a vertical furnace 30 feet high in the chamber is attainable by such means. The recirculation fan is placed in a lagged housing, and the bearings are usually water-cooled. The fan is, of course, constructed of heat-resisting metal. Temperature control is readily effected on the main fuel valve, and automatic temperature control thereby made relatively simple.

(*d*) *Disposition of Heating Surfaces.* This implies attention to the disposition of the charge in the furnace as well as to the location of the hot spots. The further the hot spot is from the material heated the more is the heat diffused. The raising of the crown, the placing of the side walls a distance back from the working hearth, the provision of well-radiused crowns, the use of recessed bricks in internal wall surfaces, underfiring of hearths, and the raising of the charge on supports are all expedients serving the same purpose.

Finally in attaining uniformity of temperature there is nothing so important as the avoidance of cold air inleakage.

(ii) TEMPERATURE REGULATION

The use of a pyrometer is essential. A suitable instrument should be placed at a safe point adjacent to the controls. It should be easy to read, and, if of the recording type, give a record which is distinct and readily observable by the furnace operator. It should be robust, dust proof, accessible, simple to maintain, and installed in such a manner that its importance in the furnace operation is readily appreciated. It is becoming the practice in the most efficient installations to incorporate all control instruments in a panel, and either to place the panel in a protected cabin or in such a position that a maximum effectiveness is attained. For the purpose in hand a clock is also necessary.

Automatic temperature control is the logical development of efficient furnace technique.

(iii) PROTECTION OF THE CHARGE

This subject has already been discussed under " Control of Furnace Atmosphere," Chapter XVIII. For general heat treatment of metals it becomes a problem in combustion control. For special processes such as bright annealing some form of muffle heating or its equivalent is applied and a special atmosphere supplied from an external gas generator, with the possible addition of automatic pressure regulation.

(iv) HANDLING OF THE CHARGE

In many heat treatment operations the handling of the charge is mechanised to save labour, for convenience and for rapid discharge to quenching tanks. As with reheating furnaces pusher mechanism is frequently used. Where small parts are to be treated they may be pushed on trays.

CONVEYOR FURNACES

Chain Conveyors. Chain conveyors are convenient for automatic furnaces. The design of the chain varies widely with the shape of the material to be transported. Below 650° C. commercial link chains are frequently used, particularly when the chain does not pass through the heating chamber, but carries an attachment reaching into the furnace. For operation in temperatures above 650° C. more specialised designs may be necessary.

The take-up of the chain due to expansion when it becomes heated must be carefully watched and the tension on the spring controlling the take-up adjusted to avoid stresses which may cause the chain to be strained. Sprocket teeth on driving wheels must be given adequate clearance in order to avoid the links riding up on the side of the tooth.

Chains passing through the working chamber are subjected to repeated heating and cooling even if the return passes through a closed chamber below the hearth, and this cyclical change of temperature imposes a repeated stress which can lead to cracking and failure, particularly in incompletely stable materials. The incidence of cracking is promoted by passage of the hot chain

through a quenching bath, and under such conditions careful selection of the materials of the chain is imperative. Certain austenitic alloys are the most suitable, since they exhibit an absence of change points on heating and cooling, and remain stable under the conditions of service.

Success with chain conveyors depends not only upon the properties of the chain, but also upon uniform distribution of temperature in the furnace. If parallel strands of the chain are subjected to great differences of temperature unequal expansion is bound to arise. Wherever in any individual members non-uniform distribution of temperature occurs the stresses arising result in warpage. Overheating can result in failure due to creep and oxidation. Chains may be equipped with refractory tops and may then transport material through very hot furnaces.

An important element of conveyor design is the proportion of sensible heat lost in the conveyor, and this may be of serious consequence in electrical heating.

ROLLER CONVEYORS

Roller conveyors are usually made of heat-resisting metals though they may also be of composite construction in which refractories play a part. Where made of metal they must rotate just enough to prevent any noticeable temperature difference between top and bottom ; otherwise warping is inevitable. For the conveyance of sheets and plates discs may be used on water-cooled cross-shafts, the discs being staggered on successive shafts. The water-cooling may give rise to considerable heat losses. The driving of the rollers is effected by means of worm gear drives or level gears on a lay shaft, both being external to the furnace.

MONORAIL CONVEYORS

Monorail conveyors are favoured where only part of the piece is to be heated as for instance in brazing work, and in japanning ovens.

SPECIAL CONVEYORS

Many types of special conveyors are now in operation, notably the type known as the walking beam, which is used for conveying sheets in annealing and normalising furnaces. In these the carrier beam is supported by means of rocker bars, which normally rest below the hearth. This member is moved by suitable mechanism to lift the material off its supports placed between the moving beams, carry it forward and deposit it a distance forward again on its supports. Means have been devised for heating the sheets from below as well as from above, and accordingly the design and technique of operation of such furnaces has become a specialised field.

Vertical conveyors are amongst the less common types. Rotating drum furnaces are used for small articles which can be readily handled in heaped masses, and will travel freely and pass through openings without choking them, such as screws, rivets, nuts, rings, punchings.

MOVING HEARTHS

Moving hearths are operated in the form of bogies or car bottoms, as circular or as annular hearths. The last two are usually supported on rollers and are moved by a suitable type of drive and gearing. They have the important advantage that the hearth is always heated, which is conducive to rapid and uniform heating of the stock. The bogie type is favoured for heavy masses. It may be applied to the batch furnace, the tunnel continuous furnace, or to the compartment furnace. This last is in effect a tunnel furnace divided into sections by means of partition doors which are all raised together to allow the

bogies or car bottoms to be moved forward to the next compartment. By this means any specified cycle of heating and cooling can be applied by varying the time of moving the bogies forward and controlling the temperature change in each chamber.

TOP-HAT FURNACE

Finally, there may be instanced the type of sheet annealing furnace in which the sheets are piled on bogie carriages running on a ball track, and shielded from the flame by the use of a steel cover. The old type of furnace consisted of a brick chamber which had one or more fireplaces disposed at convenient points according to the ideas of the builder. These have been replaced by the continuous type, employing the tunnel or mechanical conveyor, and by a batch type operating on piled sheets over which the furnace body is dropped.

This last type, referred to colloquially as the " top-hat " furnace, is heated either electrically or by means of gaseous fuel. In the electrical type the resistors are placed inside the furnace cover which is lifted over the pile of sheets. In the gas-fired type, use is made of the radiant tube already described, and arranged either vertically or horizontally. Controlled atmospheres are operated underneath the covers by the admission of suitable gas from an external generator.

This type is applicable to moderate annealing temperatures. The operation of the furnace is comparatively simple as clean gas is used. Each radiant tube made of heat-resisting alloy is provided with a simple burner blowing into the tube. The burners are connected to gas pipes and combustion may be controlled by any of the conventional methods.

For large outputs of material, mechanical furnaces have many advantages. They are virtually complicated machines, and may be fully equipped with instruments such as combustion controls, pyrometers and draught regulators. Temperature and atmosphere control are matters of special concern, and so are careful attention to mechanical parts and the selection of the materials used for the conveyor.

LEAD POTS AND SALT BATHS

Lead pots, salt baths and similar baths of molten material are used to provide rapid and uniform heating at lower temperatures. This is possible since the heat transfer coefficients of a liquid in contact with a solid are of a relatively high order compared with the alternative methods of heating applicable to low temperatures with the exception of that employing forced circulation of gases.

The most glaring sources of waste are concerned with the question of loading. These molten baths must normally be operated at constant temperatures. They are generally fired either by gaseous or liquid fuels, though electricity is also used. A steady input of heat is therefore essential, but when they have been properly adjusted they should operate continuously with little attention. They cannot be readily shut off when the supply of work for treatment ceases, and it is necessary, therefore, for economical working for a steady throughput of material to be provided. Otherwise it is desirable to work for several days at a time and then shut down.

They have, however, disadvantages. The salt retained on the surface of the articles has a tendency to change the composition of the quenching bath. The molten content of the bath has a tendency to creep up the sides and over the lip and when in contact with the outer surface to cause severe corrosion of the material of the pot. Uneven heating, the local influence of the flame, temperature differences in heating up with the resulting strains, the incidence of too sharp corners may all contribute to cracking of the containers. Unless satisfactory materials are used for the pots their durability is reduced. In this

respect too the disposition of the burners is an important factor, and there must be precautions against local overheating from this cause.

HIGH TEMPERATURE HEAT TREATMENT FURNACES

For the hardening of tools of high-speed steel, chamber furnaces, fired either by gas or oil, or electrically heated are employed. The main requirement is rapid heating attained by the use of burners of adequate intensity. The waste gases from the finishing chamber are used in an upper chamber for the purpose of preheating. Temperatures are sometimes controlled visually since the hardening temperature is revealed by a change in the appearance of the scale on the tool, but the use of the optical pyrometer is recommended for certain and accurate control.

Recently high temperature salt baths heated electrically have been developed for these purposes, and they are capable of a high standard of scientific control by means of suitable instruments.

GAS EQUIPMENT

For many heating operations in the industrial furnaces described in the preceeding sections, gas producers are used as the source of heating gas. The layout of a producer plant will be seen from Fig. 160. Town gas is used in many heating operations and the equipment is described in Chapter XXVII.

FURNACE CONTROL

Much has already been said in the foregoing chapters in regard to recommendations on the practical measures which may be taken as a day to day means to ensure the efficient operation of furnaces. There remain, however, certain points which apply only to particular types. The following are typical of a list of hints, which may be devised for the purpose in view. They are not by any means exhaustive, and others may be found in the various handbooks which are prepared by individual industries for their special purposes.

I. MELTING FURNACES

(i) All melting equipment should be used to its maximum capacity.

(ii) Where practicable the optimum rate of driving to give quick melting should be discovered by investigation and maintained.

(iii) Control of combustion conditions and of draught are of special importance in high temperature furnaces.

(iv) Advantage should be taken when rebuilding to establish construction on the lines of the best practice. The correlation of the changes made with the furnace performance systematically recorded assists in arriving at the most effective design for the practice applied.

(v) Constant watch should be maintained on fuel supply, valves, dampers, flues, draught, sources of waste heat as in covers, doors, linings, and points of possible air inleakage.

(vi) Instruments installed should be well maintained and used.

II. REHEATING FURNACES

(i) Records of fuel consumption and rate of output should be systematically kept and studied in relation to varying shop conditions.

(ii) Optimum rates of fuel consumption should be determined for the outputs required, and controls adjusted accordingly. Correct settings of valves and dampers may be marked for the use of the operators. The quickest heating is the most economical if practicable.

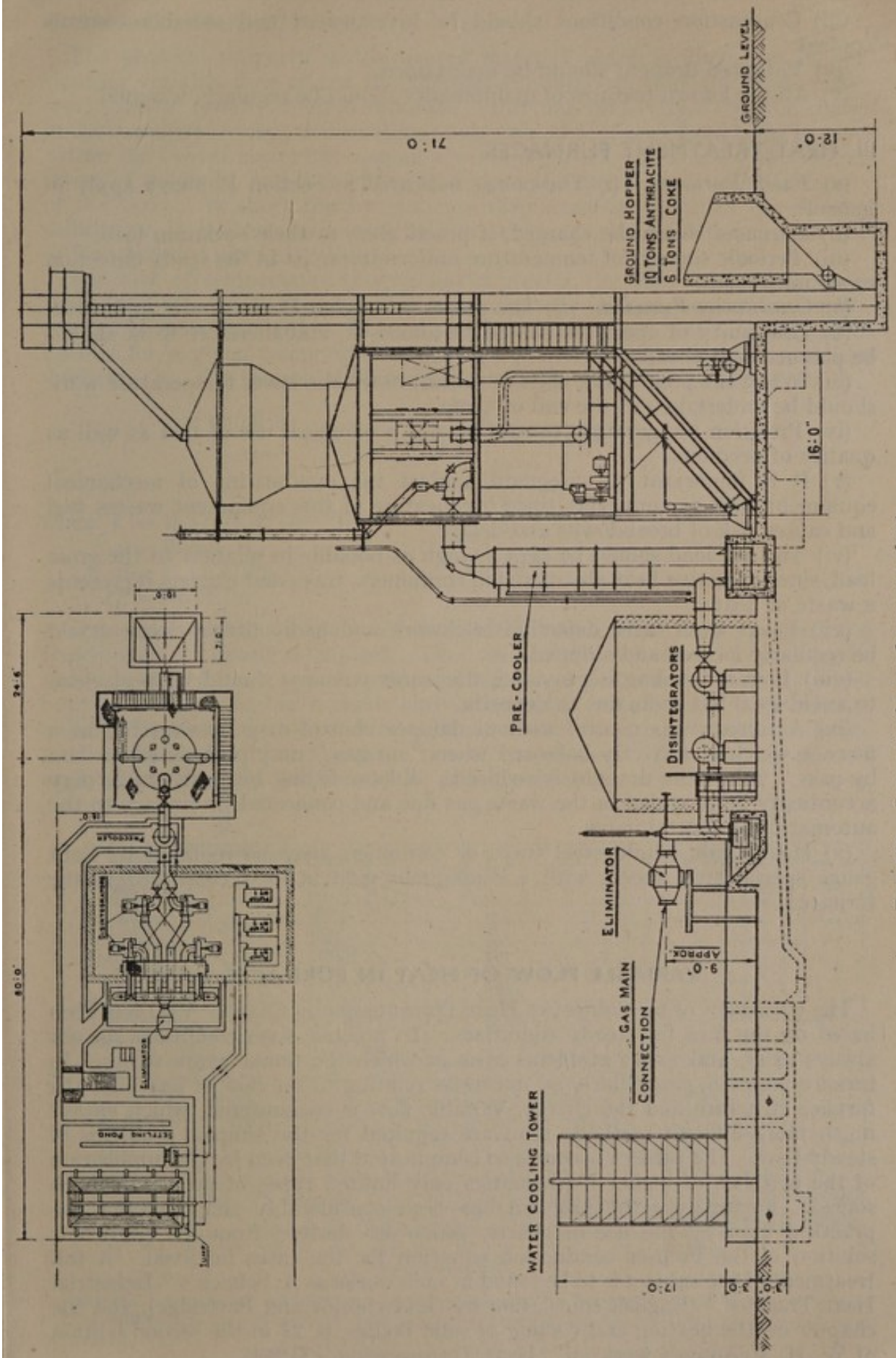

FIG. 160. Lay-out of a producer gas plant.

(iii) Combustion conditions should be investigated and suitable controls applied.

(iv) Balanced draught should be maintained.

(v) All the known features of maintenance should be regularly watched.

III. HEAT TREATMENT FURNACES

(*a*) *Batch Furnaces.* (i) The points indicated in Section II above apply in general.

(ii) Furnaces should be charged, if practicable, to their optimum load.

(iii) Periodic surveys of temperature uniformity assist in the ready detection of air inleakages.

(*b*) *Continuous Furnaces.* (i) The points in Section II above are applicable.

(ii) Continuity of operation should be aimed at, and therefore work should be planned well ahead.

(iii) When the programme calls for a shut down the lower temperature work should be undertaken at the end of a run.

(iv) Precision temperature control attains economical use of fuel as well as quality of products.

(v) It is important to safeguard against the overheating of mechanical equipment such as conveyors since overheating of this equipment wastes fuel and causes risk of breakdowns and delays.

(vi) The net load should be kept as high as possible in relation to the gross load, since excessive heat abstracted in containers, trays and carriers represents a waste of heat.

(vii) Leaks from seals, defective brickwork and badly fitting doors should be regularly located and stopped.

(viii) If water-cooling is provided, the water passages should be kept clean to avoid local hot spots due to deposits.

(ix) Automatic gas control without damper control may be wasteful, as a furnace which is correctly balanced when " on gas " may pull air when " on by-pass," unless the draught is reduced. A loose-fitting butterfly valve may accordingly be provided in the waste gas flue and connected by linkage to the automatic gas control valve.

(x) Heavy gauge mild steel trays or containers may be replaced by light gauge special steel sheets with a consequent gain in heat conserved in the furnace.

VARIABLE FLOW OF HEAT IN FURNACES

The treatment of the subject of Heat Transmission in Chapter VIII has been based on the flow for steady conditions. In practice such conditions do not always exist, and many problems arise in which the temperature varies. In furnace practice, particularly on questions relating to the flow of heat into the furnace structure and the charge, variable flow is encountered, which entails much more difficult methods than are required for the simpler problems of steady flow. The subject becomes so complicated that even by the application of the methods of higher mathematics only limited types of problem can be solved. Fortunately the problem has been considerably simplified for the practical man by the use of charts, which are derived from the particular solution of the Fourier conduction equation for the cases involved. A full treatment of the subject is to be found in such works as A. Schack's " Industrial Heat Transfer " (English translation by Goldschmidt and Partridge), and the chapter on the heating and cooling of solid bodies, p. 27 in the second edition of W. H. McAdam's work on " Heat Transmission " (1942).

THERMAL DIFFUSIVITY

The physical property of the heated material which is important in considering variable flow is not the thermal conductivity, but the temperature conductivity, usually referred to as the " thermal diffusivity." As the charge in the furnace is heated there will not only be a flow of heat from layer to layer within the heated body, but also an absorption of sensible heat by the various layers of the body, and this will depend upon the specific heat and the density of the body. In short the temperature displacement, of which the thermal diffusivity, D, is a measure, must be inversely proportional to the specific heat of the material per unit volume.

The rate of temperature progression is greater the faster the temperature rise of the adjacent layer, and this temperature is in turn proportional to the amount of heat flowing in. The laws of heat flow define that this amount of heat is for a given temperature difference proportional to the thermal conductivity, k. The thermal diffusivity, D, with the choice of suitable units

$$\text{is equal to } \frac{k}{\text{specific heat per unit volume.}}$$

$$\text{or } D = \frac{k}{c\rho}$$

when k = B.Th.U./ft./hr./°F.
ρ = lb./cu. ft.
c = B.Th.U./lb./°F.
D = sq. ft./hr.

The cases which have been solved generally require that some specified and steady rate of heating is implied. Thus the surface of the solid body may be suddenly raised to a specific temperature, a condition which strictly never happens in practice, but a result may be obtained of sufficiently approximate value to be of practical use. Alternatively a rate of heating may be applied to give a steady rate of temperature rise at the surface. This may again only apply approximately, but the results may be of value.

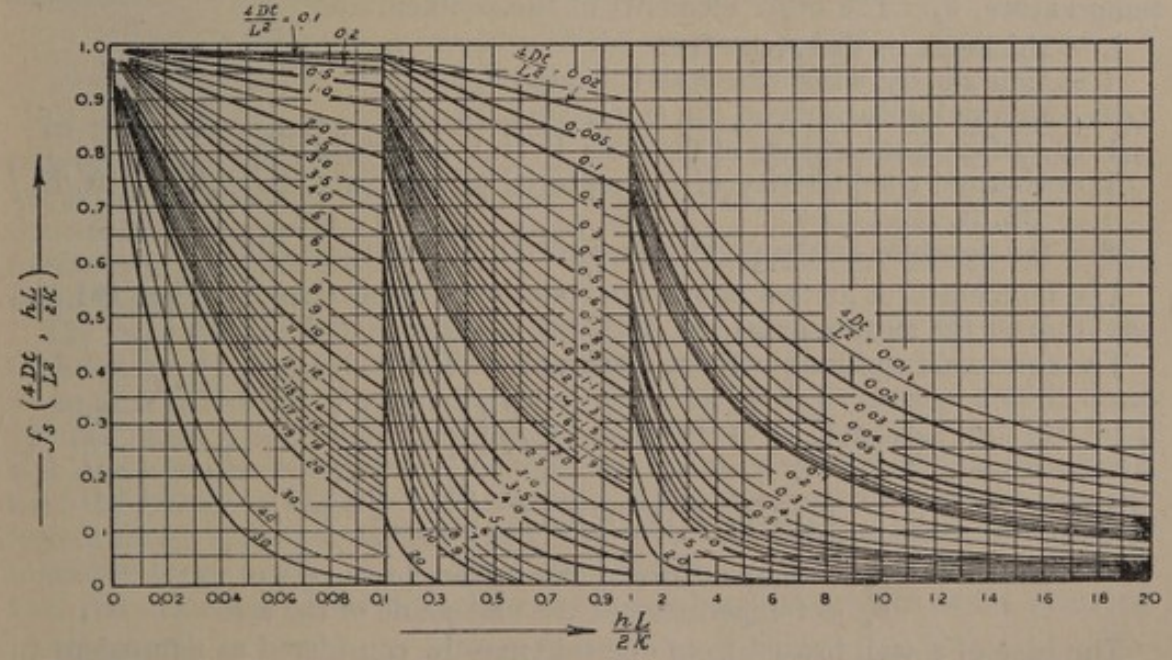

FIG. 161. Graphical calculation of rate of change of temperature of the surface of a wall of finite thickness in contact with a medium from which heat is being transferred to it (Grœber.)

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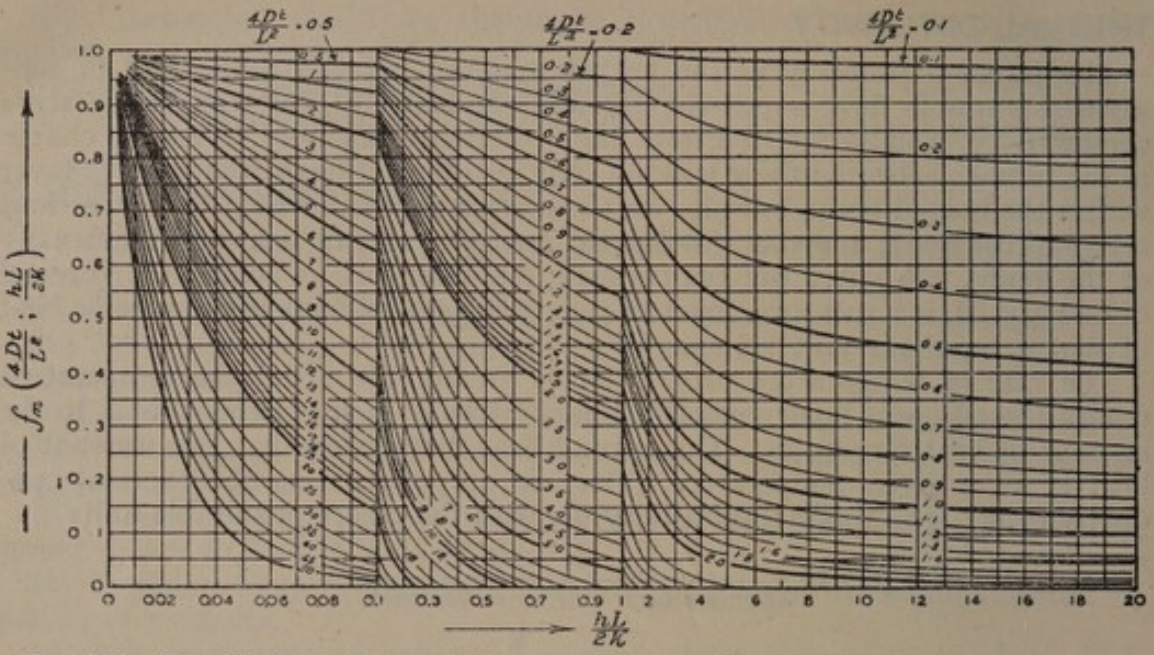

FIG. 162. Graphical calculation of the rate of change of temperature of the central plane of a wall of finite thickness in contact with a medium from which heat is being transferred to it. (Grœber.)

(Reprinted by permission of the publishers (Chapman and Hall) from "Industrial Heat Transfer," by A. Schack.)

By way of example, the data shown in Figs. 161 and 162, which refer to solutions obtained by H. Grœber (Ref. "Einführung in die Lehre von der Wärmeübertragung," Introduction to the Study of Heat Transmission, J. Springer, Berlin (1926), p. 40), apply to a wall or plate of finite thickness in which the flow of heat is normal to the surface, as in large walls, slabs or plates. Any end effect introduces flow in more than one direction and then the problem becomes still more complicated.

The charts enable the problem of the slab or wall to be solved when the plate at an initial temperature, θ_1, is placed in a heat transfer medium at a temperature, θ_2. The other elements of the problem are :—

L = thickness of the plate, feet.
t = time during which the slab is heated, hours.
D = thermal diffusivity of the material of the plate.
h = surface conductance of the slab, B.Th.U./sq. ft./hr./°F.
k = thermal conductivity of the material of the plate, B.Th.U./sq. ft./hr./°F./ft.
θ = temperature at the point investigated after t hours, °F.

The temperatures at two points are considered respectively in Figs. 161 and 162, that at the surface, Fig. 161, and that at the centre, Fig. 162.

The equations applying may be expressed as follows :—

$$\theta_s = \theta_2 + (\theta_1 - \theta_2) f_s \left(\frac{4Dt}{L^2} \cdot \frac{hL}{2k} \right)$$

and

$$\theta_m = \theta_2 + (\theta_1 - \theta_2) f_m \left(\frac{4Dt}{L^2} \cdot \frac{hL}{2k} \right)$$

θ_s = temperature at the surface of the slab
θ_m = temperature at the mid plane of the slab

The case of a wall heated from one side may be considered as equivalent to that of a slab of double the thickness, ignoring in the first approximation the character of the heat flow from the opposite surface, or alternatively considering it as a perfectly lagged wall.

The method will now be applied to a 9-inch firebrick wall, covered by 3 inches of insulating brick ; it is desired to know the heat absorbed by the wall at the end of twelve hours if a temperature of 1,000° C. (1,832° F.) is suddenly applied to the inner face. The wall is initially at a temperature of 15° C. (60° F.). The surface conductance, h = 4 B.Th.U./sq. ft./hr./°F. The thermal properties of the firebrick may be taken as given by :—

$$\left.\begin{array}{l} k = 0{\cdot}75 \text{ B.Th.U./sq. ft./hr./°F./ft.} \\ c = 0{\cdot}26 \\ \rho = 115{\cdot}4 \text{ lb./cu. ft.} \end{array}\right\} \text{ whence } D = 0{\cdot}025 \text{ sq. ft./hr.}$$

The wall may be considered as equivalent to a slab 18 inches thick heated from two sides, the insulating brick being regarded for the purposes of a useful approximation as equivalent to a perfect insulator (see *Trans. Ceramic Soc.* XXX, p. 106, 1932).

$$\frac{4Dt}{L^2} = \frac{4 \times 0{\cdot}025 \times 12}{(1{\cdot}5)^2} = 0{\cdot}53 \; ; \; \frac{hL}{2k} = \frac{4 \times 1{\cdot}5}{2 \times 0{\cdot}75} = 4$$

From Fig. 161 $f_s\left(\frac{4Dt}{L^2}, \frac{hL}{2k}\right) = 0{\cdot}16$

From Fig. 162 $f_m\left(\frac{4Dt}{L^2}, \frac{hL}{2k}\right) = 0{\cdot}535$

whence temperature after twelve hours

at the surface, $\theta_s = 1{,}832 + (60 - 1{,}832)\ (0{\cdot}16) = 1{,}548{\cdot}5°$ F.

at the mid plane or interface in the case chosen between the firebrick and the insulating brick, $\theta_m = 1{,}832 + (60 - 1{,}832)\ (0{\cdot}535) = 883{\cdot}5°$ F.

The mean temperature of the firebrick is accordingly 1,216° F. and the heat absorbed after twelve hours per square foot of internal surface

$$= 1{,}156 \times \text{heat capacity of the firebrick per °F.}$$
$$= 1{,}156 \times 0{\cdot}26 \times 0{\cdot}75 \times 115{\cdot}4 = 26{,}010 \text{ B.Th.U.}$$

In this example, the surface conductance depends upon the nature of the convective and radiative heating. In a furnace with a luminous flame, the value of the conductance, h, may rise as high as 24 B.Th.U./sq. ft./hr./°F., and then the heating would be more rapid, the mean temperature at the end of twelve hours would rise to 1,486° F. (808° C.), and the heat storage to 33,400 B.Th.U./sq. ft. of wall surface.

The experiments cited (loc. cit.) show that the insulation at the back of the firebrick would absorb less than 5 per cent. of this quantity of heat.

An alternative set of charts based on the work of H. P. Gurney and J. Lurie (*J. Ind. Eng. Chem.*, **15,** 1,170 (1923)), are given in Figs. 163 and 164 for a slab and cylinder respectively. The basis is similar to that of Grœber, except that the groups of variables are arranged differently and the charts within the legible range are more easily read. The groups of non-dimensional variables resemble those applicable to the former diagrams, the main difference being that the value R is the half thickness of the slab and the radius of the cylinder.

$$\Delta = \frac{\theta_2 - \theta}{\theta_2 - \theta_1}, \; m = \frac{k}{hR}$$

$$\rho = \frac{r}{R} \quad , \quad \tau = \frac{Dt}{R^2}$$

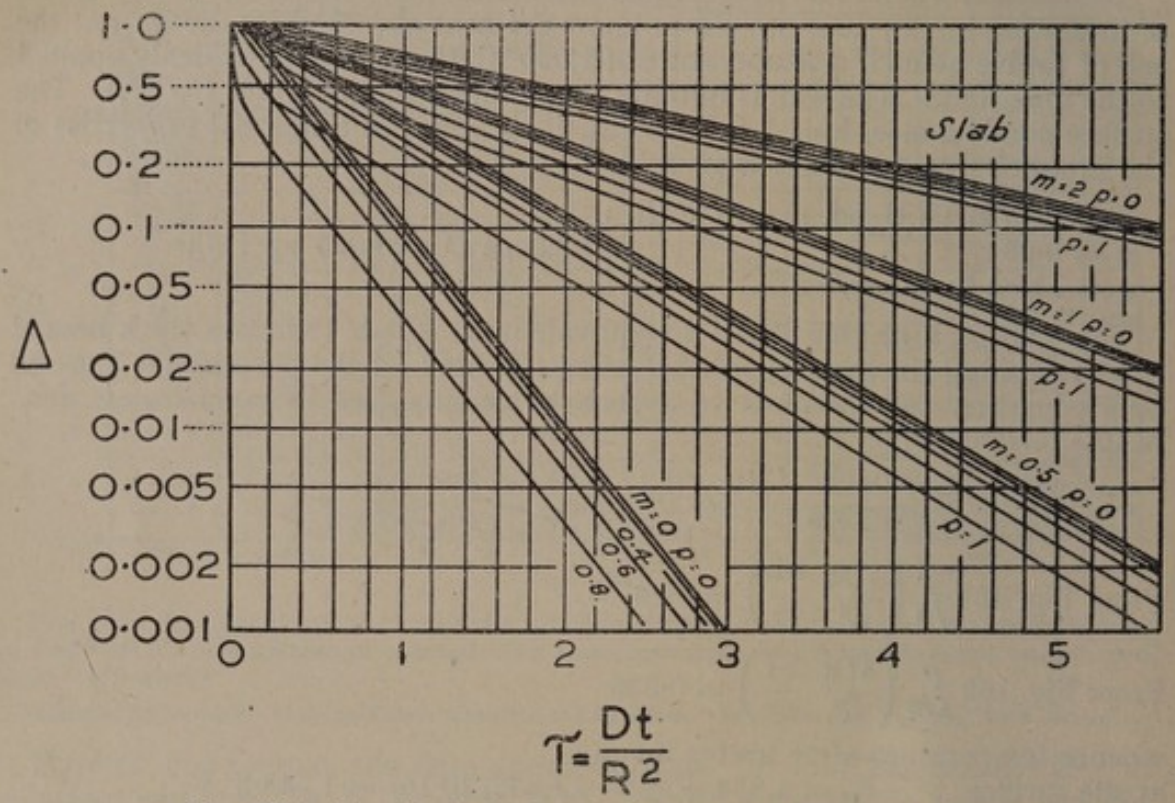

FIG. 163. Graphical calculation of the rate of heating of slabs.
(Gurnie and Lurie, *J. Ind. Eng. Chem.*, XV, 1170.)

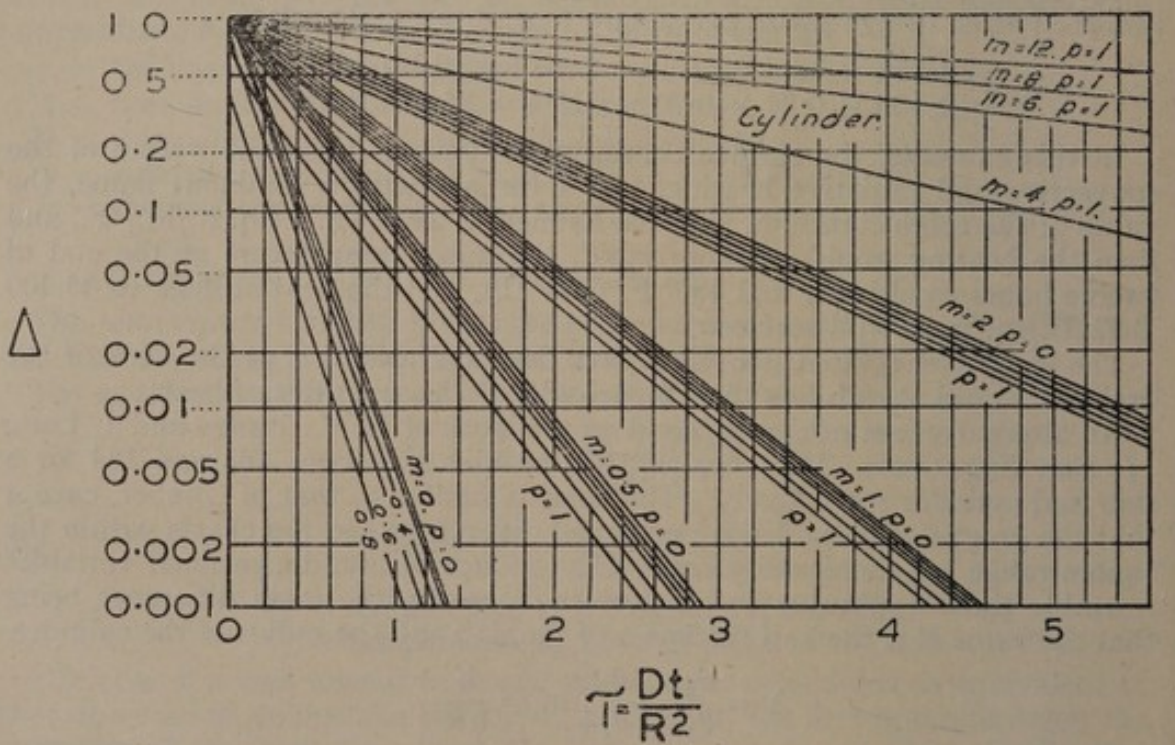

FIG. 164. Graphical calculation of the rate of heating of cylinders.
(Gurnie and Lurie, *J. Ind. Eng. Chem.*, XV, 1170.)

where r = the distance (feet) from the centre of the material to the point at which the temperature θ, after time t hours is required. τ is the "relative time." θ_1 is the initial temperature of the mass ; θ_2 is the temperature of the heating medium.

Thus when $\rho = 0$, the centre temperature, θ_m, is obtained, and when $\rho = 1$, the surface temperature, θ_s, is given.

The use of the chart is shown by an example, in which it is required to determine the uniformity of temperature between centre and surface of a cylindrical steel bloom 24 inches in diameter, after periods of 4, 6, 8 and 10 hours heating. The furnace chamber is at 1,150° C. (2,102° F.), θ_2, and the bloom on charging is at 500° C. (932° F.), θ_1. The heating is to be considered to be uniform all round the mass, a condition which is not strictly applicable in practice but which has to be assumed for the purpose of the computation. The thermal constants of the material, and the surface conductance are to be taken as follows :—

$$h = 40, \quad k = 22, \quad D = 0{\cdot}29$$

whence $$\tau = 0{\cdot}29t$$

and $$m = 0{\cdot}55$$

$$\Delta_m = \frac{\theta_2 - \theta_m}{1{,}170}; \quad \Delta_s = \frac{\theta_2 - \theta_1}{1{,}170}$$

The following values of the temperature differences, $(\theta_2 - \theta_m)$ and $(\theta_2 - \theta_s)$ may be derived from values read from Fig. 164.

TABLE 93

t =	4	6	8	10
$\theta_2 - \theta_m$ ($\rho = 0$)	82	22	5	1·4
$\theta_2 - \theta$ ($\rho = 1$)	41	8	2	0·6
$\theta_m - \theta_s$	41	14	3	0·8

The time required in practice for heating such a bloom to a serviceable uniformity at the temperature stipulated would be of the order of 8–10 hours. There are the following differences between theoretical and practical conditions usually ruling. The furnace picks up heat gradually ; it is not suddenly raised to the working temperature. There is always a gradient from hot gases to metal. Temperature distribution is not ideally uniform and heating is usually greater from one side. A uniformity of ± 5° C. (± 9° F.) must be regarded practically as very satisfactory and ± 10° C. (± 18° F.) as workable.

REFERENCES

British Standard Specifications re Furnace Testing—

No. 992–1941. Fuel Fired Melting Furnaces used on the Non-ferrous Metals Industry
No. 979–1941. Open Hearth Melting Furnaces for the Refining of Steel.
No. 1081–1942. Kilns for Heavy Clay Ware, including Refractory Materials.
No. 995–1942. Gas Producers.
No. 999–1942. Coke Ovens.

Heat Transmission Furnaces—

" Industrial Heat Transfer," by A. Schack (translated by Goldschmidt and Partridge)
" Heat Transmission," by W. H. McAdams, 2nd edit.
" Heat Transmission," by M. Fishenden and O. A. Saunders.

CHAPTER XX

WASTE HEAT BOILERS

The availability of waste heat—Principles of heat recovery in waste heat boilers—Draught—Air inleakages—Feed water treatment—Tube cleaning—Ancillary plant—Efficiency—Economics—Boiler selection.

ACCORDING to the temperature required in industrial heating operations the flue gases must leave at a more or less elevated temperature which will obviously be above the temperature to which materials are heated in the furnace. Part of the sensible heat of these gases will be required to create the necessary draught if a chimney is used for this purpose. If a fan is used for creating the draught the outlet temperature can be very much lower. The gap between the outlet temperature of the furnace and the minimum permissible temperature of discharge represents heat which could be saved. In Chapter XVIII the use is described for this purpose of recuperators or regenerators which preheat the air, and sometimes also the fuel gas.

The amount of heat abstracted from the gases by regenerators may be so great that the outlet temperature is only sufficiently high to provide the necessary draught in a chimney. An example of this is found in regenerative coke ovens where the gases are reduced to some 200°–250° C. (393°–483° F.), at the outlet of the regenerators and in consequence no further heat can usefully be abstracted from the gases.

The primary purpose of regenerators, however, is not to extract all possible heat from the gases in order to improve the overall efficiency, but to return sufficient heat in the fuel gas and air to produce the necessary temperature in the furnace. In consequence it frequently happens that the waste gases leave regenerators at a considerable temperature. An example of this is the open hearth steel furnace, where the gases may leave the regenerators at temperatures of the order of 450°–600° C. and will thus contain a good deal of heat which can be recovered and put to useful purpose.

Another example is in the retorts used in the gas industry. Here the air required for combustion is preheated by recuperation. The producer gas is generally made from coke in producers using very little steam. Under these conditions every cubic foot of producer gas requires about 0·8 cu. ft. of air and yields 1·6 cu. ft. of total combustion products. There are thus 1·6 cu. ft. of outgoing products leaving the combustion chambers at a temperature of some 1,000°–1,200° C. (1,832°–2,192° F.) ; flowing in counter current is less than half this volume of incoming air. Even if the incoming air were heated to the temperature of the outgoing producer gas the heat abstracted would only be half of that initially present and thus at least half the total original heat in the flue gases leaving the settings is available for further recovery. Waste heat boilers are used under these circumstances.

There are many furnaces, particularly those operating at relatively low temperatures, in which no system of recuperation or regeneration of the air is used ; if the gases are exhausted directly from the furnace to the atmosphere, their sensible heat would be lost completely. Under these circumstances the heat is recovered by a waste heat boiler.

Some examples of the temperature of gases leaving furnaces of various types are given in Table 94.

The waste heat boiler has been briefly described in Chapter XIII. This present chapter contains a more detailed discussion of the recovery of surplus heat by waste heat boilers and of the operating conditions necessary for efficiency.

TABLE 94

	°F.	°C.
Nickel-refining furnaces	2,500–3,100	1,371–1,704
Beehive coke oven	1,950–2,300	1,066–1,260
Black-ash or black-liquor furnaces	1,800–2,200	982–1,204
Copper reverberatory furnaces	1,650–2,000	899–1,093
Zinc-refining furnaces	1,700–2,000	927–1,093
Heating and puddling furnaces	1,700–1,900	927–1,038
Copper-refining furnaces	500–2,200	260–1,204
Cement kilns (dry process)	1,150–1,500	621–815
Cement kilns (wet process)	800–1,200	427–649
Open hearth furnaces (producer gas)	850–1,100	454–593
Gas retorts (non-regenerative)	1,300–1,500	704–815
Oil stills	900–1,000	482–538
Glass tanks	800–1,000	427–538

HEAT RECOVERABLE FROM FLUE GASES

The heat recoverable from flue gases depends on the efficiency of the boiler, on the quantity of the flue gases and on the gas temperature drop through the boiler.

The theoretical heat recovery, H, can be calculated from the expressions (Chapter V) :—

$$H = Ws_w (t_1 - t_2) \text{ B.Th.U.} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

$$H = Vs_v (t_1 - t_2) \text{ B.Th.U.} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

where W = weight of gases in lb.

V = volume of gases in cubic feet measured at 60° F. and 30 inches bar.

s_w = mean specific heat of gases between t_1 and t_2° F. in B.Th.U. per lb. in (1).

s_v = mean specific heat of gases between t_1 and t_2° F. in B.Th.U. per cubic foot in (2).

t_1 = temperature of gases entering the waste heat boiler — °F.

t_2 = temperature of gases leaving the waste heat boiler — °F.

In practice there is a certain amount of loss from exposed surfaces which may account for some 5 per cent. so that the actual heat recovered is likely to be about 95 per cent. of the figures given by this calculation.

It will be seen that the higher the temperature of the incoming gases and the lower the temperature of the outgoing gases, i.e. the greater the value of $t_1 - t_2$, the greater will be the amount of heat that should be recovered (cf. Table 98).

An example of this from gas works practice is given in Fig. 165, in which the gases result from the combustion of a producer gas having the composition : CO, 25 per cent. ; H_2, 8 per cent. ; CO_2, 5 per cent. ; N_2, 62 per cent. The gas is generated in a producer having a hot gas efficiency of 86 per cent. The flue gases leave the waste heat boiler at 230° C. (446° F.), the boiler operating at 160 lb. per square inch (page 491).

In this figure particular attention is directed to two points : (1) the great effect of the boiler inlet temperature ; (2) the very considerable effect of excess air used in the combustion of the gases. It will be shown later in this chapter that if the excess air is introduced as a result of leakage after the combustion chamber the effect is even more marked.

Some examples from industrial practice are given in Table 95.

TABLE 95

Furnace	Output	Fuel consumption	Gas Weight lb./hr.	Boiler Heating Surface sq. ft.	Entrance and exit gas temp. ° F.	Output lb. steam from and at 212°F. per hour
Cement kiln, 2 kilns per boiler	52 barrels per kiln per hour	103 lb. coal per barrel	195,280	15,330	1,225 429	26,800
Cement kiln, 1 kiln per boiler	44 barrels per hour	100 lb. coal per barrel	58,224	4,688	1,485 422	9,300
Beehive coke ovens	2,600 lb. coal coked per hour	—	145,000	10,200	2,158 477	37,200
Gas retorts, 5 per boiler	No record	1,055 lb. coke per hour	17,736	1,339	1,225 425	2,330
Glass furnace	2,380 lb. batch per hour	2,850 lb. coal per hour	43,660	2,860	808 401	2,850
Malleable melting furnace	1·8 tons per hour	830 lb. coal per hour	25,150	2,620	1,650 430	4,700
Puddling furnace	758 lb. per hour	857 lb. coal per hour	20,000	1,880	1,977 501	4,900
O.H. steel furnace	75 tons	550 lb. coal per ton	61,000	5,830	1,436	9,200
O.H. steel furnace	59 tons	47 gallons oil per ton	83,720	4,220	1,065 500	7,600
Steel reheating furnace	No record	112,760 cu. ft. gas per hour	80,757	5,840	1,445 545	1,200

PRINCIPLES OF HEAT RECOVERY IN WASTE HEAT BOILERS

A primary distinction between fuel-fired boilers and waste heat boilers is that the heat transmission in a fuel-fired boiler is largely by radiation, whereas in waste heat boilers flame is absent and heat transmission is essentially by convection or contact. The difference between these two processes will have been appreciated from Chapter VIII, from which it will also be seen that radiation is much the more rapid process.

With gases making contact with the heating surface of ordinary fuel-fired boilers at 1,250° C. (2,280° F.) and leaving at 250° C. (482° F.) some 70 per cent. of the evaporation has been estimated to be due to radiation and the remaining 30 per cent. to convection or contact between the gases and the boiler surface (Gregson, *J. Inst. Fuel*, XI, 85). It is doubtful, however, whether gas radiation was taken into account in this estimate. From the principle that radiation depends upon the difference of the fourth power of the absolute temperature of the source of heat and the cooler surface receiving radiation (see Chapter VIII), it has been calculated (Gregson, loc. cit.) that at 900° C. and 450° C. the percentages of the heat transferred by radiation and by contact are as follows :—

Temperature of incoming gases	1,250° C. 2,280° F.	900° C. 1,652° F.	400° C. 752° F.
Percentage of heat transferred by radiation	70	24·7	3·6
,, ,, ,, ,, contact	30	75·3	96·4

Surfaces merely exposed to gases do little work with low temperature heat. Intimate contact must be obtained so that the surface is scrubbed by the gases. In a modern waste heat boiler the design is such as to give that intimate scrub-

bing contact between gas and heating surface which is essential for reasonably effective heat transfer.

From the laws of conduction and convection discussed in Chapter VIII, it will be recognised that (*a*) the transfer of heat by convection is slow compared with that by radiation, (*b*) with streamline flow there is a layer of flue gas of very low conductivity (see Chapter IX) between the main stream of gas and the boiler plate that causes a serious hindrance to the transmission of heat, and (*c*) the thickness of this film can be decreased by considerably increasing the rate of flow of the gases, and preferably by increasing the rate of flow so much as to render it turbulent. This subject has been fully discussed in Chapters VIII and IX.

From the discussion in Chapter VIII on heat flow parallel to and at right angles to tubes, it would be anticipated that a water-tube boiler with the gases travelling at right angles to the tubes would provide the best arrangement for rapid heat transfer. This type of boiler has been used in the past and is still often used for large installations. There are, however, practical reasons why the fire-tube type of boiler in which the gas is passed through a number of tubes of the order of $1\frac{3}{4}$–$2\frac{1}{2}$ inches diameter, is generally preferred to-day.

The essence of contact heating is to break up the gas stream and force the particles to give up their heat by physical molecular contact. This is effected in various ways according to the basic type or design of waste heat boiler :—

(*a*) In water-tube boilers, by suitable disposition of the tubes and baffling in order to give a long path with constant changes of direction of the gas flow. This applies also to the thimble-tube design of boiler.

(*b*) In fire-tube boilers, by operating at high velocities which give turbulent flow with a high degree of molecular impact on the tube walls.

(*c*) Special fire-tube designs—where instead of depending on turbulent flow created by velocity, contact is obtained by means of retarders or by curvature of the tubes, this curvature having the effect of throwing the gas particles at the tube walls at each change of direction.

DRAUGHT

The waste heat boiler is to be regarded as an ancillary apparatus to the furnace to which it is attached. Consequently in its operation it must create no effect detrimental to the operation of the furnace.

The quantity of gases dealt with by a waste heat boiler is very much larger in proportion to the boiler size and steam output than the quantities in a fuel-fired boiler and therefore the draught required is altogether excessive by the standards of normal boiler operation, being four or five times that usual in a fuel-fired boiler in industrial practice. In addition, the temperature at the base of the chimney is reduced by the abstraction of heat and this also causes greater draught to be required.

These boilers must also impose a high resistance to the flow of gases when the velocity is so far increased as to be turbulent. The drop in pressure is seldom lower than 1 inch w.g. under favourable conditions in fire-tube boilers, or may be as high as 4 inches w.g. under unfavourable conditions. When it is recognised that a chimney will only produce draught of the order of $\frac{1}{2}$–$\frac{3}{4}$ inch w.g. unless it is very tall and with an uneconomically high temperature at its base, it is clear that the installation of a waste heat boiler generally requires chimney draught to be replaced by induced draught from a fan. The water-tube type of boiler absorbs a good deal less pressure and can generally be operated on a chimney. One of the advantages of the installation of waste heat boilers is that induced draught is necessary and the additional draught may improve the working of the furnace.

In considering the question of draught it must not be forgotten that some

furnaces, particularly those fired by coal gas or oil, will remain in their original condition and require much the same amount of draught throughout their working life if the walls are kept gas-tight. Other furnaces, particularly those on solid fuel or on uncleaned producer gas will become partly choked with dust and particularly so if regenerators are installed ; the draught required therefore increases with the age of the furnace. A waste heat boiler operating on a chimney may thus be quite satisfactory when the furnace is new, but unsatisfactory when the furnace is old. The use of induced draught fans overcomes this difficulty.

Alternatively, the improved draught conditions obtained by the use of an induced draught fan attached to a waste heat boiler instead of the natural chimney draught previously used, can so improve the operation of the furnace that the amount of waste heat left for recovery is below that which may be expected from calculations.

In some furnaces mechanical draught improves the efficiency of combustion, but also results in higher rates of combustion, so that the amount of fuel burnt per hour is increased, and so also is the amount of heat available for a waste heat boiler. Each case should be considered on its merits.

There is sometimes an appreciable vertical distance between the point at which the hot products of combustion leave the furnace and the point at which they leave the regenerators. This distance may easily amount to 25 feet in a furnace of quite moderate size, and as the gases may be at an average temperature during their passage from the furnace to the regenerator outlet of over 1,000° C. (1,832° F.) it will be realised that there exists in this part of the system a very appreciable chimney effect, which the flue gases in the chimney have to overcome in addition to the normal frictional resistance of the plant. If for any reason the available draught at the regenerator outlet is reduced, it will be necessary to reduce the rate at which gas and air are supplied to the furnace and the furnace will necessarily be slowed down.

AIR INLEAKAGE

The quantity of heat that will be recovered in a waste heat boiler depends on the initial temperature of the gases. If this initial temperature is reduced, the rate of heat transfer will be diminished. The size of a boiler is calculated in the first place on a given inlet temperature and a given outlet temperature. If the gases are diluted with air, they may still carry the same total quantity of sensible heat through the boiler, but since the temperature drop will be less the efficiency of the boiler must diminish.

On account of the high draught needed, a waste heat boiler is under a considerable suction in comparison with the surrounding atmosphere. Trouble is encountered in the operation of water-tube waste heat boilers due to air infiltration through the comparatively extensive surfaces of external brickwork associated with this type of boiler. Conditions in this respect would be greatly improved by the practice of casing the boiler brickwork, but as the boiler steam output is already naturally low in relation to the size and cost of the boiler (in comparison with normal boiler practice) this arrangement would undoubtedly affect the economics of the installation.

The ingress of cold air :—

(*a*) Reduces the temperature of the gases, dilutes them, and reduces the potential evaporation owing to the greater volume at a given outlet temperature to atmosphere.
(*b*) Increases the load on the induced draught fan, often to the extent of causing it to fail to give the requisite pull at the furnace.
(*c*) Causes boiler leakage due to local cooling effect.

(*d*) If unburnt constituents have passed over in the waste gas flue—having failed to ignite before entering the boiler—there is the possibility of local gas explosions in the boiler setting, provided the ignition temperature can be reached at a point of air ingress.

The effect of air infiltration depends upon where the air enters. If the additional excess air is admitted to the furnace and thus heated to a high temperature the result is an increase in the amount of heat recovered in the waste heat boiler for a defined inlet temperature as shown in Fig. 165.

On the other hand, if the air enters after the furnace it serves only to dilute the flue gases and to reduce their temperature without causing the temperature

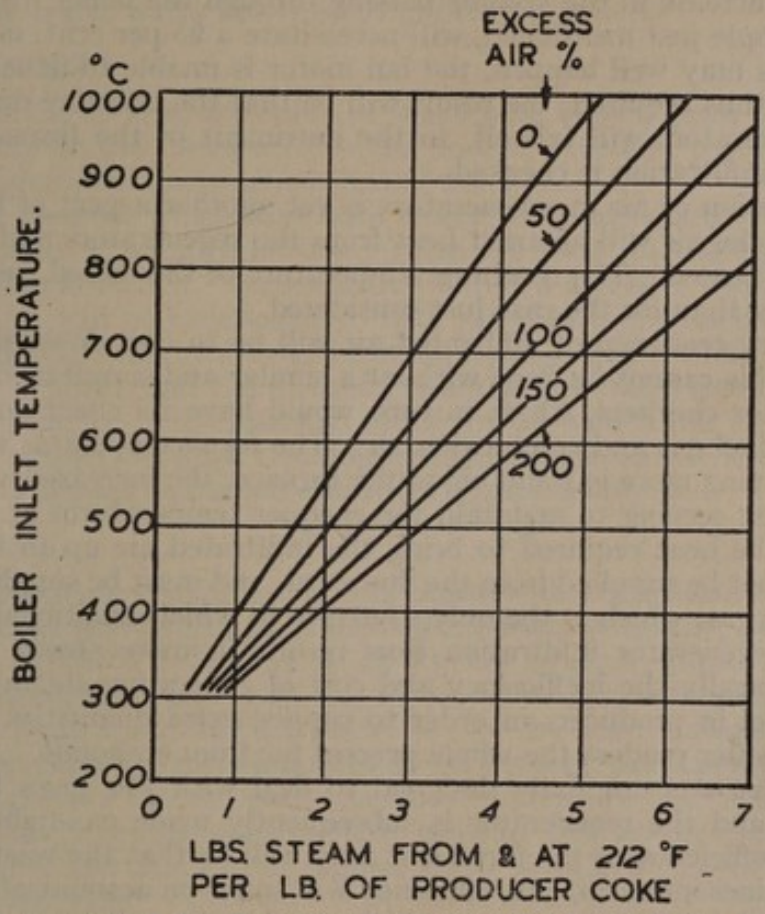

FIG. 165. Heat content of the flue gases derived from a furnace heated by a coke-fired producer.

(Reproduced by permission of the publishers (Eyre and Spottiswoode) from "Modern Gasworks Practice," by Alwyn Meade.)

head to perform any useful work. The total sensible heat of the air and gas remains unchanged by dilution at this stage.

Thus in a waste heat boiler operating with an outlet temperature of 230° C. (446° F.) the addition, by infiltration, of 200 cubic feet of air at atmospheric temperature to 800 cubic feet of furnace gases at 500° C. (932° F.) will result in a volume of 1,000 cubic feet of mixed gases at a temperature of approximately 400° C. (752° F.), all volumes being taken at atmospheric temperatures. This process will clearly be one of unmixed loss from the point of view of the steam-generating unit. It is true that the total quantity of sensible heat is unaltered, but as 25 per cent. more gas has to be rejected to the chimney, at approximately the original outlet temperature, 230° C. (440° F.) greater heat losses are thereby incurred. so that steam production is reduced.

The effect of infiltration of air into the flue can be clearly shown by an

exaggerated example. If every 800 cubic feet of furnace gases at 500° C. (932° F.) were mixed with 1,000 cubic feet of infiltrated cold air at atmospheric temperature, the temperature of the resulting 1,800 cubic feet of mixed gases would be about 220°–230° C. (428°–446° F.), and the amount of steam raised would be practically nil.

Not only is " flue infiltration " entirely detrimental to the boiler in reduced steam output, but the fan power necessary to handle the increased volumes through the boiler will increase. For any boiler once installed, the draught loss (and hence the fan pressure head) will vary as the square of the rate of gas flow, and as the fan power is the product of the fan pressure head and the volume moved, the power required will vary as the cube of the gas flow. A 25 per cent. increase in the volume passing through the boiler, therefore, as in the first example just mentioned, will necessitate a 95 per cent. increase in fan power. If, as may well happen, the fan motor is unable to develop the additional power thus required, the result will be that the effective draught on the furnace regenerators will fall off, to the detriment of the furnace operation, until the air infiltration is checked.

The infiltration of air in regenerators is yet another aspect of this problem, because here the air will abstract heat from the regenerators and will thus be itself further heated. The resulting temperature of the mixed gas and air will thus be higher than for the case just considered.

The primary tendency of infiltrated air will be to cool down the hot waste gases. But this cannot be done without a similar and simultaneous cooling of the regenerator checkers, which in turn would have its effect on the preheat given to the fuel gas and combustion air. The furnace operator would correct this by supplying more gas and air to the furnace, the increased volume of hot flue gases then serving to maintain the chequer temperatures at the outgoing end. Thus the heat required to bring the infiltrated air up to flue gas temperature cannot be supplied from the flue gases, and must be supplied indirectly from the fuel gas, which is the only source from which additional heat can be supplied. Regenerator infiltration thus results in more steam being raised, although naturally the inefficiency and cost of generating steam by gasifying additional fuel in producers in order to supply extra quantities of low grade heat to the boiler renders the whole process far from economic.

If the boiler was originally designed to deal with the gases from a leaky regenerator, and the regenerator is subsequently made gas-tight in order to improve the efficiency of the furnace it may well be that the waste heat boiler will become uneconomical. Its efficiency will suffer on account of the fact that all radiation and convection losses, which are already rather important in view of the comparatively small steam output from a fairly bulky boiler, will have to be subtracted from a reduced heat input, although themselves unchanged. The condition of the furnace and its ancillary apparatus must therefore be carefully considered before a waste heat boiler is installed.

FEED WATER TREATMENT FOR WASTE HEAT BOILERS

The working pressure of waste heat boilers is comparatively low being rarely higher than about 160 lb. per square inch. Obviously the lower the pressure the lower will be the temperature in the water and the more heat can be abstracted from the gases (Chapter VII).

The difficulties of securing an adequate rate of heat transmission by convection and conduction only are such that resistance to the flow of heat should be eliminated as far as possible. It is therefore necessary for the efficient operation of waste heat boilers that the feed water should be softened as completely as is practicable in order that the formation of scale and sludge in the boiler and its deposition on the heating surface shall be reduced to a minimum.

The hardness of the feed water should not exceed 3 degrees, and should preferably be below this, especially as it is highly desirable to operate the boiler continuously for the duration of the furnace campaign without the formation of any appreciable amount of scale or sludge on the heating surfaces.

The conditions of operation of a normal waste heat boiler are not at all severe from the point of view of boiler water condition, since a large volume of boiler water is associated with a comparatively low rate of evaporation. In a typical waste heat boiler the water storage may be equivalent to several hours' steaming, as compared with a modern high-rated water-tube boiler, which may only contain the equivalent of twenty minutes' evaporation. It will usually be found sufficient to blow down at such a rate as will maintain the concentration of salts in the boiler water below 1,000 grains per gallon.

TUBE CLEANING

It is important to keep the tubes of waste heat boilers clean partly in order to prevent any further resistance to the flow of heat and partly because the deposition of dust will cause serious loss of draught.

Among the methods which have been used are the introduction of wood chips or coke breeze into the gas inlet box ; these are swept through the boiler and exert a scouring influence. The difficulty is that if a tube is badly blocked the velocity of flow will be insufficient to take the scouring material through and thus the condition of that tube will be made worse ; only the less contaminated tubes will be cleaned, and the method cannot be generally recommended.

The usual practice is to clean the tubes with a wire brush (or preferably a scraper head cleaner) through cleaning holes in the gas outlet box of the boiler. This operation should be carried out at least once per week.

SUPERHEATERS

Superheaters are in general use in connection with waste heat boilers. Superheaters are of necessity situated in the boiler intake chambers ; for waste gases up to 700° C. (1,290° F.) they are set in the flow of gases, but above this figure they must be suitably protected from the gas stream flow and arranged for radiant heat absorption, for the protection of the tube material. For these higher temperatures the rate of flow of steam through the superheater must be sufficient to prevent undue metal surface temperature and with a correctly designed assembly arranged for radiant heating, this arrangement is entirely effective in an intake chamber as high as 1,100° C. (2,015° F.) (Gregson, *Trans. Inst. Gas E.*, **84,** 952).

LIMITATION OF EXIT TEMPERATURE OF GASES

If a boiler is working at 160 lb. per square inch the water will be at a temperature of 371° F. and it is clear that the gases must leave the boiler at a temperature equal to this figure plus an amount depending on the efficiency of heat abstraction in the boiler.

The difference between the boiler water temperature and the gas outlet temperature will usually be of the order of 80° F., so that the gas exit temperature from a boiler working at 160 lb. pressure will be about 450° F. As the gas inlet temperature may be only 1,000° F., it will be apparent that only 55 per cent. of the heat contained in the gas at the inlet can possibly be converted into steam in the boiler, even if all radiation and convection losses were to be eliminated. In normal circumstances it is undesirable to allow the temperature of the exit gases to fall below 450° F. (238° C.) as experience shows that at lower temperatures the fan casing, outlet duct and neighbouring steelwork may be seriously and rapidly corroded, depending on the SO_2 and SO_3 content of the gases.

ECONOMISERS

The use of economisers is subject to the limitations mentioned under the heading " Limitation of Exit Temperature of Gases."

Economisers are used in installations with cement and lime kilns and metal-refining furnaces where the draught required to operate the kiln or furnace properly is low and where there is a reasonably uniform supply of flue gases. With open hearth furnaces where the required draught is high, and where there are often variations in the amount of flue gases from a given furnace, economisers are seldom, if ever, used.

BOILER EFFICIENCY

The efficiency of a waste heat boiler can be expressed in several ways. Thus in a valuable report (" Waste Heat Boilers in Open Hearth Practice ") issued in 1935 by the Iron and Steel Institute in the course of which an analysis was made of the working of a waste heat boiler installation, the performance of a boiler operating on a 50-ton open hearth furnace was given as in Table 96. This boiler was too large for the quantity of gases passing through it and better results can be obtained.

TABLE 96

Size of boiler	18 ft. × 9 ft. diameter.
Heating surface	3,593 sq. ft. mainly composed of 2 in. O/D. fire-tubes.
Gas temperatures :—	
(*a*) Inlet	455° C. (851° F.)
(*b*) Outlet	222° C. (431° F.)
Total heat in gas :—	
(*a*) Inlet, above 60° F.	115·03 therms per hour.
(*b*) Outlet, between boiler water temperature 183° C. (362° F.) and 60° F.	58·14 " " "
Total heat given to steam	35·29 " " "
Heat lost in blow-down	0·19 " " "
Other losses from boiler	13·30 " " "
Total heat given up by gases to boiler	48·78 " " "

Evaporation	Actual lb.	" Equivalent " lb.
Evaporation per hour	3,208	3,740
Evaporation per lb. of coal	1·75	2·05
Evaporation per ton of ingots	821	958

While for practical and economic estimates the data provided in this table are used, the boiler efficiency is a factor of importance. The overall efficiency is the ratio of the effective output to the total heat input, expressed as the percentage :—

$$\frac{\text{Heat absorbed by steam}}{\text{Total heat in gas at inlet}} \times 100 \text{ per cent.}$$

Substituting in this fraction the appropriate figures from Table 96, the efficiency is found to be (35·29 × 100/115·03 =) 30·7 per cent., which is very low in comparison with normal boiler efficiencies. Actually the efficiency of heat transfer in this boiler was extremely good, as is shown by the fact that the gases

were cooled to a temperature only 40° C. above that of the water in the boiler, and by the ratio :—

$$\frac{\text{Heat absorbed by steam}}{\text{Heat given up by gases in boiler}} = \frac{35{\cdot}29}{48{\cdot}78} = 72{\cdot}3 \text{ per cent.}$$

The last figure, however, flatters the boiler unduly, since all heat losses to the stack are ignored.

The coldest heat-absorbing surface must be at a temperature at least as high as that of the water in the boiler, which will be the saturation temperature corresponding to the boiler pressure. The most efficient boiler possible, perfectly insulated and working with perfect heat transfer across the heating

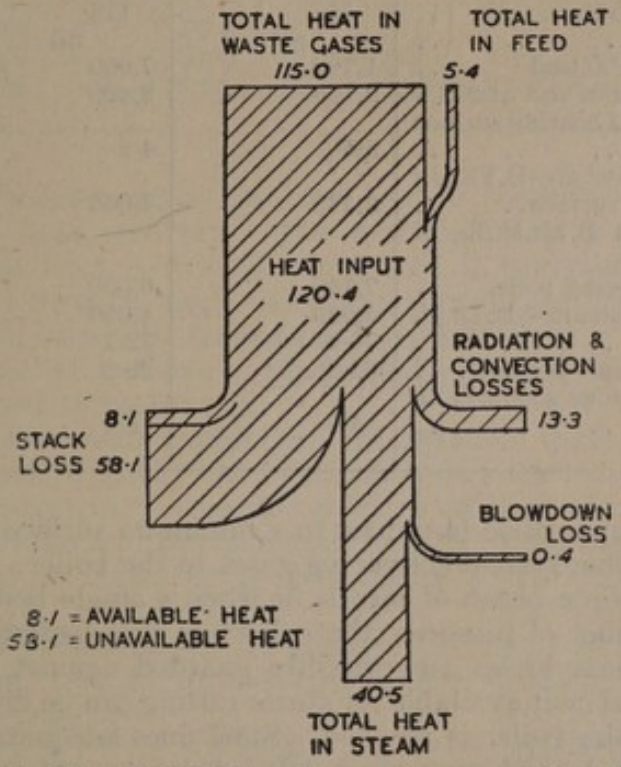

FIG. 166. Heat distribution diagram for a waste heat boiler.

(From the Second Report of the Iron and Steel Industrial Research Council, "Waste Heat Boilers in Open Hearth Practice.")

surfaces, could only reduce the gases to this temperature. The true indication of the performance of the boiler under trial, therefore, is the ratio of the overall efficiency (30·7 per cent.) to the efficiency of a perfect boiler working under the limitation described above. The efficiency of a perfect boiler is the ratio :—

$$\frac{\text{Available heat in gas above boiler water temperature}}{\text{Total heat in gas at inlet}}$$

$$= \frac{115{\cdot}03 - 58{\cdot}14}{115{\cdot}03} = \frac{56{\cdot}89}{115{\cdot}03} = 49{\cdot}4 \text{ per cent.}$$

The results were expressed in the publication in question in the form of a diagram, Fig. 166.

As an illustration of the operating results of waste heat boilers, Table 97, taken from gas works practice (Gregson, *Trans. Inst. Gas E.*, **84,** 934) may be cited. The heat recovered as steam was measured in these tests, but in order to show the order of magnitude of the losses, the radiation loss has been assumed to be 5 per cent. of the input heat, thus enabling the heat lost in the stack gases to be calculated by difference. The quantity of gas passing through the boiler was not measured, but the quantity of coke fed to the producers was known,

and all the gases resulting from burning the whole of the producer gas derived from this coke passed through the waste heat boiler.

TABLE 97

Works :—	D	E	G
Producer fuel (coke) used per hour—lb.	403	1,860	1,300
Temperature of gases—° C. :—			
Boiler inlet	780 (1,434° F.)	800 (1,472° F.)	560 (1,040° F.)
Boiler outlet	230 (446° F.)	220 (428° F.)	220 (428° F.)
Boiler pressure—lb./sq. in.	115	118	98
Superheat—° F.	nil	nil	200
Evaporation/hr. ; lb. actual ..	1,790	7,000	3,750
Evaporation/hr. ; from and at 212°F.	2,100	8,400	4,500
Evaporation/sq. ft. of heating surface from and at 212° F.	3·5	4·2	2·25
Mean heat transmission—B.Th.U./hr./sq. ft. of heating surface.. ..	3,380	4,055	2,175
Heat distribution in B.Th.U./lb. of producer fuel :—			
Heat in gases entering boiler ..	7,140	6,000	5,500
Heat recovered as steam—B.Th.U.	5,025	4,360	3,340
per cent.	70·4	72·7	61·0
Heat in stack gases—per cent. ..	24·6	22·3	34·0
Radiation, etc., losses assumed to be—per cent.	5·0	5·0	5·0

It is of great importance to reduce to a minimum surface losses, both in the boiler and in the flue system conveying gases to the boiler. Frequently, as for example in a multiple bench of retorts or when a single boiler operates on the gases from a number of furnaces, the collecting flue system is of considerable length. Unless heat losses are carefully guarded against, a very large percentage of the total heat available for steam raising can be dissipated from these gas ducts before the boiler is reached. Steel flues adequately lined with hot-face insulating brick, and properly hooded dampers, are essential.

It will be appreciated that with low grade gases, rating per unit of heating surface is low compared with a fuel-fired boiler under equivalent gas weight flow conditions, hence for a given output the proportional loss from the boiler is high compared with ordinary boiler practice. To counteract this, it is essential to design waste heat boilers to be as compact as possible, in order to reduce surface to the minimum.

Another source of loss in flue systems which are carried below ground is that due to water, either water in the ground cooling the flues by contact, or water soaking into the flues.

EFFECT OF INTERMITTENT OPERATION

If the flow of flue gases through the boiler fluctuates considerably, the rate of steam generation will be subject to similar fluctuations. Steam from other boilers on the range may pass into the waste heat boiler during these periods, so that the boiler will be condensing from other sources steam that has been made with the expenditure of coal.

ECONOMICS OF WASTE HEAT BOILERS

The efficiency of waste heat recovery is relatively low on account of the comparatively small temperature drop through the boiler. The following

figures which are based on the assumption that the various losses excluding flue gas loss are 5 per cent. of the inlet heat, and that the outlet temperature is 450° F., indicate the general trend of recovery and its dependence on the inlet temperature. All figures are based on 1,000 lb. of flue gases having a specific heat of 0·25 B.Th.U. per lb. (Table 98).

TABLE 98

Inlet temp. ° F.	Outlet temp. ° F.	Heat in gas therms	Losses therms	Heat in chimney gas therms	Heat-recovered therms
2,000	450	4·85	0·24	1·12	3·49
1,000	450	2·35	0·12	1·12	1·11
600	450	1·35	0·07	1·12	0·16

The waste heat boiler must be considered on its merits and not as an alternative method of heat recovery. The heat recovery in regenerators and recuperators, as has been mentioned previously, is governed by the needs of the furnace. If the thermal conditions in the furnace are not such as demand that the air and gas should be preheated, the installation of regenerators may not be justified, and when there is adequate demand for steam a waste heat boiler should be used in preference.

On the other hand, when regenerators are installed, there may be sufficient heat left in the gases to justify the installation of a waste heat boiler as well. Again, the regenerators may of necessity be so efficient that the temperature of the outlet gases is too near the temperature of the water in the boiler to make a waste heat boiler an economic proposition as shown from the figures just calculated.

In planning new installations, the relative advantages of waste heat steam and steam generated in ordinary boilers should be considered from the point of view of fuel economy.

The fuel-saving effect of a waste heat boiler is sometimes expressed as a percentage of the fuel charged to producers used for supplying a furnace. This is unsatisfactory because the cost of producer coal differs considerably from that of boiler slack, and the monetary saving will be less than the saving in coal quantity would indicate. There are also other costs to be taken into account both on the waste heat boiler and on the fuel-fired boiler, including that of the fan used with the waste heat installation.

All relevant factors can be taken into account by determining as accurately as possible the average steam output from the boiler, and then ascertaining all costs—electric power, cleaning maintenance, repairs and inspection, capital costs and depreciation. These costs must be set against the cost of generating this amount of steam in the fuel-fired boilers.

Where the operation of the factory is largely dependent upon waste heat boilers, special precautions should be taken to ensure that the induced draught fans can be run from alternative sources, as otherwise the stoppage of fans would result in a rapid falling-off of steam, which would bring the plant to a standstill within a very short period.

SELECTION OF BOILER

It is very difficult to draw sharp lines limiting the purposes for which each type of waste heat boiler is best suited. According to Gregson (*J. Inst. Fuel*, XI, 89), these are approximately as follows :—

(*a*) Water-tube designs for handling large volumes of high temperature gases, particularly where high steam pressures are required with stand-by fuel firing. These designs are not so successful where a heavy furnace draught is superposed on the circuit, owing to potential inleakage at the casings, and if gas explosions are likely to occur in the boiler setting—for example, on the change-over of the reversing valves of the regenerators of an open hearth furnace, the brickset boiler is not suitable.

(*b*) Fire-tube designs form the standard waste heat boiler of the steel and gas industries in this country, and also for furnaces demanding the handling of medium volumes of medium to low temperature gases with considerable suction or, in water gas plants, considerable pressure. The fire-tube design positively avoids inleakage and consequently troubles therefrom.

(*c*) Thimble-tube designs—essentially compact vertical units which are very well suited for internal combustion engine exhaust gases.

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CHAPTER XXI

ELECTRIC FURNACES

Electrical terms—Conversion of electricity to heat—Arc heating—Resistance heating—Types of electric furnace—Load characteristics of electric furnaces—General operation of the various types—Electric supply and furnace operation.

BEFORE considering electric furnaces in detail some notes on electrical engineering terms and a brief theoretical review of the conversion of electricity to heat will be given.

ELECTRICAL TERMS

There are two types of electric current—direct and alternating. As their names imply, the one is constant in magnitude and direction, the other varies with cyclic regularity. Either may be shown in the form of a graph upon which direct current will be a straight line, whereas alternating current is represented by a curve of which the form is dependent on a number of factors, but for all practical purposes, can be taken as a pure sine wave (Fig. 167).

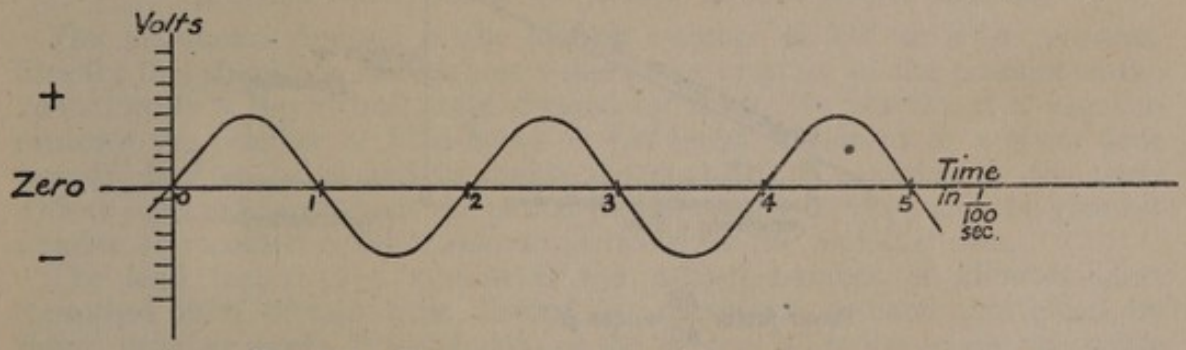


FIG. 167. Sine curve of voltage (alternating current).

The frequency of an alternating current is defined as the number of complete waves which occur in one second and has been standardised in Great Britain at 50.

In a direct current circuit a voltage is absorbed in overcoming the resistance of a conductor. If V (volts) be the voltage, I (amps) the current, and R (ohms) the resistance of the conductor, the relation between the three is given by Ohm's Law :—

$$V = IR.$$

The power expended in overcoming resistance is voltage $\times$ current, that is, VI, but from the relationship given above ($V = IR$) the power, P, becomes $P = I^2R$, and this is dissipated in heat. The unit of power is the watt.

$$\text{watts} = \text{volts} \times \text{amps.}$$

Since an alternating current fluctuates in magnitude from instant to instant, some convention is necessary to determine its numerical value. The convention adopted is to give the virtual value of the current, that is to say, to give the value of an equivalent steady current that would produce the same heating effect as the alternating current in one period, when passing through a given resistance. Since the heating effect is dependent on the square of the current, it can be shown mathematically that the virtual value is the square root of the average value of all the squares of the current over one period—known as the "root-mean-square". This expression is usually shortened to "R.M.S. current." Similarly the virtual voltage is the "R.M.S. voltage."

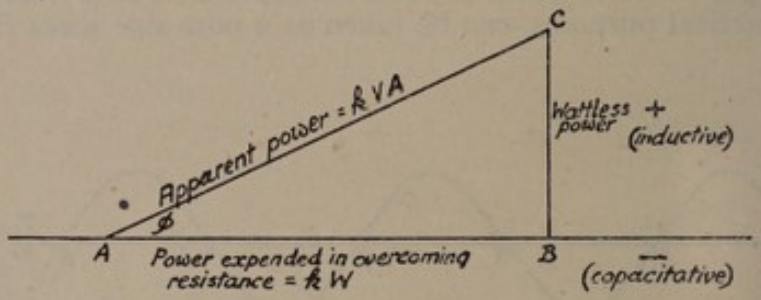
Measuring instruments operate on this principle and the scale reading auto-

matically gives the root-mean-square of the current. It is this virtual value, or instrument reading, that is always referred to in practice.

When dealing with direct current circuits Ohm's Law holds good. With alternating current, however, two other factors enter into consideration—inductance and capacity. Inductance is caused by fluctuations of current in a conductor inducing counter-voltages and therefore counter-currents, while capacity is the effect produced by a condenser.

In a circuit having negligible resistance and high inductance the maximum counter-voltage occurs with the maximum rate of change of the current. Since the resistance is negligible the voltage will be zero when the current is a maximum (assuming sine waves). It is conventional to say that under these conditions the current is lagging on the impressed voltage.

Where the capacity is in effect the only factor in the circuit, the maximum rate of change of voltage produces the maximum charging current, and we then have the converse of the inductive circuit and, again conventionally, the current is said to lead over the impressed voltage.



$$\text{Power factor } \frac{AB}{AC} = \cos\phi$$

Apparent power in circuit $= AC$

Effective or true power $= AC \times$ power factor

$= AC \cos\phi$

FIG. 168. Power vectors (alternating current).

For either capacity or inductance, consideration of sine waves will show that in every alternate quarter-period an amount of power is taken out of the circuit which is returned to the circuit during the other quarter-periods; in other words, no effective power is being supplied. For this reason the power taken by an inductance or a condenser is referred to as "wattless power."

Depicting the conditions by a diagram, the power in a pure resistance may be represented by a horizontal straight line of a definite magnitude which would be termed mathematically the "resistance vector" (Fig. 168). The wattless power referred to above can be represented by a vertical line drawn at one end of this line. It is conventional to consider those lines drawn above the resistance vector as positive or inductive and those below as negative or capacitive. Where both inductance and capacity exist the net wattless power is the arithmetical difference between the two and this gives the length and direction of the vertical line required. Completing the triangle the hypotenuse is the "apparent" circuit power.

The ratio of the true power (absorbed in the resistance part of the circuit) to the apparent power is the power factor and is mathematically the cosine of the angle between the resistance power vector and the total circuit power vector (abbreviated to cos ϕ).

Power in a direct current circuit, as has been stated above, is volts × amps. The power in an alternating current circuit is also volts × amps, but this is the apparent circuit power (AC, Fig. 168) and the effective or true power is : volts × amps × power factor (AB, Fig. 168). Since volts × amps produces an inconveniently large product it is usual to divide this by 1,000. In a direct current circuit or in the resistance component of an alternating current circuit AB (Fig. 168) gives kilowatts (abbreviated to kW). In a similar way the apparent power in an alternating current circuit is called kilovolt-amperes (abbreviated to kVA).

The kVA given by the product virtual volts × virtual amps for one line of a circuit must be increased by multiplying by $\sqrt{3}$ to determine the total kVA in a three-phase circuit, which is the commonest alternating current arrangement.

The total energy is kilowatts × time, and the usual time period for the unit of energy is one hour, resulting in kW-hours. On rare occasions the term kVA-hours is also met.

The kilowatt-hour (kWh) is the Board of Trade unit of electrical energy. It is the work done between two points in a circuit when

$$\text{pressure (volts)} \times \text{current (amps)} \times \text{time (hours)} = 1{,}000.$$

The maximum demand is the highest number of kW or kVA recorded. Strictly this should be the highest value at any instant of the product of the virtual volts × the virtual amps divided by 1,000. In practice it is usual to measure the number of kVA-hours or kW-hours consumed in a given time (usually half an hour) and to divide the quantity so recorded by the time. This is repeated throughout the period (week, month, or year) and the greatest number so recorded is the maximum demand for the period chosen.

The load factor of a system is the ratio :—number of kilowatt-hours consumed in a certain time divided by maximum demand multiplied by time ; in other words, it is the ratio of the average kilowatts to the maximum kilowatts in a specific period. Where the period chosen is, say, the hours of operation of a particular machine, it is usual to speak of the ratio as the running hour load factor. Where the ratio is for the whole year it is usual to speak of it as the annual load factor.

CONVERSION OF ELECTRICITY TO HEAT

Broadly, there are two ways of converting electricity into heat :—

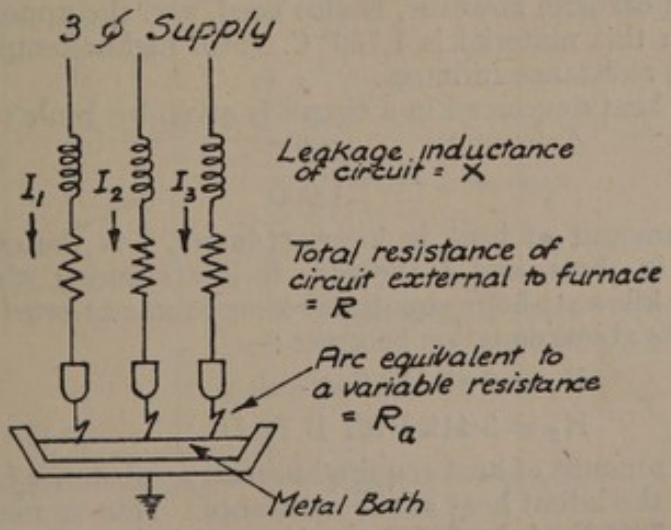


FIG. 169. Equivalent circuit for 3-phase arc furnace.

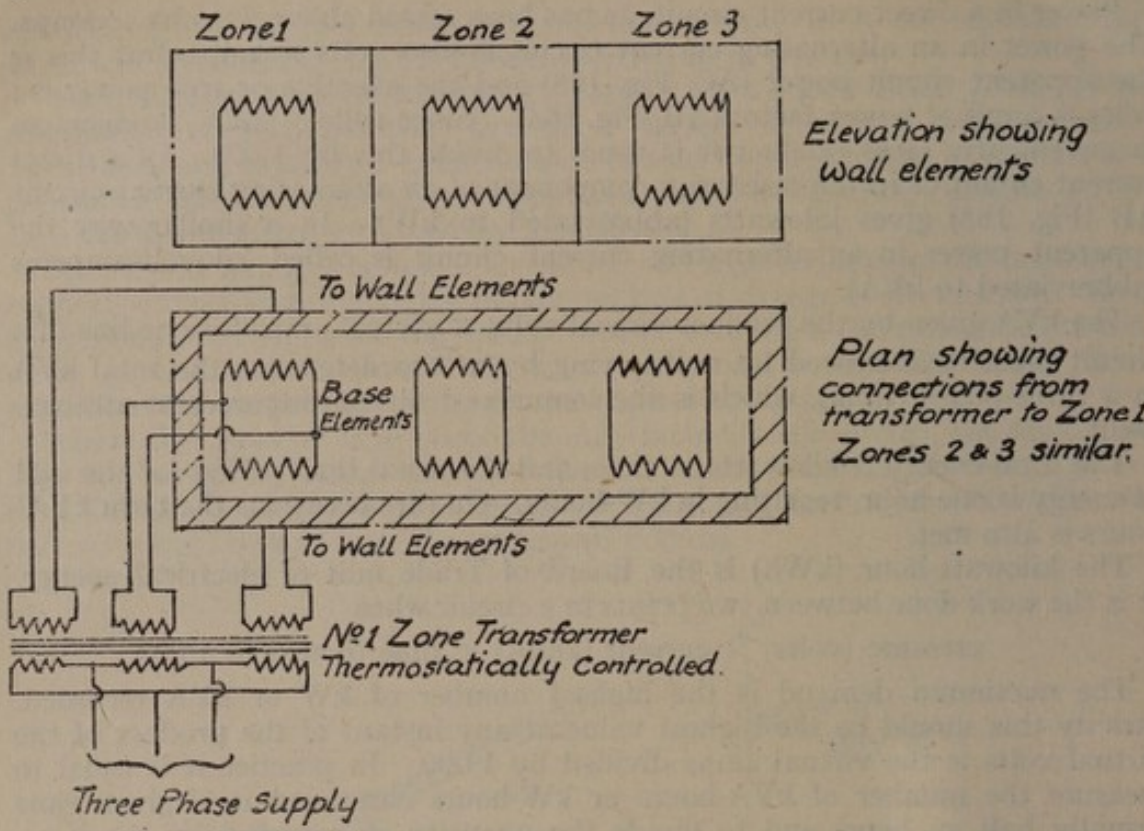


FIG. 170. Electrical diagram for 3-zone resistance furnace with " on-off " thermostatic control.

(1) By means of the electric arc which produces a concentrated source of heat at a high temperature (about 3,000° C.). The current flows through an indeterminate conductor, consisting of the vapour formed between the electrodes (Fig. 169).
(2) By means of a resistance. The current flows in a definite conductor and the heat is evenly spaced over each unit of length, assuming a constant cross-section and resistivity of the conductor (Fig. 170).

Theoretically this latter form of heating is suitable for any temperature. In practice, however, it is limited by the maximum temperature that the resistance element will stand without disintegration. Generally speaking, resistance elements are of nickel chromium alloy and the limiting temperature is about 1,000° C. Silicon carbide, however, is also used, and the approximate limit of temperature with this material is 1,750° C. Still higher temperatures can be used with carbon resistance furnaces.

The amount of heat developed in a circuit is given by Joule's Law, and is :—

$$H = \frac{I^2RT}{1{,}000}$$

where H is the amount of heat in kilowatt-hours, I is the current in amps, R the resistance in ohms, and T the time in hours during which the current flows. Since one kilowatt-hour equals 860 kilogramme calories or 3,412 British Thermal Units, the above equation becomes :—

$$H_1 = 0{\cdot}86\ I^2RT \text{ kg-cal.}$$

or

$$H_2 = 3{\cdot}412\ I^2RT \text{ B.Th.U.}$$

The theoretical amount of heat required to melt a substance is the sum of the sensible heat and the latent heat of the substance. Thus to melt 1 ton of steel at 1,500° C. (= 2,730° F.) from 60° F. the theoretical heat consumption is :—

Sensible heat 0·162 × 2,670 × 2,240	=	970,000 B.Th.U.
Latent heat	=	195,000 ,,
Total heat	=	1,165,000 ,,

The equivalent electricity consumption is therefore :—

$$\frac{1{,}165{,}000}{3{,}412} = 342 \text{ kW-hours.}$$

A similar calculation may be made for any other material. In practice, of course, electric heating is not 100 per cent. efficient ; typical efficiencies that should be obtained in practice are given later in the chapter.

ARC HEATING

For operating reasons the electric arc furnace has a certain amount of inductance introduced into the circuit. There is, in addition, the natural resistance of the conductors, together with the resistance of the arc itself. From what has been said previously, it will be understood that the effective heat is derived from the arc resistance and that the resistance of the leads and of the inductance are pure losses.

When V = voltage of one line above earth.
X = resistance due to the inductance, in ohms.
R = line resistance, in ohms.
z = vector sum of X and R, i.e. $z = \sqrt{X^2 + R^2}$.
R_a = resistance of arc, in ohms.
Z = vector sum of inductive resistance X and arc + circuit resistance $R_a + R$, i.e.

$$Z = \sqrt{X^2 + (R_a + R)^2}.$$

$\cos \phi$ = power factor.
I = current in line.
P_a = arc power (cf. Fig. 169),

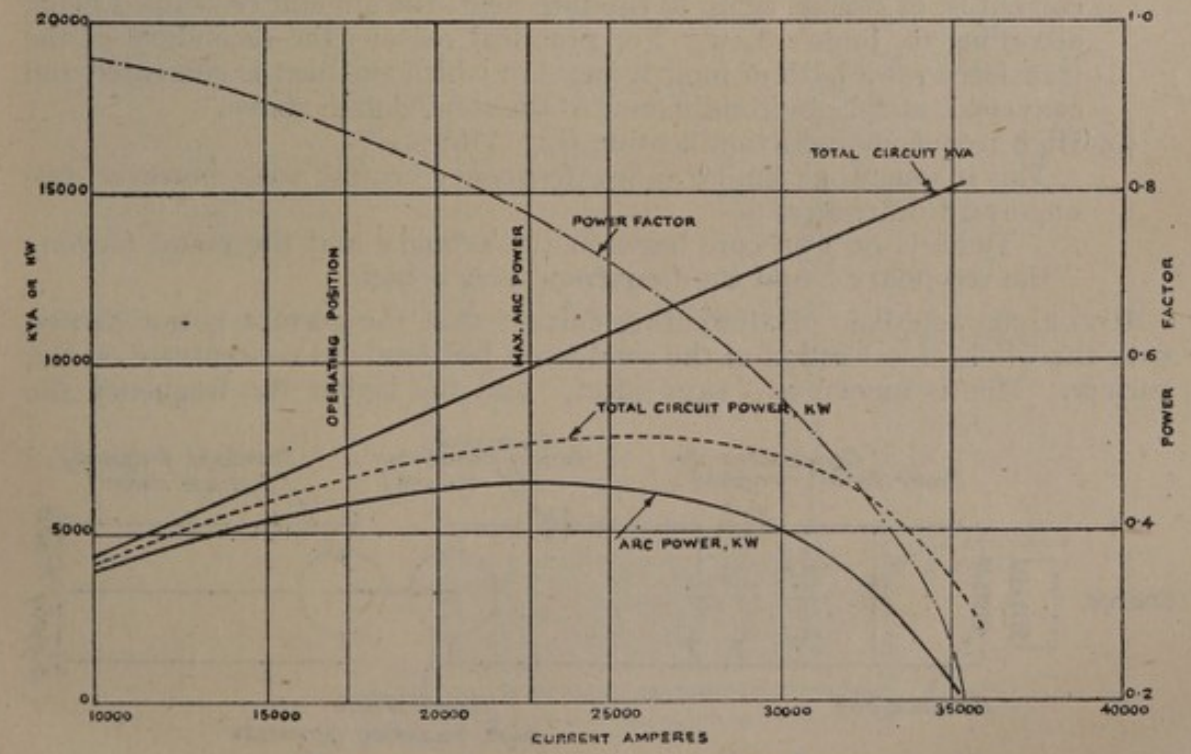


FIG. 171. Characteristic curve of arc furnace.

it can be shown that in a three-phase arc maximum power occurs when

$$\cos\phi = \frac{R_a + R}{(R^2_a + 2R_aR + z^2)} = \sqrt{\frac{(z + R)}{(2z)}}.$$

In practice the power factor resulting from operating under maximum arc power conditions is poor and therefore it is customary to run at somewhat lower power and improve the power factor (Fig. 171).

If the arc is run for a certain number of hours with a known input the energy consumption is known approximately and hence the heat produced is known.

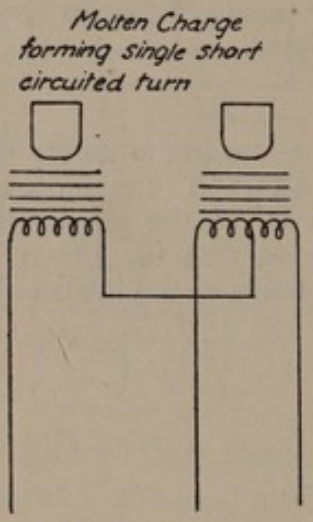


FIG. 172. Electrical diagram for 2-phase normal frequency operating from 3-phase supply.

RESISTANCE HEATING

Although it is customary to consider the resistance method of heating as a single type, for convenience it will be subdivided into three sections here :—

(1) Heat produced in a direct connected resistance element :—

This is the true resistance furnace and of course the heat developed is that given previously by Joule's Law. It is immaterial whether the current flows in a solid or liquid conductor. An example of a liquid conductor is found in the salt bath furnace.

(2) Low frequency induction heating (Fig. 172).

In effect this method of heating consists in supplying the power to the primary of a transformer with a magnetic core, the secondary of which is formed by the charge to be melted. There are therefore two stages in the arrangement—the transformation of a fairly high primary voltage to a low secondary voltage, this secondary voltage producing a high current in the material. The effect of the high current is, of course, again to produce heat, the amount developed being according to Joule's Law. For practical reasons the secondary of the transformer is a bath of molten metal in which the heat is generated and conveyed, mainly by conduction, to the solid charge above.

(3) High frequency induction heating (Fig. 173) :—

This is somewhat similar to low frequency heating with, however, two important differences :—

There is no iron core between the primary and the metal forming the secondary ; and the frequency used is high.

It is a characteristic of alternating current that the current is not carried over the whole cross-section of the conductor, but tends to concentrate on the surface. This is known as " skin effect," and the higher the frequency the

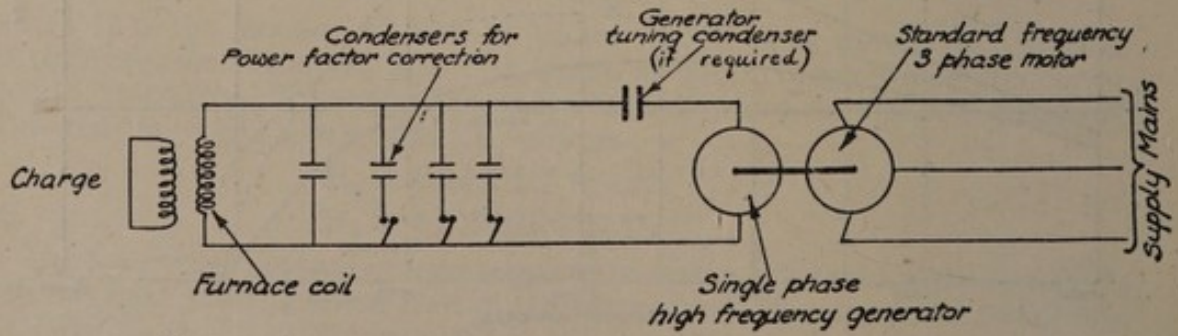


FIG.73. Electrical diagram for high frequency induction furnace.

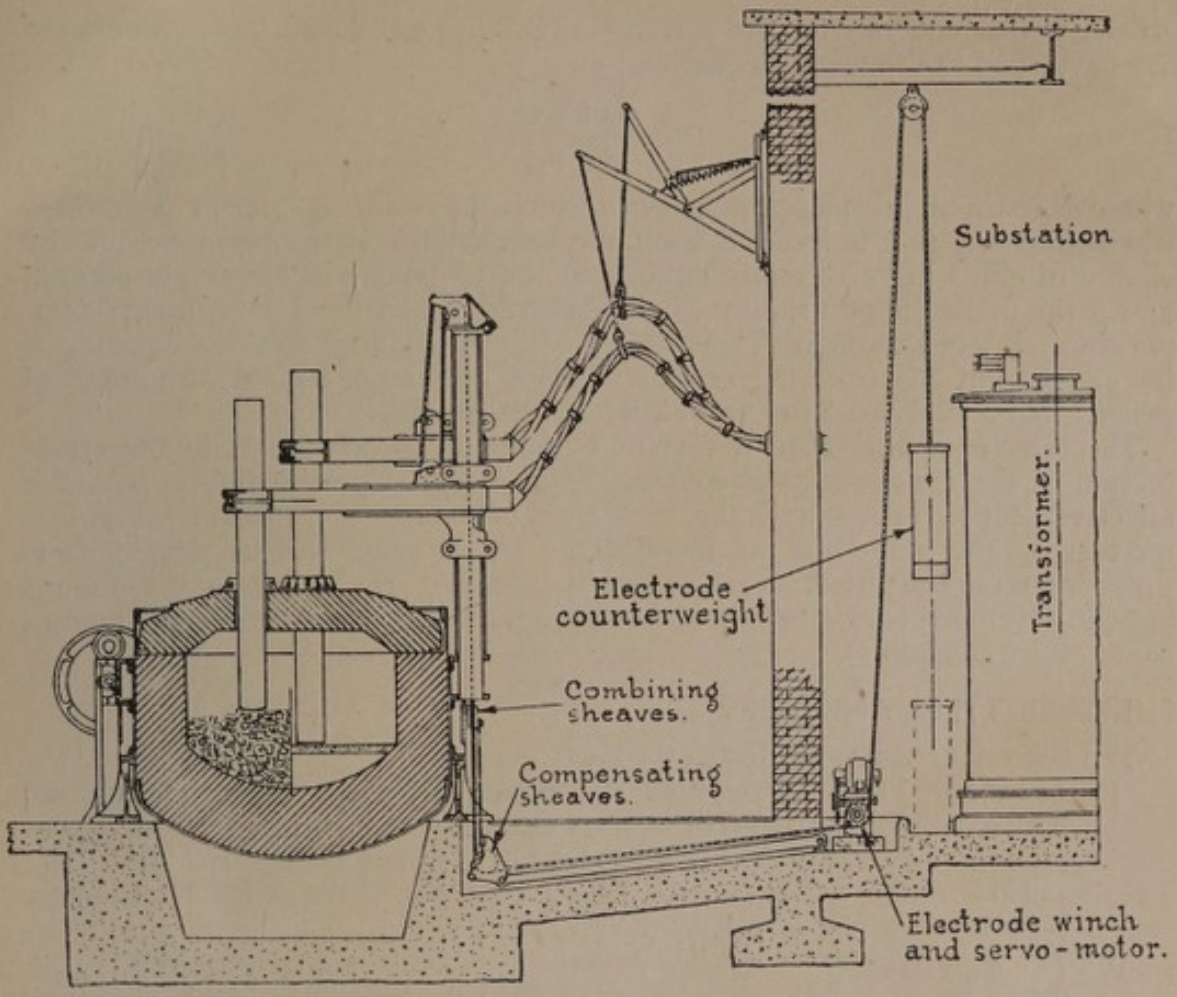


FIG. 174. Arc furnace.

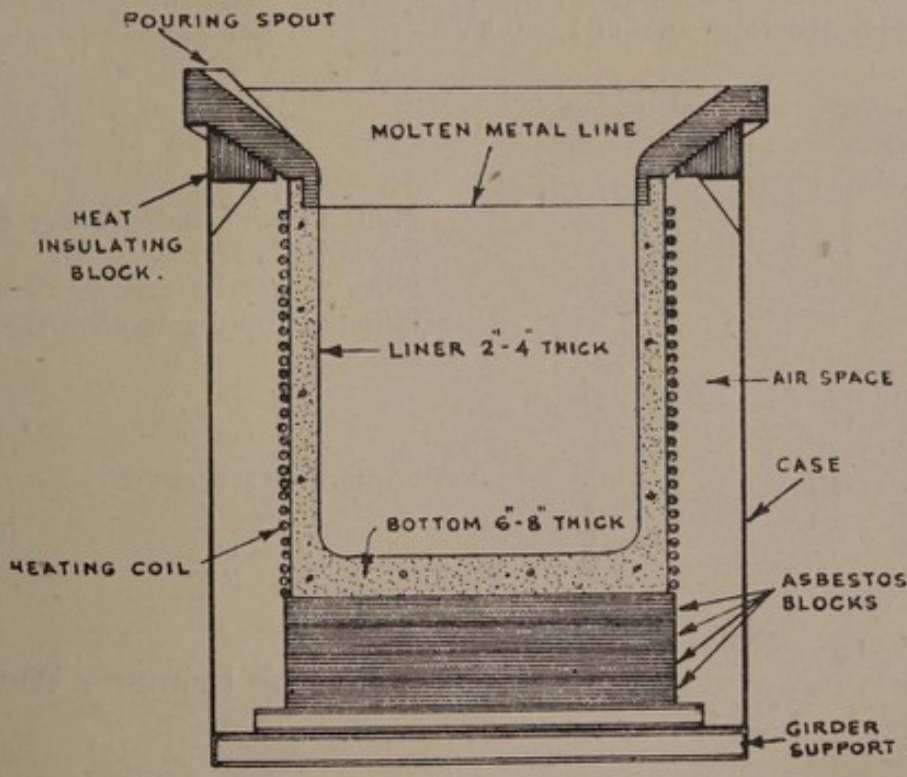


FIG. 175. Section through high frequency induction furnace.

greater is the tendency for the current to remain near the surface. Steinmetz has developed the following equation :—

$$P = \frac{5{,}030\sqrt{r}}{\sqrt{F\mu}}$$

where P is the depth in centimetres to which the resulting current penetrates when magnetic flux is induced from the primary : r is the resistance of the charge in ohms per centimetre cube, F is the frequency in cycles per second, and μ the magnetic permeability of the charge, which for iron at higher temperatures is equal to unity. Thus for a given material if the frequency is increased from 50 cycles per second to 2,500 cycles per second, the depth of penetration is decreased to approximately one-seventh.

The resistance offered to a current is inversely proportional to the cross-section of the conductor. Therefore, if the depth of penetration is less, the effective section is less and the resistance offered to the flow of current is increased. But according to Joule's Law the amount of heat developed is proportional to the resistance, therefore by increasing the frequency the amount of heat developed is also increased for a given current and true cross-section of conductor (the charge).

FURNACE TYPES AND USES

There are four types of electric furnace :—

(i) *The Arc.* The heat required is produced by means of an electric arc struck between each of three electrodes and the charge. Arc furnaces are used for melting and refining ferrous metals (cf. Figs. 169 and 174).

They range in size up to 30–40 tons per charge with transformer capacities up to 6,000–8,000 kVA.

(ii) *The high frequency induction furnace* in which the heat required is produced by a combination of induced currents and skin effect. This type is used for melting some alloy steels, and furnaces are not usually much above 2 tons in capacity, though there are 5-ton furnaces in use with motor generator sets of 1,500 kVA (cf. Figs. 173 and 175).

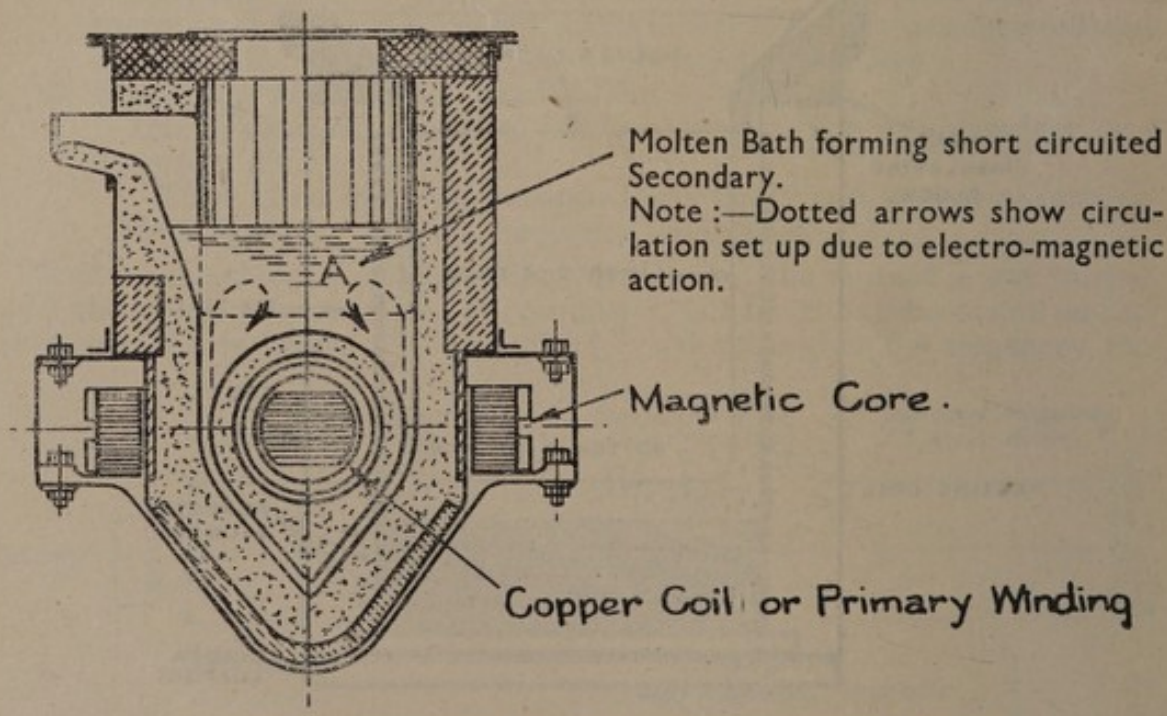


FIG. 176. Section through normal frequency induction furnace.

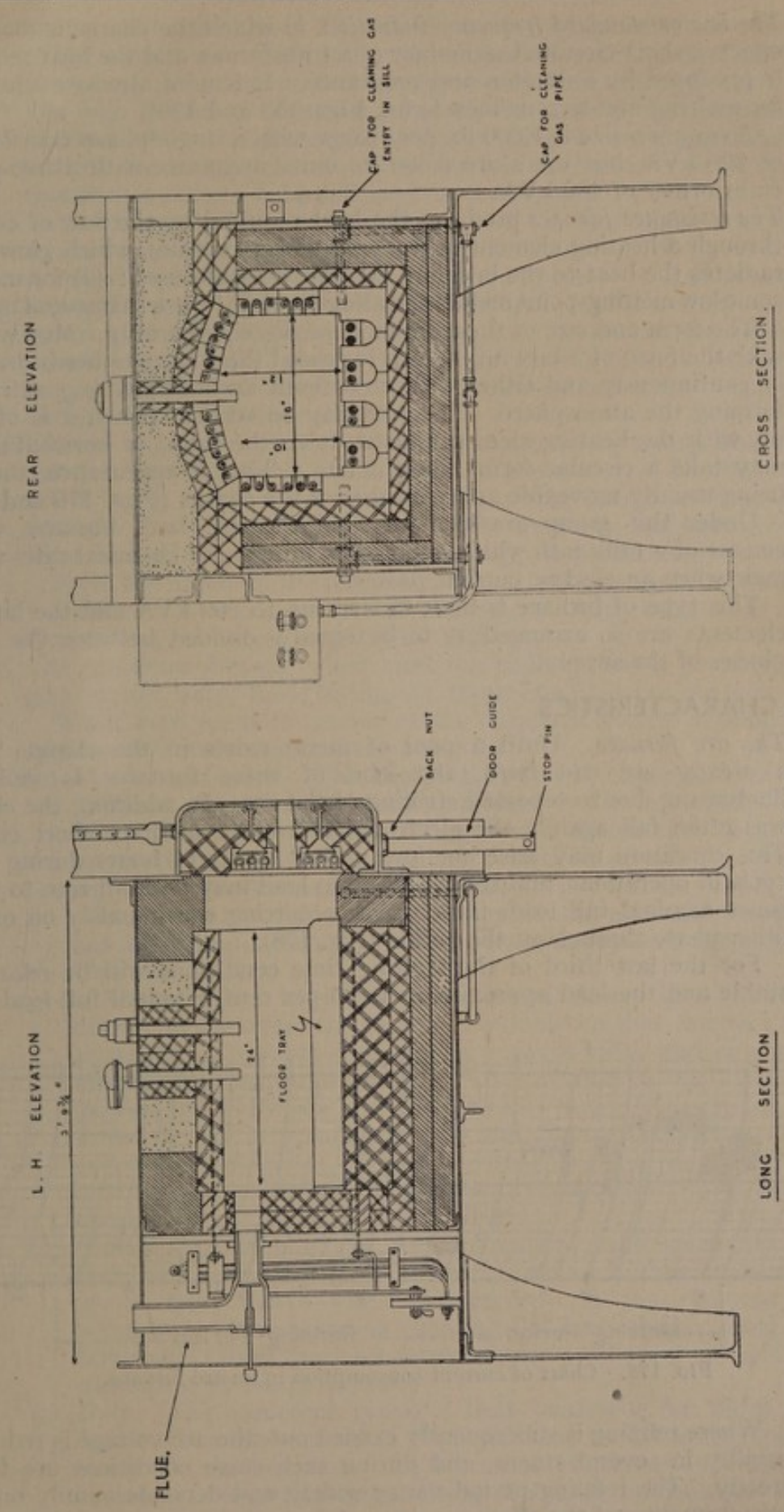


FIG. 177. Resistance furnace.

(iii) *The low or standard frequency induction*, in which the charge is made, in effect, a short circuited secondary of a transformer and the heat required is produced by induction and resistance. It is used almost exclusively for melting non-ferrous metals (cf. Figs. 172 and 176).

A common size is 1,000 lb. per charge with a single-phase transformer of 100 kVA, but there are 3,500 lb. units in service with a two-phase transformer of 400 kVA.

(iv) *The resistance furnace* produces the heat required by the flow of current through a heating element (commonly nickel-chrome), which glows and radiates the heat to the interior. It has two main uses : (*a*) for melting some low melting-point metals and alloys, and (*b*) heat treatment of metals.

The form and size of these furnaces varies considerably. Mostly they take the form of a box into which is passed the charge either in batches or continuously and either with or without an inert gas (e.g. nitrogen) forming the atmosphere. The box may be vertical (when it is often a pit with the heating elements fitted down the sides) or horizontal ; or may take a circular form, when it has a " bell " appearance, the bell being usually moveable and placed over the charge (Figs. 170 and 177).

Under this group are also included the salt bath furnaces which consist of a tank into which is placed a mixture of chemical salts which melt when an electric current passes.

This type of furnace is built in sizes up to 800 kVA and the heating elements are so arranged as to be equally divided between the three phases of the supply.

LOAD CHARACTERISTICS

(i) *The arc furnace.* Until a pool of metal exists in the charge, when a steady arc can burn, the load of these furnaces is violently fluctuating due to repeated striking of the arc. In addition, the charge will often fall against the electrodes and cause a direct short circuit. This condition may arise for any period up to 1½ hours during each cycle of operations, and the variation in load may be from zero to three times nominal full load, the changes occurring continuously on one or other phase throughout the period (Fig. 178).

For the last third of the melting time conditions will be relatively stable and the load approximate to 85 per cent. nominal full load.

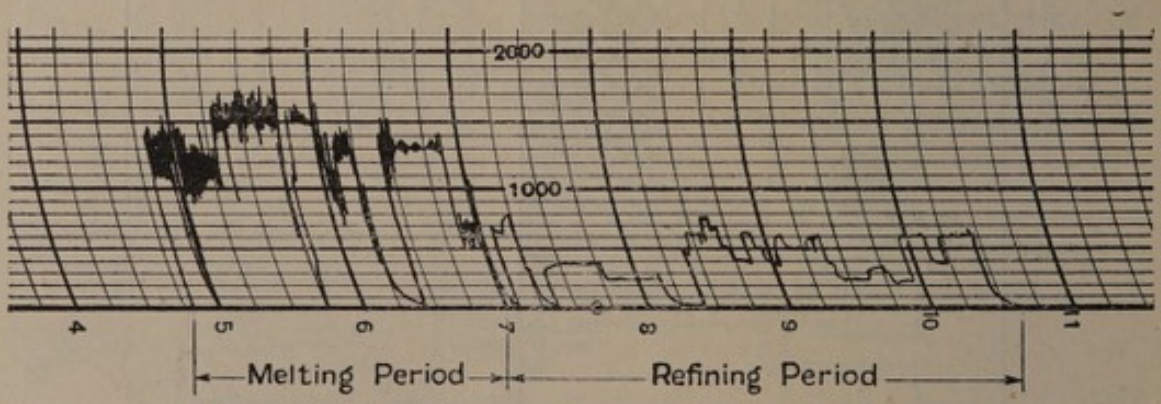


FIG. 178. Chart of current consumption in an arc furnace.

Where refining is subsequently carried out, the arc voltage is reduced, usually in several stages, and during each stage conditions are fairly steady. The refining period varies widely and depends mainly on the initial quality of the charge and the degree of accuracy required in the

final material analysis. The load is now roughly from one-quarter full at the start to one-tenth full at the end of the refining time which may be as long as five hours, but usually does not exceed 2½ hours.

(ii) *The high frequency induction furnace.* Because of the frequency used (1,200–2,500 cycles per second), this requires a motor generator set to provide the supply, the motor of which is of the three-phase type, and therefore ensures a balanced load on the supply.

The demand of the furnace has a well-marked trend. Assuming a one-hour melting time, the load falls steadily from 80 per cent. when switched on first, to 50 per cent. in eight minutes. It then rises steadily during the next eight minutes to 100 per cent., and thereafter remains at between 90 and 100 per cent. Throughout the period the power factor is bad, and to avoid an excessively large generator, banks of condensers require to be brought into circuit, the maximum correction being required during the latter half of the period.

(iii) *The low frequency induction furnace* has a practically steady load throughout the melting period, which is normally one hour. The large air gap which exists between the primary coil and the charge or short circuited secondary makes the power factor of the unit low, and again correction must be applied. In this case, however, it is permanently in circuit.

From the nature of the furnace (single or two-phase) a Scott connected transformer is used, the smaller furnaces being connected in pairs thereto and the larger singly (Fig. 172).

(iv) *The resistance furnace*, when used for melting, has a perfectly steady good power factor load, lasting for about one hour per charge.

When used for heat treatment the electrical load is dependent on a variety of factors among them being :—

(*a*) *Method of Control.* The cheapest and most common form of temperature (and thus electricity) control is the so-called "on-off." When this is used, after the furnace and charge have been brought up to temperature, the supply is switched off until a pyrometer registers a predetermined permissible drop when the supply is again switched on and left on until the maximum allowable temperature has again been reached. This sequence is repeated throughout the treatment cycle.

Alternatively, when the furnace and charge have reached the correct temperature the supply is reduced so that the heat input is as nearly as possible equal to the furnace losses and only rarely is the full load again switched on. A variation of this method is to provide an induction regulator which varies the applied voltage smoothly and in accordance with the furnace demands. Although expensive in first cost, this latter form of control has the advantage of subjecting neither the resistance elements nor the supply system to sudden changes of load.

(*b*) *Method of Charging.* In batch charging each group of articles is treated separately and the charge has each time to be brought up to temperature when the process on the previous charge is finished. In continuous charging the fresh charge is fed in continuously ; thus the load is more nearly constant as cold charge succeeds cold charge, steadily.

(*c*) *Duty.* The numerous types of heat treatment for which electric furnaces are used are all of them different in the temperatures and times that they require. The five usual treatments are : hardening, annealing, tempering, normalising, and case hardening. To these must be added solution and ageing for aluminium and "electron"

metals. The temperatures and times vary between 1,350° C. for five minutes for hardening tool steels to 180° C. for two hours for tempering ordinary steel, and include a steady temperature of 500° C. for 96 hours for nitriding.

Again in the ceramic industry there are three treatments for earthenware and china : biscuit firing for about 72 hours at 1,200°–1,400° C., gloss firing for 24 hours at 1,150° C., and decoration firing, when painted, for 15 hours at 700°–900° C.

Finally, steatite treating is a single pass continuous process lasting 24 hours, and comprising three zones of temperature—0°–1,000° C., 1,000°–1,400° C., and 1,400°–700° C.

It is clear therefore that with this variety of conditions to be met no accurate generalisations can be made relating to the electrical load on a resistance furnace.

GENERAL OPERATION

Some precautions which should be taken in the operation of each of the above types of furnace with a view to heat economy will now be given.

ARC FURNACES

One of the largest sources of loss in an arc furnace originates from intermittent operation. The large mass of material going to make up the furnace radiates heat quickly when left idle, and this heat has to be restored when the furnace is re-started. An arc furnace should therefore run continuously and the charging time should be reduced to a minimum. It is preferable to use a smaller furnace (if ingot sizes allow of it) and keep it in continuous operation 168 hours a week rather than a slightly larger furnace for 120 hours.

The charging time can be much reduced by having a movable roof. The swinging of the roof allows of practically the whole charge being dropped in from one or two baskets in ten minutes instead of being fed by hand or mechanical charger from the side in ½–¾ hour. If a movable roof is not fitted and conditions do not allow of one being installed, mechanical charging is usually preferable.

Each charge, however inserted, should be ready for immediate feed as soon as the previous charge is teemed, and any patching to the liner that may be required is complete. The necessary material for liner and roof repairs should be immediately to hand and care taken that the right man for this work is standing by to commence operations as soon as the furnace is available.

Next in importance is the upkeep of the furnace, particularly at doors and at the electrode holder. At both these points care should be taken to see that the joints are tight.

It should be borne in mind that reduction of radiation loss by the above methods not only reduces heat consumption, but prolongs the life of the liner and reduces the electrode consumption.

The water cooling of the electrode holders should also be watched and care taken to see that the water supply does not foul the tubes as this implies greater power for circulating the necessary quantity of water, increased heat loss due to easier conduction along the electrode, and higher electrode consumption.

On the electrical side special care should be given to the maintenance of the oil switch, as this operates many times a day—the duty being far more onerous than for ordinary industrial purposes ; and, apart from the potential risk to life of a faulty switch, damage thereto may hold up production for an appreciable time, which again results in heat losses during the idling period. The frequency with which switch contacts and oil need attention will depend on the design of the switch and the severity of service ; this can only be judged from experience.

A good modern furnace should have approximately the following heat balance when melting :—

	Per cent.
To the charge	62 (376 units per ton when melting steel).
Electrical losses	9
Cooling water	4
Radiation, door loss and conduction	25

The electrode consumption should be of the order of 14 lb. per ton (graphite) or 28–30 lb. per ton (amorphous carbon).

HIGH FREQUENCY INDUCTION FURNACES

The small body carrying the charge makes these furnaces excellent for intermittent operation, but as the current may be converted by a motor generator set, great care should be taken to shut off the set whenever the furnace is not required for any considerable period, and thus avoid heavy no-load losses. It is important to switch the condensers off when the furnace is not in use.

Where successive melts are carried out, the time between charges should not exceed 15–20 minutes. If the converter house is accessible the set can well be shut down, but if (as so often happens) it is some way away, then it is probably not justifiable. For delays of 30 minutes or over, however, the set should always be shut down.

The motor of the transforming set should be of the synchronous type and full advantage should be taken of the capabilities of such a motor for system power factor improvement which will reduce appreciably the electricity losses in the works.

It will usually be better to have two equal-sized furnace bodies operating off one motor generator and the supply fed to either by a simple change-over switch. It is then possible to keep the power on as long as melting has to be carried out, as teeming from one body and recharging takes place while the other is in service. At the same time, extra care can be given to ensuring that the liner is properly patched after each heat, a very necessary precaution with the thin walls that are here used.

The heat balance of such a furnace should be approximately as follows :—

To the charge	59 per cent.
Cooling water	3 ,, ,,
Electrical losses	22 ,, ,,
Radiation, etc.	16 ,, ,,

Radiation losses from these furnaces can be minimised by having a cover over the body during melting and by ensuring that the charge is not of such a size as to foul this cover.

LOW FREQUENCY INDUCTION FURNACES

It has already been noted that these furnaces work in pairs, or more rarely singly, off a Scott connected transformer. Care should always be taken to see that the installation is so arranged and operated that the transformer is properly utilised. It is even more important to ensure continuity of operation. A certain amount of molten metal must be left in the base of the furnace after each charge has been poured to act as the secondary of the transformer for the next charge. (When starting up, molten metal must be poured in initially and this metal must be kept molten.) Between 10 and 20 per cent. of the transformer rating is required to ensure that the metal does not solidify, and this represents a serious heat loss which can only be avoided by keeping the furnace in constant use. Where long shut-downs are necessary the furnace should be

completely emptied rather than the supply kept on to keep the priming charge molten.

Radiation, particularly from the top of the furnace, is again a serious source of loss, but as these furnaces are chiefly used for copper, brass, aluminium and its alloys and zinc, it is not easy to find a suitable cover ; covers can be made, however, and the cost is warranted when the reduction in losses is taken into account.

The heat balance per melt should be of the following order :—

	Per cent.
To the charge	65
Electrical losses	5
Radiation, etc.	25–35
Standby losses	Depend on time out of service.

RESISTANCE FURNACES

For melting service these furnaces are fairly good. They have low electrical losses. The radiation losses are, however, high on account of the fact that melting by such a furnace takes place by conduction and convection and is slow ; hence the losses continue over a long period. In total the electricity consumption per ton is approximately the same for either a resistance or low frequency induction furnace.

The heat or electricity consumption of a heat treatment furnace is made up of :—

(i) Heat imparted to charge.
(ii) Heat imparted to carrier, support, etc., for charge.
(iii) Radiation losses, and
(iv) Heat lost between charges.

Headings (ii), (iii) and (iv) are the losses and must be kept as low as possible.

The minimum of metal in the support or carrier should be aimed at. Baskets with as coarse a mesh as possible are best for small parts such as springs, nuts and bolts, etc. Wires and strip in coils can be dropped on to a tray with a vertical guide to keep them in position. Shafts and forgings treated in a horizontal furnace are usually sufficiently rigid to be carried on a light bogie at either end. Vertical (pit) furnaces are excellent for gun barrels and long shafts as movable adjustable radial guides can be fitted at intervals and only the minimum for the safety of the piece being treated need be inserted. For continuous furnaces the makers will have provided trucks or wire band and little alteration can be made to this, but care can be taken to see that the charge is spread evenly over the surface and that maximum weights are put on each unit of surface.

Radiation losses from the body of the furnace can only be affected by changes in internal temperature. Care should be taken therefore to ensure that the furnace is never run at a temperature higher than that for which it is designed. Again, if the furnace is used for a lower temperature than the design value, care should be taken to ensure that facilities exist for obtaining an even distribution of heat. Thus a furnace running normally at 500° C. relies on natural heat distribution, but at 200°–250° C. (tempering of some steels) fans on the roof are essential to maintain a proper and economic distribution of heat. A source of considerable radiation loss is the door, and this area, particularly in a continuous furnace, must be kept as small as possible and carefully curtained. In a batch furnace it should not be necessary to open the door between charges, and each charge should be ready for immediate insertion after the previous one is finished—provided of course that the treatment process allows of this.

Some form of recuperative heat recovery is also advantageous in certain conditions, particularly in long tunnel furnaces, the air from the hot section passing down an adjacent preheating section and then back to the main section again. The temperature gradient across the outside wall of the recuperative section being lower than in the hot section, the radiation losses per square foot for equal insulation will be less.

Heat loss between charges may be the heaviest item in some processes. It is considered necessary when annealing certain low carbon steels to allow charge and furnace to cool together to a certain temperature before removing the charge. The losses here involved cannot be avoided. The majority of annealing and normalising processes, however, call for the cooling of the pieces in still air and in such circumstances the next charge should be immediately ready when the previous one has been removed. It is desirable that if only a few charges are to be put through, these should be allowed to accumulate and the furnace kept in steady operation for some time rather than pass a few charges per day. In many works the habit has grown up of heating up a furnace at times of low electrical load on the works—thereby avoiding the payment of the fixed kilowatt or kVA charge. This may be uneconomical from the heat aspect, however, particularly in the conditions just envisaged, as the furnace may not be required immediately it is hot or after a few charges have been treated it may be allowed to cool for the remainder of the day and then heated up again at night. As stated, continuous operation should be aimed at once the furnace is hot. Occasionally some saving of heat may be made by inserting small parts among some large parts and preheating these to a certain temperature before placing them in another furnace where they are brought up to a still higher temperature. (Such practice is essential in the case of tool steels which are preheated to 850° C. before being hardened at about 1,350° C.)

In all furnace operations heat can be conserved and the process improved by the maximum use of automatic devices.

Electrode control in arc furnaces should never be by hand and the control impulse should preferably be a combination of voltage and current control, rather than current control alone. Electrical control of the hoisting gear is more common than hydraulic, but a good design of either type is satisfactory provided always that if water servomotors are used the water supply is absolutely clean and will not foul pipes or valves.

The condensers for correcting the power factor of high frequency induction furnaces should also be taken in or out of circuit automatically by means of a contact-making power factor indicator and never by hand.

The cooling water supply both to the electrode bosses of an arc furnace and through the coil of a high frequency furnace should have at least automatic indication and an audible alarm operating if the flow ceases. An automatic change-over to an alternative supply in the event of one source of supply failing is an added advantage, as is also an excess temperature alarm.

Heat treatment resistance furnaces must always have thermocouples inserted at various points in the furnace to ensure that the correct temperature for the treatment is maintained. These couples regulate the electricity supply to the particular section of the furnace that they control and, from the heat economy aspect, the more control points there are the better.

All furnaces should also be provided on the instrument panel with a chart recorder showing the instantaneous load and with a kilowatt-hour meter which should be read and logged at each stage of the operations in the furnace.

ELECTRICITY SUPPLY AND FURNACE OPERATION

When considering electricity supply a distinction must be drawn between public supply and private plant. The former source of power is, with much

justification, regarded as limited only by the capacity of the supply cables and the rating of the transformers both of which can be increased relatively quickly. Generally too, high overloads do not affect either the voltage or the frequency, and therefore the supply to the remainder of the works is not disturbed.

Private plant, on the other hand, is limited in most cases by the boiler capacity and even where ample spare plant exists the fuel that is available for burning in industrial stations may be such as to de-rate the boilers quite considerably. Furthermore, heavy overloads on private plant may reduce, albeit temporarily, voltage and frequency. Reduced frequency implies reduced speed and reduced voltage means reduced torque (reduction as the square of the voltage) on motors.

The industrialist with private plant has therefore frequently to adapt his furnace practice to his plant capacity and in so doing has sometimes improved his heat balance—though sometimes the result may be the reverse.

The most troublesome load is that of the arc furnace during the melting period. With several furnaces installed much effort should be directed to staggering the melting period. This avoids the incidence of a high maximum demand which in turn means more economic running of the station supplying the load, and particularly may avoid the necessity for forcing boilers. The peak demand may also be reduced by instructing the operatives not to shorten the melting time by running on a high voltage tap before an arc can be properly maintained. It is seldom that the melting time can be appreciably shortened while the total consumption is actually increased by this action, owing to the very heavy demand each time the arc is struck after being extinguished.

Where melting only is carried out in the furnaces, one group (consisting of one or more units) should be allowed to complete about 50 per cent. of the melt before the next is brought in—though this action should not be carried so far as to involve delay in any group and bring about the worse condition of heavy radiation losses.

If refining as well as melting is being carried out, more still can be done to avoid heavy loadings, provided some knowledge of the length of the refining period is available. In any event careful planning by the authorities can make a big reduction in the calls made on the supply.

Both the high and normal frequency furnaces have a regular load curve, but their cycle times are short and (except for the standby losses inherent in the low frequency furnace) they are particularly suitable for fitting in " valleys " in load curves and thus ensuring better boiler loading. Where the output requirements of the factory allow of intermittent operation of these furnaces this procedure should be adopted; and as a general principle low frequency furnaces should be emptied if the standby period is likely to be of some hours duration, but the supply should be maintained on the furnaces coupled to one transformer so that there is a supply of molten metal immediately available for the other furnaces when these are re-started.

Resistance furnaces are also a good load on the supply, but the characteristics of the duty that they have to perform limit their flexibility from an electricity utilisation aspect. They should be heated up wherever possible at times of low load elsewhere, provided that they are not then required to stand idle for some time until they receive their first charge.

CHAPTER XXII

REFRACTORY AND INSULATING MATERIALS

Refractories :—Influence of refractories on fuel economy—Types of refractories—Raw materials—Manufacture—Constitution of refractories—The chemical and physical properties of refractories—The action of slag, gases and vapours—Jointing cements—Selection and testing.

Insulating materials :—Classification—Conductivity—Insulating firebricks—Lagging and lagging materials.

Construction :—Methods of building refractory structures and insulating walls.

REFRACTORY MATERIALS

THE subject of refractory materials is so vast that it would be impossible here to do justice to its many aspects. There are interesting problems concerning the use, manufacture, and behaviour of even the commonest refractory material which are still the subject of research. There are equally many problems that may be solved by the use of less common refractories for special purposes. Considerations of space make it necessary here to concentrate on those properties of refractory materials which affect every-day practice and which in particular influence the efficiency of fuel utilisation.

The influence of refractories on fuel economy arises from many causes (Chapter XVIII) among which may be mentioned :—

(1) The stability of the furnace structure is obviously an essential factor in the efficient performance of industrial operations.

(2) The maintenance of a sound construction free from gaps and cracks is essential. Cracks in the brickwork may result in short circuiting of flue gases or leakage of live gas into air or air into gas with disastrous results on the uniformity of temperature in the furnace, upon the amount of fuel used, and on the brickwork.

(3) The action of slag on furnace brickwork may be such as to destroy the brickwork and in extreme circumstances may result in holes through furnace walls into the heating flues.

(4) Correct thermal gradient is also important ; in many kiln operations bricks wear down to a limiting thickness, no matter how thick they may be originally, and the effects of insulation may be important.

(5) In continuously operated furnaces the longer the periods between letting down for repairs the less is the expenditure of fuel on heating up the furnace from the cold. On the other hand, planned maintenance may be the more economical even though it may mean doing repairs whether wanted or not.

(6) In many industrial operations heat has to be transmitted through a wall of refractory material to the furnace, while in all furnaces a great deal of heat is stored in the bricks. The thermal properties of the bricks are therefore of considerable interest to the furnace operator and affect the quantity of fuel used.

TYPES OF REFRACTORIES

The more common types of refractory are defined as follows :—

Silica brick contains 92 per cent. or more of silica, almost entirely free or uncombined.

Siliceous or semi-silica brick contains by analysis from 75–92 per cent. of silica ; upwards of half of this is free silica, the remainder being in combination as aluminium silicate.

Firebrick contains by analysis less than 75 per cent. of silica and less than 38 per cent. of alumina. Up to 40 per cent. of the silica is free, and the alumina is in combination with the remainder of the silica as aluminium silicate.

Aluminous firebrick contains by analysis over 38 per cent. and up to 45 per cent. of alumina, almost entirely in combination with silica as aluminium silicate. The more aluminous bricks may contain free alumina.

The accepted names of other types of refractory give a sufficient indication of their general composition, e.g. magnesite brick, chrome brick, chrome magnesite brick, silicon carbide brick, bauxite, sillimanite.

Typical compositions of refractory materials are given in Table 99.

RAW MATERIALS

In Great Britain there are abundant raw materials for the more common refractories—firebricks and silica bricks. Fireclays are associated with the coal measures throughout the country; these clays vary in composition from siliceous plastic materials to the hard bauxitic shales of Ayrshire. Thus clays are available for all purposes. Ganister and quartzite rocks, consisting of 97 per cent. or more of silica, are also widely scattered in the Carboniferous series; these rocks are worked for brickmaking in the Sheffield district, North and South Wales, Durham and Scotland.

Magnesite as such is non-existent in this country, but a process has been developed during the past seven or eight years for the synthetic production of magnesia; this process is employed to supply some of the magnesite used for brickmaking. Dolomite is plentiful in this country and is used in the making and fettling of furnace hearths and, when partially or wholly stabilised, as bricks for furnace construction and maintenance.

Chrome ore occurs only in minor quantities in Great Britain, the chief home source being the island of Uist in the Shetlands. Raw materials for special refractories such as sillimanite and bauxite bricks, zircon, zirconia, etc., are not found within our own borders.

MANUFACTURE

Acquaintance with the general processes of manufacture of refractory materials is of great assistance in appreciating some of the properties and limitations of these products. In particular, if furnace designers were familiar with the difficulties in shaping and drying large and intricate shapes, it is probable that they would contrive to utilise smaller and simpler designs; sharp re-entrant angles would be avoided wherever possible.

In the manufacture of firebricks the raw materials are subjected to preliminary treatments including mixing, grinding, and screening. The subsequent processes may vary to suit the type of clay and product. In the normal plastic process, grog, that is to say graded, burnt refractory material, is worked into the clay, and the bricks are shaped by hand-moulding, extrusion or pressing. Dry-pressing consists in subjecting the clay and grog, at low moisture content, to a high pressure, the plasticity of the clay playing a less important part. The clay may be de-aired before or during shaping by these processes; this removal of entrapped air leads to the production of a dense brick if such is required.

The "green" (i.e. unfired) bricks are dried on a hot floor, or in a chamber or tunnel dryer. The removal of water leads to shrinkage, and unless carried out under controlled conditions is liable to cause strain and distortion, if not actual cracking. Drying difficulties are particularly marked when dealing with large, intricate shapes. The dried bricks are fired in intermittent or con-

TABLE 99. TYPICAL COMPOSITION OF REFRACTORY MATERIALS

Constituent of brick	Type of Brick								
	Firebrick	Aluminous firebrick	Siliceous or semi-silica brick	Silica brick	Sillimanite brick	Magnesite brick	Chrome brick	Chrome magnesite	Dolomite brick (stabilised)
Silica—SiO_2 ..	55–70	50–55	75–90	94–96	25–35	1·5–5	3–10	3–10	12–15
Alumina—Al_2O_3 ..	25–38	38–45	8–15	0·5–2	55–65	1–4	10–25	5–15	2–3
Titania—TiO_2 ..	1–2	1–2	0·5–1	0·2–0·5	0·5–1	tr	tr	tr	tr
Iron oxide—Fe_2O_3 ..	2–5	2–4	1–2	0·5–1·5	0·5–1·5	2–8	12–25	8–20	2–4
Chromic oxide—Cr_2O_3	—	—	—	—	—	—	35–45	20–35	—
Magnesia—MgO ..	0·5–1	0·5–1	0·5–1	0·1–0·5	0·5–1	84–92	15–35	40–60	38–42
Lime—CaO	0·5–1	0·5–1	0·5–1	2–2·5	0·5–1	2–5	2	1–4	38–42
Alkalies—K_2O+Na_2O	1–2	0·5–1	0·5–1·5	0·5–1·5	0·5–1	tr	tr	tr	tr

tinuous kilns, at a temperature depending on the nature of the clay. This temperature is chosen to give a strong product free from after-contraction.

In consequence of the almost complete lack of plasticity in ground ganister or quartzite, a somewhat different moulding technique is used in the manufacture of silica bricks. The rock is crushed and graded, and is then mixed with milk of lime and a temporary bond in an edge-runner mill. The temporary bond, an organic material such as sulphite lye, gives some strength to the " green " bricks, enabling them to be handled without damage to the corners and edges ; the milk of lime forms the final bond in the fired product, dissolving some silica and forming, together with the impurities, a strong matrix. The moulded bricks are dried, and then fired to a high temperature. The firing process in silica brick manufacture leads to a quite different set of reactions from those which occur on firing clay. These reactions depend on the ability of silica to exist in a number of different crystalline modifications ; in nature, silica normally occurs as quartz, but this, on firing, changes to forms of different physical properties. These forms, known as cristobalite and tridymite, and their significance to the user of silica refractories, will be discussed later.

Most other refractory raw materials resemble quartzite in their rock-like nature, and in consequence of their lack of plasticity the production of bricks from such materials either depends on the addition of sufficient clay to make moulding possible, or on the use of a high forming pressure. Sillimanite bricks are made by the first method, or by a combination of the two methods ; magnesite, chrome and dolomite bricks are shaped under pressure without clay addition. Precalcination of the base material may be necessary, and may exert a profound effect on ultimate properties, as for example in sillimanite.

Two further methods occasionally adopted in the production of refractories are worth noting. The first, chemical bonding, is sometimes used in the production of certain basic refractories ; the bricks are not fired, their strength being developed in the cold by chemical reaction. Chrome-magnesite bricks made in this way generally have a high resistance to spalling, and have been successfully used in metallurgical furnaces. In the second special method the raw materials are treated rather like a metal, being fused and cast in moulds. This technique is difficult and expensive, due to the very high melting point of the materials to be fused, but it brings the constituents into a state of equilibrium and enables crystallisation to develop ; fusion casting has been used with conspicuous success in the production of mullite tank blocks for use in the glass industry.

CONSTITUTION

On the constitution of a refractory depend its chemical and physical properties, and hence its durability. Clays consist of a number of hydrous aluminosilicate minerals which break down on heating, and after passing through a somewhat ill-defined intermediate stage, tend to form the only compound of alumina and silica which is stable at high temperatures, namely, mullite ($3Al_2O_3$, $2SiO_2$). X-ray examination of clays fired at increasingly high temperatures has shown that mullite begins to develop at about 1,000° C., although the crystals are then too small to be seen under a microscope.

Clay bricks after firing contain much fused material together with varying amounts of mullite. In addition, most firebricks contain quartz grains ; these are not in equilibrium with the mullite or the glass, but their solution by the glass is extremely slow, and even after a firebrick has had prolonged use at high temperature, quartz may still be found. However, it is on the mullite development and the glassy bond present that the high temperature strength, resistance to slagging, and other properties largely depend.

Perhaps the most interesting constitution of any refractory material is that

of silica products. At atmospheric temperature, silica can exist in the three crystalline modifications: (1) quartz, (2) cristobalite, and (3) tridymite; quartz and cristobalite themselves have two crystalline forms, while tridymite has three forms. These crystalline varieties of silica are formed from one another on heating and cooling, according to Table 100.

In using Table 101, the reader should imagine that he is starting with a brick containing all three minerals, quartz, cristobalite and tridymite, and that this brick is built into a furnace which is then to be heated up to working temperature. By proceeding up the temperature scale he can then follow easily the modifications that occur as the structure is heated and the volume changes to which they give rise.

Table 101 gives the conventional way of expressing these changes. The

TABLE 100. TRANSFORMATIONS OF THE SILICA MINERALS

Temp. °C.		Linear change on increasing temp. Per cent.	Velocity of reaction	Notes
1,800–				
1,700–	–1,710 cristobalite melts			In the range 1,470–1,710° C. β_2 cristobalite is the stable form and on long continued heating is slowly formed from quartz and tridymite.
1,600–	β quartz ⟶ β cristobalite	+ 4·85	Rather slow at 1,470, increases as temp. rises.	
1,500–	–1,470 Also β_2 tridymite			
1,400–	⟶ β cristobalite.	− 0·65		
1,300–				
1,200–	β quartz ⟶ β_2 tridymite	+ 5·5	Very slow at 870° C., increases as temp. rises.	Between 870° C. and 1,470° C. β_2 tridymite is the stable form and is slowly formed from cristobalite and quartz.
1,100–	β cristobalite ⟶ β_2 tridymite			
1,000–				
900–				
	– 870			
800				
700–				Up to 870° C. quartz is the stable form, but any cristobalite and tridymite that have been formed at higher temps. only invert back to quartz very slowly.
600–				
	– 572 α quartz ⟶ β quartz	+ 0·45	Instantaneous.	
500–				
400–				
300–				
	– 250 α cristobalite ⟶ β			
	– 220 cristobalite	+ 1·05	Instantaneous.	
200–				
	– 165 β_1 tridymite ⟶ β_2			
	– 117 tridymite	+ 0·06	Instantaneous.	
100–	α tridymite ⟶ β_1 tridymite	+ 0·15	Instantaneous.	
0–				

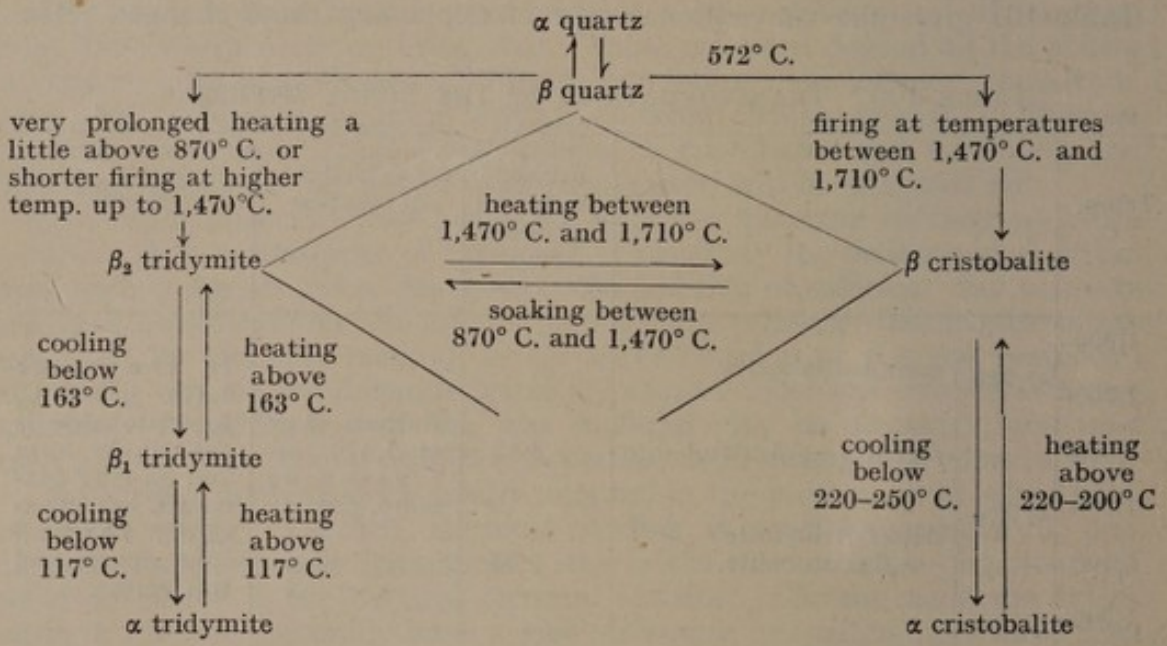
Arrows indicate the direction of the change with increasing temperature.

reversed arrows in this table signify reversable transformations which take place respectively on heating and cooling.

The terms α and β are sometimes referred to as the low and high modifications.

The different forms (α, β, β_1 and β_2) of the three major crystalline varieties of silica form practically instantaneously at the temperatures stated, but the changes from one major variety to another, e.g. from quartz to cristobalite is much slower. The significance of these changes to the user of silica bricks lies in the considerable changes in volume which accompany certain of these transformations.

TABLE 101. TRANSFORMATIONS OF THE SILICA MINERALS



In the unfired state a silica brick consists of α quartz; when this brick is placed in the kiln and fired the first change is the transformation to β quartz at 572° C., accompanied by a linear expansion equal to about 0·45 per cent. The volume expansion is three times the linear expansion. At higher temperatures, particularly when the added lime begins to form a viscous bond, some of the β quartz is dissolved and is later precipitated as cristobalite or tridymite depending on the temperature conditions. In the fired brick we thus have quartz, tridymite, and cristobalite, in varying amounts.

When such a brick is used in a furnace lining the following sequence takes place: on warming up the furnace, the α tridymite suddenly changes to β_1 tridymite at 117° C. (only a little above the boiling-point of water); sudden expansion equivalent to about 0·15 per cent. linear takes place. At 163° C. the β_1 tridymite undergoes a further transformation, but the expansion is small. In the temperature range 220°–250° C. the α cristobalite suddenly changes to β cristobalite, the exact temperature of this transformation depending on the thermal history of the cristobalite. This cristobalite change is the most serious from the standpoint of spalling, for the linear expansion is over 1 per cent. On further heating, any unchanged quartz suffers transformation at 572° C., with a volume change of the order previously noted. Above this temperature the expansion of a silica brick is small and regular unless much residual quartz is present; the quartz is converted into tridymite and/or cristobalite during use, and if sufficient quartz is present permanent growth of the brickwork is likely to result. The close relationship between the mineral constitution of silica bricks and their sensitiveness to thermal shock at low

temperatures will be evident ; further notes on this subject will be found in the section on spalling.

Magnesite bricks have a relatively simple constitution. Natural magnesite, magnesium carbonate, on dead-burning loses carbon dioxide, and develops crystals of periclase (magnesium oxide). This is the only crystalline form of magnesia, but on continued firing, the crystals tend to grow. The matrix of magnesite bricks varies according to the nature of the impurities in the raw material ; if siliceous, forsterite ($2MgO, SiO_2$) may be formed ; if both siliceous and calcareous, the matrix may approximate to monticellite (CaO, MgO, SiO_2). Iron oxide can enter the periclase grains in solid solution or can form magnesio-ferrite (MgO, Fe_2O_3).

Chrome and chrome magnesite refractories have a particularly complex constitution. Chrome ore consists of a complex spinel, which may be written $(Fe'', Mg)O.(Cr, Al, Fe''')_2O_3$, together with gangue, usually siliceous. On firing, the spinel may decompose and reaction may follow between it and the gangue. The precise mechanism of these reactions is at present the subject of research, and must evidently have a close bearing on the properties of bricks containing chrome ore.

Dolomite, the double carbonate of lime and magnesia, breaks down on heating to form the two oxides. On exposure to the air the lime takes up moisture and the mass soon disintegrates. To overcome this defect, dolomite is stabilised by firing with a siliceous mineral, serpentine for example, whereby the lime of the dolomite combines with the silica to form, chiefly, tricalcium silicate. A stabilised dolomite brick therefore consists mainly of periclase and tricalcium silicate ; some calcium orthosilicate, calcium ferrite and brown-millerite ($4CaO, Al_2O_3, Fe_2O_3$) may also be present in small amounts.

Sillimanite bricks are made from one or other of the sillimanite minerals, viz. sillimanite, andalusite and kyanite. These three minerals all have the composition Al_2O_3, SiO_2, but have different physical properties. In particular, they break down at different temperatures on heating ; the expansion accompanying this breakdown varies from one mineral to another. The final product is the same whichever mineral is heated however, mullite and cristobalite being formed. Fired sillimanite bricks consequently consist of a mass of mullite needles in a siliceous ground mass. Commercial sillimanite bricks in this country are manufactured from calcined kyanite.

Other types of refractory are of less general importance. Silicon carbide bricks may consist almost entirely of that mineral (recrystallised bricks) or may contain a proportion of clay as bond (clay-bonded bricks). Forsterite refractories are composed essentially of olivine crystals $(2Mg, Fe)O, SiO_2$. Alumina bricks consist mainly of corundum.

REFRACTORY PROPERTIES

Few, if any, refractory materials have a sharp melting point. In consequence of their complex nature, they normally have a softening range. This is particularly true of fireclay products, as will be more readily appreciated by reference to the Al_2O_3-SiO_2 equilibrium diagram (Fig. 179). This diagram may be taken to represent the melting-point relations of firebricks, siliceous bricks, sillimanite and alumina bricks ; the error introduced by the small proportion of fluxes may be considered to lead merely to a general lowering of the softening temperatures. The fusion relations of silica bricks cannot be considered on the basis of this diagram owing to the presence of an important though small amount of lime ; however, it may be stated that the presence of much alumina in a lime-bonded silica brick is particularly detrimental in its effect on the fusion and slagging of the brick.

Firstly, the eutectic composition is 94·5 per cent. silica, 5·5 per cent. alumina.

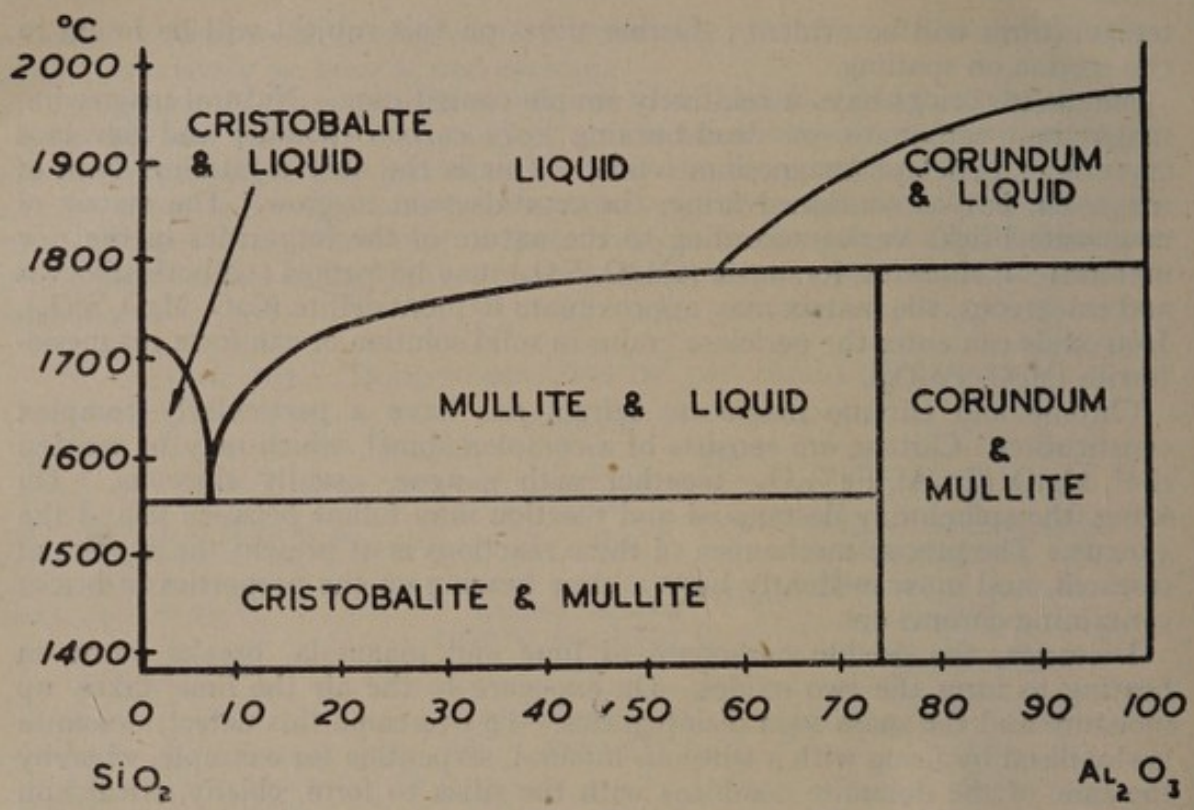


FIG. 179. Alumina-silica equilibria diagram.

This suggests how important is the absence of alumina in the raw material used for silica brick production ; in the presence of lime the eutectic temperature is still further depressed to 1,170° C. Siliceous bricks containing about 80 per cent. SiO_2 are past the eutectic composition, but in all alumino-silicate refractories containing less than 72 per cent. Al_2O_3, some liquid will form at some temperature below 1,545° C. (below the binary eutectic temperature owing to the presence of impurities such as lime, magnesia, iron oxide, alkalies, etc.). The proportion of liquid will increase with rising temperature until eventually the whole mass is fluid.

At some temperature between that at which liquid first forms and that at which the whole brick is fluid, the proportion of liquid will be such that the brick cannot support its own weight and it will slowly deform. If the test piece is in the form of a pyramid, 1½ inch high or a triangular base of ½ inch side (one edge of the pyramid being perpendicular to the base), and the specimen is heated at the rate of 50° C. per five minutes, the temperature at which the test piece has bent until the tip is level with the base is called the " refractoriness." The temperature at which this will occur evidently increases with increasing alumina content (see Table 102).

A sillimanite or high-alumina brick containing over 72 per cent. Al_2O_3 will not form liquid at 1,545° C. ; no liquid will be produced (neglecting the fluxes present) until a temperature of 1,810° C. has been reached. At this temperature any mullite present decomposes into corundum and a liquid. The high refractoriness of sillimanite bricks is therefore readily explained. Most other refractories have a fusion point (refractoriness squatting value) above that at which normal furnaces are operated (see Table 102).

The refractoriness of a product is often of less importance than the refractoriness-under-load. In all furnace construction the brickwork is subjected to stress ; this may arise from the weight of the superincumbent brickwork, by the thrust of an arch, or by pinching due to inadequate expansion allowance. This stress may cause the bricks to deform at a temperature considerably below

TABLE 102. AVERAGE REFRACTORINESS SQUATTING VALUES

Brick	Seger cone No.	°C.	Brick	Seger cone No.	°C.
Silica	32–34	1,710–1,750	Sillimanite ..	35	1,770
Siliceous	28–31	1,630–1,690	Magnesite ..	38	1,850
Firebrick (23–30% Al_2O_3) ..	27–30	1,610–1,670	Bauxite ..	38	1,850
,, (30–35% ,,) ..	28–31	1,630–1,690	Chrome ..	38	1,850
,, (35–40% ,,) ..	29–33	1,650–1,730	Silicon carbide*	38	1,850
Aluminous firebrick ..	32–35	1,710–1,770	Carbon † ..	38	1,850
			Zirconia ..	38	1,850

* Begins to decompose at lower temperatures.
† Decomposes unless atmosphere is strongly reducing.

their fusion point. The effect is particularly marked in firebricks. Under a load of 28 lb. per square inch (the standard adopted in testing this type of brick, and equivalent to 2 kg. per square centimetre), 5 per cent. subsidence will generally occur at a temperature about 200° C. below the fusion point. Subsidence commences at an even lower temperature, the failure being one of slow

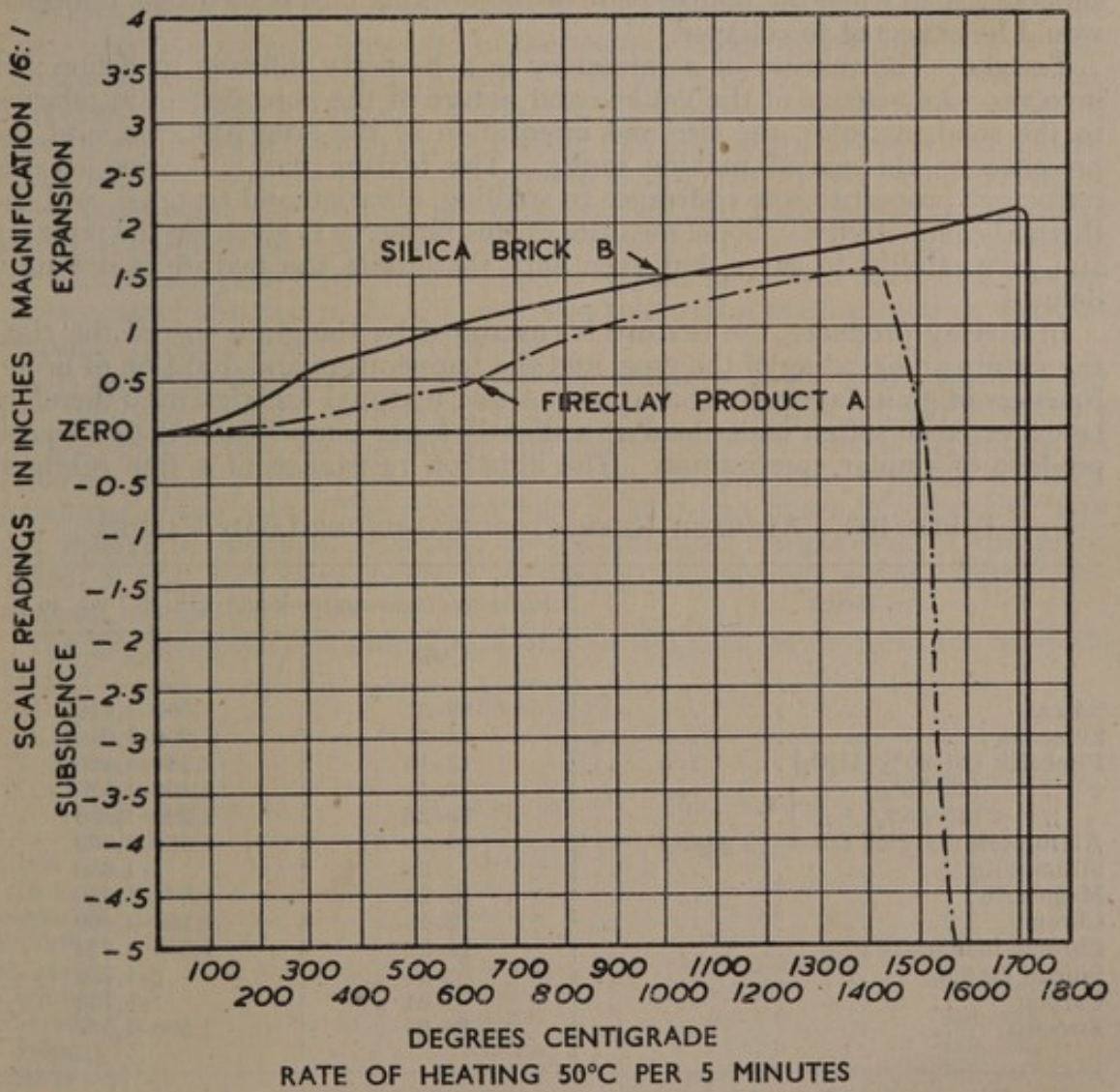


FIG. 180. Refractoriness-under-load curves, determined with a load of 50 lb./sq. in.

deformation. In contrast, silica bricks maintain their strength practically to their fusion temperature. This is attributed to the network of cristobalite and tridymite which maintains its rigidity even when the interstitial matter is quite fluid ; when the network of crystals itself breaks down, the brick shears suddenly. Fig. 180 shows typical results on a silica brick and a firebrick.

Magnesite bricks and chrome bricks also generally shear under load ; the temperature at which this occurs may be as much as 500° C. below their fusion point. This is due to the cubic crystal structure of periclase and spinel, which is inimical to crystal intergrowth ; hence, when the matrix of such bricks softens, the whole mass loses strength.

In chrome magnesite bricks, much of this weakness has been overcome : some of the magnesia combines with the siliceous gangue of the chrome ore to form forsterite ; this mineral, when pure, does not melt until 1,890° C., and if a network of forsterite can be developed within a chrome magnesite brick the refractoriness-under-load is improved in a remarkable manner.

Sillimanite bricks have good mechanical strength at high temperatures since little of the Al_2O_3-SiO_2 eutectic is present and the mullite crystals, being needle-shaped, form a strong intergrowth. Some comparative data on the refractoriness-under-load of various types of refractories are given in Table 104. It should be remembered that in practice only one face of a brick is usually at the highest temperature, so that frequently the colder part of a brick can support a load when the temperature of the working face is such that the brick would be expected to collapse.

Texture. The texture of a refractory is a property difficult to define ; it involves a knowledge of the volume and nature of the pore system in relation to the solid material, the size and orientation of the grog particles, and the presence or absence of making faults. The texture has a bearing on the refractory properties, the resistance to spalling, abrasion and slagging, and the thermal conductivity. Some measure of this property is given by the porosity and permeability to gases, but even with these data the texture is not fully defined.

In fireclay products, the texture is controlled by the grain size of the clay, the nature and grading of the grog, and the temperature and duration of firing. Fineness of grain favours vitrification. A fine textured firebrick may therefore be expected to soften under load at a slightly lower temperature than a coarse product of similar composition. The abrasion resistance of a fine textured

TABLE 103. AVERAGE REFRACTORINESS-UNDER-LOAD VALUES

Brick	Refractoriness-under-load* (28 lb./ sq. in.)	
	Cone	° C.
Silica	29–32	1,650–1,710
Siliceous	18–27	1,500–1,610
Firebrick (25–30% Al_2O_3)	13–16	1,380–1,460
,, (30–35% ,,)	16–19	1,460–1,520
,, (35–38% ,,)	19–26	1,520–1,580
Aluminous firebrick (38–45% Al_2O_3) ..	26–29	1,580–1,650
Sillimanite	>29	>1,650
Magnesite	18–28	1,500–1,630
Chrome	12–16	1,350–1,460
Chrome magnesite	18–32	1,500–1,710
Silicon carbide	>34	>1,750
Carbon	>34	>1,750
Zirconia	18–20	1,500–1,530

* Indicated by the temperature at which collapse occurs.

brick is usually high. The relationship between the spalling tendency and the texture is complex. It has generally been considered that a fine textured refractory will be more sensitive to sudden changes in temperature than a coarse textured product. This is not invariably true, however, for the process of manufacture has a modifying influence: with the dry process, for example, it is possible with some clays to develop a close texture which is still resistant to spalling. The relationship between the texture and the thermal properties will be noted later.

It is difficult to give porosity and permeability values which may be looked on as typical of various types of brick. The porosity of firebricks, for instance, may vary from about 12 to 25 per cent. or more; that of silica bricks from about 20 to 30 per cent.

SPECIFIC GRAVITY

The true specific gravity of the material composing a brick, as distinct from the bulk density of the brick as a whole (i.e. including the pore spaces), is of special significance only for silica bricks and magnesite bricks. This is due to the comparatively constant composition of those products and the important differences in specific gravity between the various forms of silica in the one, and the increase in specific gravity with periclase development in the other.

Quartz has a specific gravity of 2·65; cristobalite, 2·33; tridymite, 2·28. From what has already been said in the section on constitution, it follows that the specific gravity of a silica brick is a good guide to the degree of conversion. This does not apply to the siliceous class of refractories. Thus a brick of specific gravity 2·45 evidently contains a considerable amount of unconverted quartz, which is likely to cause permanent growth during use. The specific gravity of most present-day silica bricks lies between 2·32 and 2·37, indicating adequate quartz conversion to cristobalite and tridymite. It is not possible to assess the relative amounts of cristobalite and tridymite in a silica brick by a specific gravity determination; for this purpose micro-examination and/or thermal expansion measurements are necessary.

The fact that the density of magnesia increases with the heat treatment has been used as a method of assessing the adequacy of the kiln firing of magnesite bricks. In the past, the figure 3·65 has usually been considered the maximum value to which a magnesite brick in which the periclase crystals are fully developed will attain. The present view is that this figure is somewhat high and should be modified according to the chemical composition of the brick tested, a high iron oxide content increasing, and a high silicon content decreasing, the actual value.

The specific gravity of a firebrick is of little significance, owing to the complex

TABLE 104

Type of brick	True sp. gr.	Bulk density	Weight of bricks lb./cu. ft.
Silica	2·3–2·4	1·7–1·9	106–118
Siliceous	2·5–2·6	1·8–2·0	112–125
Firebrick	2·6–2·7	1·9–2·1	118–130
Bauxite	3·1–3·4	1·8–2·1	112–130
Magnesite	3·5–3·7	2·7–3·0	160–190
Chrome	3·7–4·2	2·8–3·5	170–220
Zirconia	4·8–5·9	4·0–4·6	249–274
Carbon	2·7–3·1	1·9–2·0	118–125
Sillimanite	3·0–3·2	2·0–2·2	125–137
Silicon carbide	3·1	2·2	137

nature of the material. Some typical data on the specific gravity of refractories (taken chiefly from " Technical Data on Fuel," edited by H. M. Spiers), are given in Table 104.

EFFECT OF RE-FIRING ON THE VOLUME STABILITY

(" After-contraction " or " after-expansion.")

The changes which occur during the firing of refractory materials belong to the class known as arrested reactions, in which the chemical and physical changes are stopped before they have gone to completion. During the firing of firebricks, for example, the actions taking place result in a certain amount of contraction. If these actions are not sufficiently completed this contraction may continue during use. If such after-contraction is considerable, the masonry of the furnace will certainly become displaced, open joints, cracks and distortions resulting. For fireclay products, an after-contraction of 1 per cent. after re-firing for two hours at 1,400° C. is considered high ; the actual temperature at which a brick should be tested and the allowable after-contraction must depend on the conditions of service.

With silica bricks the kiln treatment leads to expansion, but here again the reactions are generally arrested a little before completion. During use at high temperatures the residual quartz may be converted into the forms of lower density, and growth may thus occur. This growth will take place even against high compressive forces, so that it is essential that the after-expansion of a silica brick should be low.

Magnesite shrinks during firing due to the crystallisation of periclase from the originally amorphous magnesia. Chrome and chrome magnesite bricks may contract or expand ; it is thought that the kiln atmosphere plays a determining part with bricks containing chrome ore. Adequate firing is necessary always to take the reactions as near to completion as is practicable.

THERMAL EXPANSION

To the furnace builder, the ordinary or reversible thermal expansion of the materials of construction is of particular significance, for some allowance for its accommodation must generally be made in the design. The thermal expansion of a refractory is also an important factor in determining its resistance to thermal shock.

In general the thermal expansion is slight above about 1,000° C. At high temperatures most refractories develop a certain amount of " give," due to the softening of the bond ; it follows that some of the expansion can then be taken up within the brick itself.

Firebricks have a low, fairly uniform expansion up to 1,000° C. ; this is generally of the order of 0·5 per cent. or less. Silica bricks behave quite differently, however. This has already been indicated in the section on constitution, but will be better appreciated from the diagrams (Figs. 181 and 182).

It will be seen that tridymite shows two comparatively small irregularities, while cristobalite and quartz each show one large one. A normal silica brick will show each of these discontinuities in its thermal expansion curve, as indicated on the diagram ; careful examination of a thermal expansion curve of a silica brick, or, better, of the curve relating the coefficient of expansion with temperature, will give a good indication of the constitution of the brick. The total thermal expansion of a silica brick between 0° and 1,000° C. is usually 1·2–1·3 per cent.

The thermal expansion of magnesite bricks is comparatively high, 1·3–1·4 per cent. up to 1,000° C. ; that of a chrome or chrome magnesite brick is much lower unless the atmosphere is reducing. Sillimanite has a low thermal expansion, approximately 0·5 per cent. up to 1,000° C. ; silicon carbide is of the same

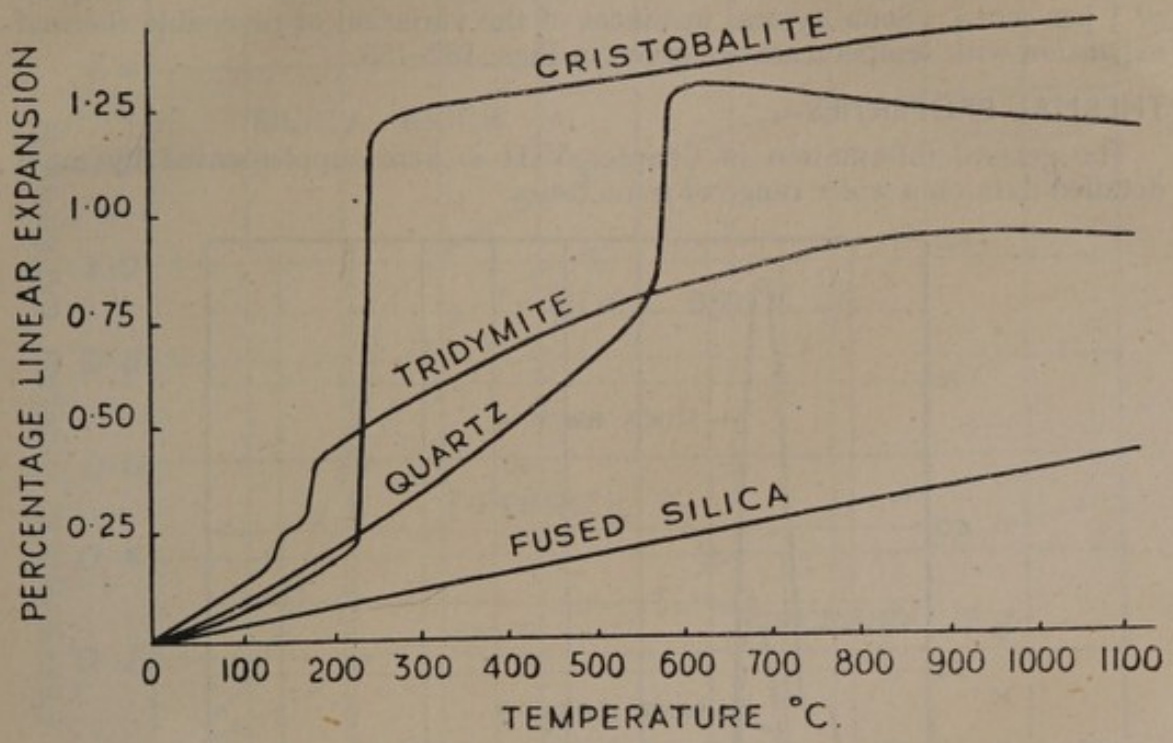


FIG. 181. Expansion of the silica minerals.

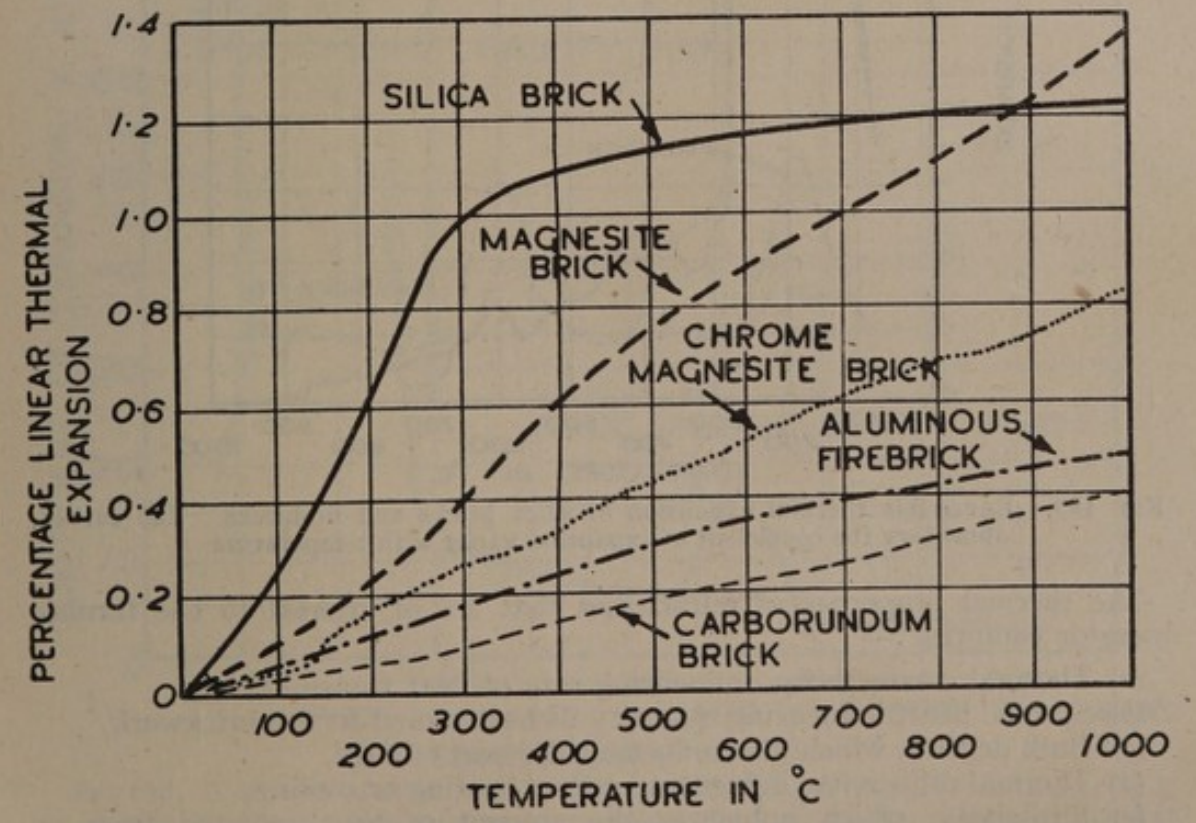


FIG. 182. Reversible linear expansion of several types of refractories.

order, but the thermal expansion of corundum bricks is higher (in the region of 1 per cent.). Some typical instances of the variation of reversible thermal expansion with temperature are given in Figs. 182–185.

THERMAL PROPERTIES

The general information in Chapter VIII is here supplemented by more detailed data on a wider range of refractories.

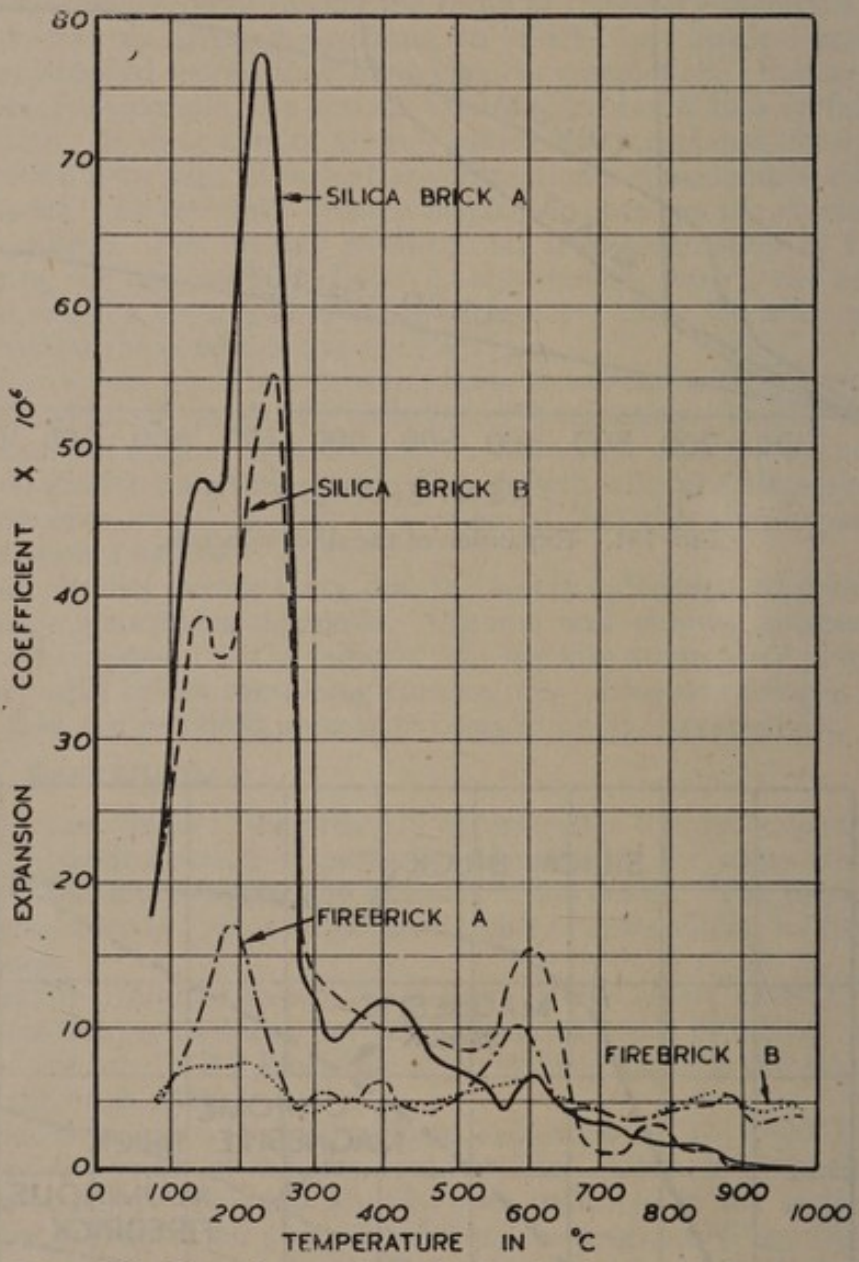


FIG. 183. Reversible thermal expansion of silica bricks and firebricks. The curves show how the coefficient of expansion varies with temperature.

The thermal properties of refractories that are of interest to the furnace operator comprise :—

(*a*) Thermal conductivity, influencing rate of heat transmission.
(*b*) Specific heat, influencing quantity of heat stored in the brickwork.
(*c*) Bulk density, which also influences the heat storage.
(*d*) Thermal diffusivity, influencing rate of heating or cooling.
(*e*) Emissivity, which influences the amount of heat radiated from or absorbed by furnace walls, roof and floor.

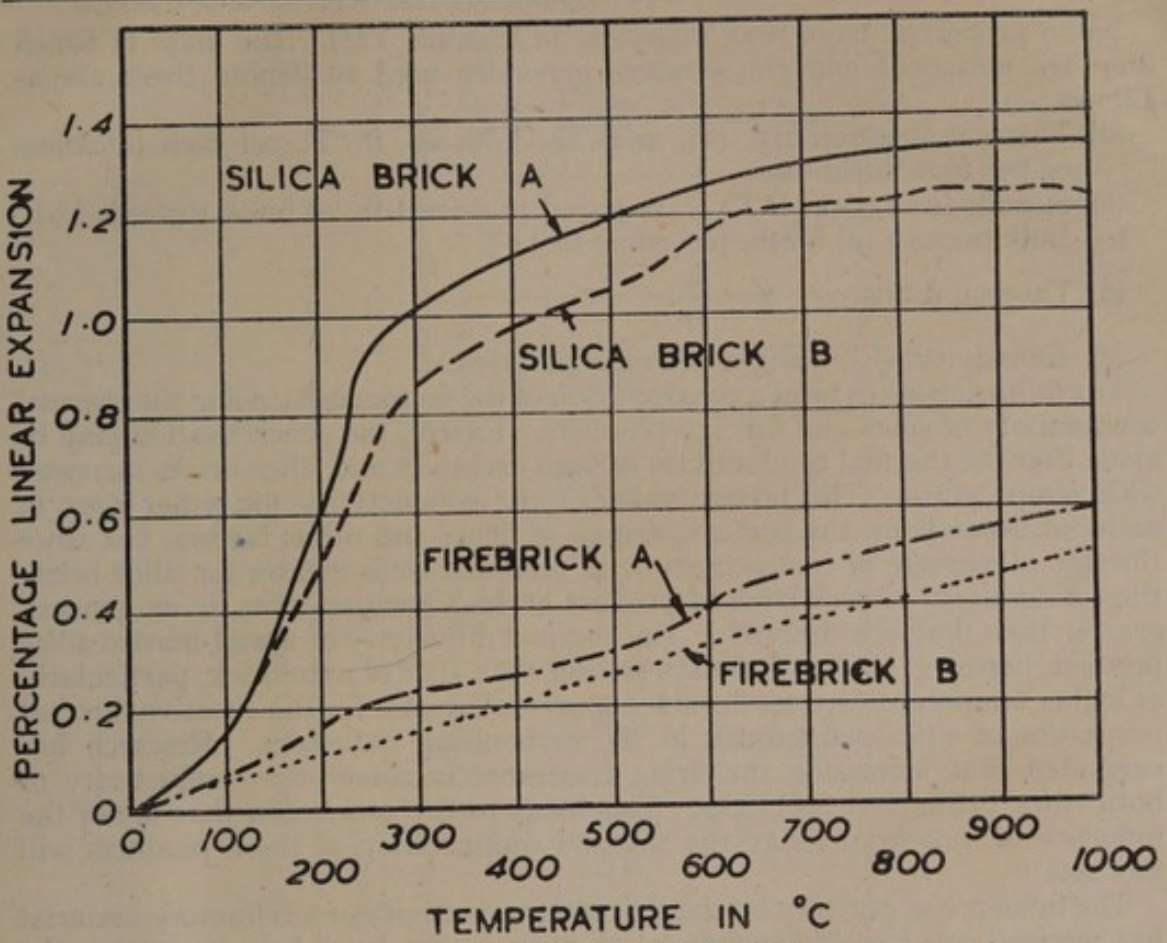


FIG. 184. Reversible linear expansion of silica brick and firebricks.

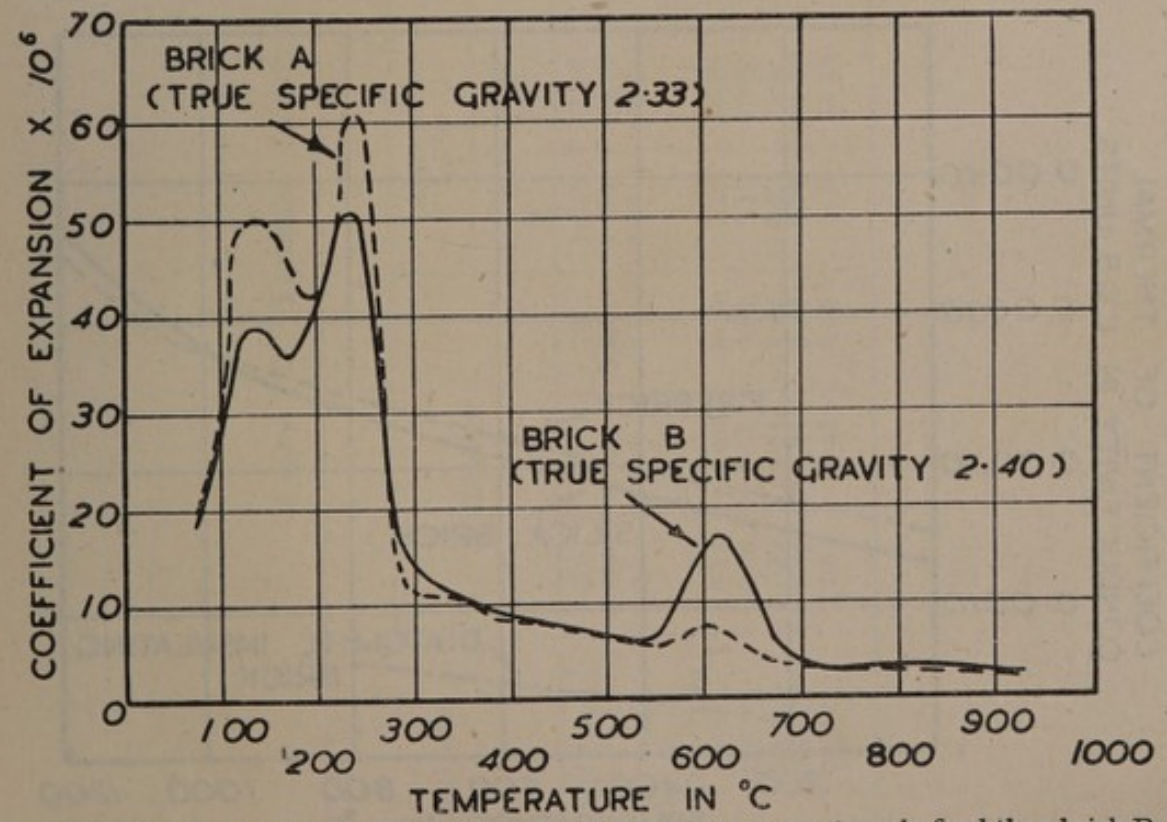


FIG. 185. Expansion of silica bricks. Brick A has been more strongly fired than brick B; its specific gravity has been reduced to an extent which shows that most of the original quartz has been converted and its expansion due to the reversible expansion of the silica minerals is reduced.

These properties have been discussed in Chapter VIII ; the units in which they are measured and the symbols generally used to denote them are as follows :—

(*a*) Thermal conductivity, (k), in B.Th.U./hr./sq. ft./°F. per inch thickness or per foot thickness.

(*b*) Specific heat (s) in B.Th.U. required to raise 1 lb. of brick through 1° F.

(*c*) Bulk density (ρ), in lb. per cubic foot.

(*d*) Thermal diffusivity, $\alpha = \frac{k}{s\rho}$.

(*e*) Emissivity, (E), being a ratio has no units.

The following notes form a consideration of the factors influencing the thermal conductivity of silica and fireclay products. Firstly, the generalisation may be made that the thermal conductivity of both firebricks and silica bricks increases with temperature. The precise value of the conductivity for either class of material depends on the texture, degree of firing and other factors, but since the rate of increase of conductivity with temperature is greater for silica brick, the conductivity of this class of product at high temperatures, is, in general, greater than that of a firebrick. The thermal diffusivity of a well-burned silica product, however, is almost always greater than that of a firebrick, particularly at higher temperatures, a fact which appears to account for the effective thermal properties of silica refractories in the carbonising industries. Research has suggested that increasing the firing treatment increases the conductivity of both silica bricks and firebricks. This leads to the conclusion that under the influence of industrial usage the thermal conductivity of these products will increase.

The influence of porosity on the thermal conductivity of a refractory material has received considerable attention. A more porous brick has, in general, the lower conductivity. However, the size of the pores, in addition to their number,

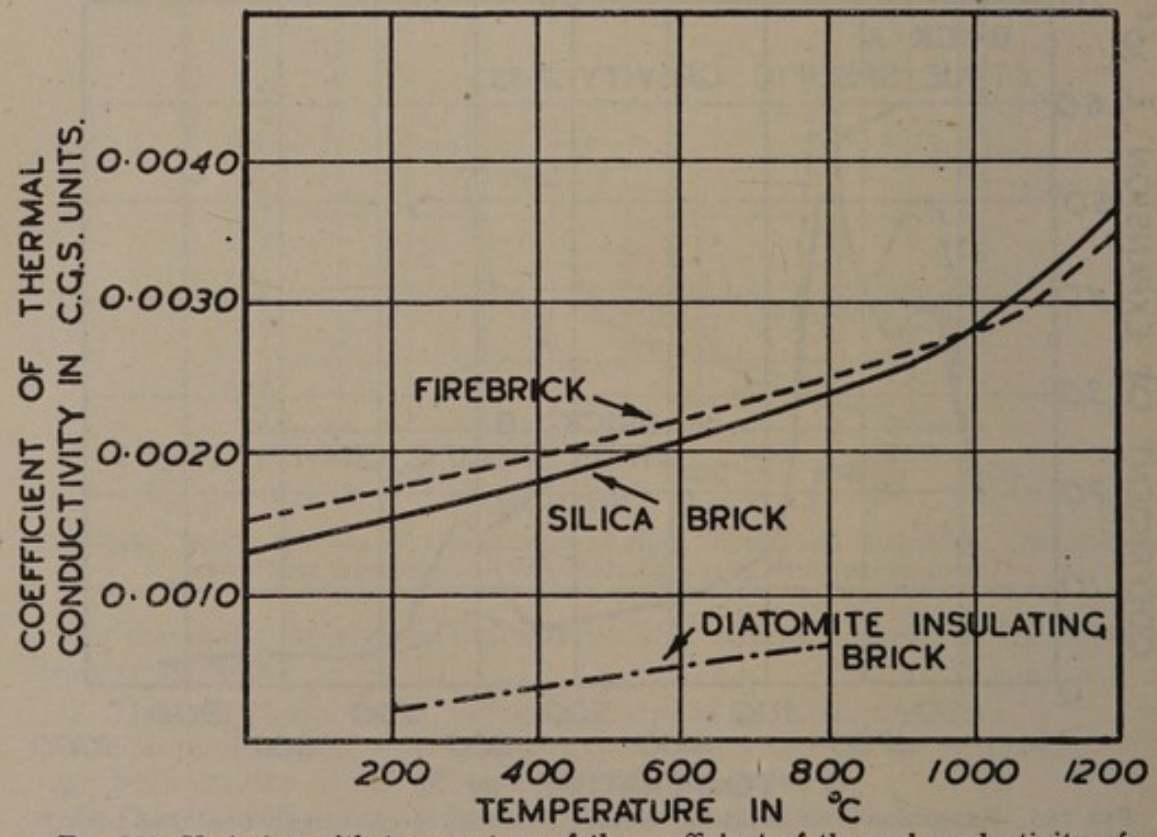


FIG. 186. Variation with temperature of the coefficient of thermal conductivity of firebrick, silica brick and insulating brick.

TABLE 105. THERMAL CONDUCTIVITY OF REFRACTORY MATERIALS

Characteristic	Material												
	Firebrick			Alumina brick		Silica brick		Zirconia brick	Magnesite brick aver.	SiC 90%	Silicon carbide bricks		
				70% Al_2O_3	81% Al_2O_3	96% SiO_2	93% SiO_2				SiC 77%	SiC 57%	SiC 49%
Burning temp.-cone ..	10	14	16	14/15	15/16	16	16	15/6		15	10/11	10	10/11
do. -° C. ..	1,300	1,400	1,460	1,420	1,450	1,460	1,460	1,450		1,435	1,310	1,300	1,310
Porosity—per cent. ..	30	26·8	26·5	38·5	25·1	24·0	28·1	18·9		25·6	21·6	22·8	22·9
Conductivity—k B.Th.U./ft.²/hr./°F./in. at													
300° C.	6·13	7·30	8·35	7·25	16·35	9·38	8·16	21·1			90	42	33
500° C.	6·76	7·76	8·92	7·10	15·32	10·12	9·02	17·35	24·9	108	80	40	32
700° C.	7·25	8·11	9·30	6·92	14·8	10·84	9·76	15·5	20·6	90	71	39	30
900° C.	7·50	8·31	9·48	6·77	14·53	11·58	10·43	14·2	17·4	77	63	34	28·5
1,100° C.	7·61	8·43	9·56	6·60	14·48	12·28	11·08	13·2	14·8	69	55	31	27

is a factor of importance. This is due to the fact that heat transfer can take place across the pores by radiation; this factor comes into particular prominence at high temperatures.

Not all refractories show a positive temperature coefficient of conductivity, however, two notable exceptions being magnesite and silicon carbide; corundum and zirconia refractories also probably have a negative temperature coefficient, but considerably less marked.

Some data on the thermal conductivity of refractories (taken, with the exception of those for magnesite bricks, from "Technical Data on Fuel") are given in Table 105.

The data in Table 105 refer to the thermal conductivity at the temperatures stated. Data for the mean value of k over the temperature range 25° C. to t° C. (also mainly from "Technical Data on Fuel") are given in Table 106. The conductivity of certain insulating bricks is given later. The character of the variation of the coefficient of thermal conductivity of firebrick, silica brick and diatomite insulating brick is illustrated in Fig. 186.

Specific heat data are shown in Fig. 187. It will be noted that the specific heats of silica bricks and firebricks differ little; the values for magnesite and silicon carbide bricks are also similar.

MECHANICAL PROPERTIES

The mechanical properties of a refractory determined at room temperature must be used with extreme caution as a guide to the probable behaviour of the

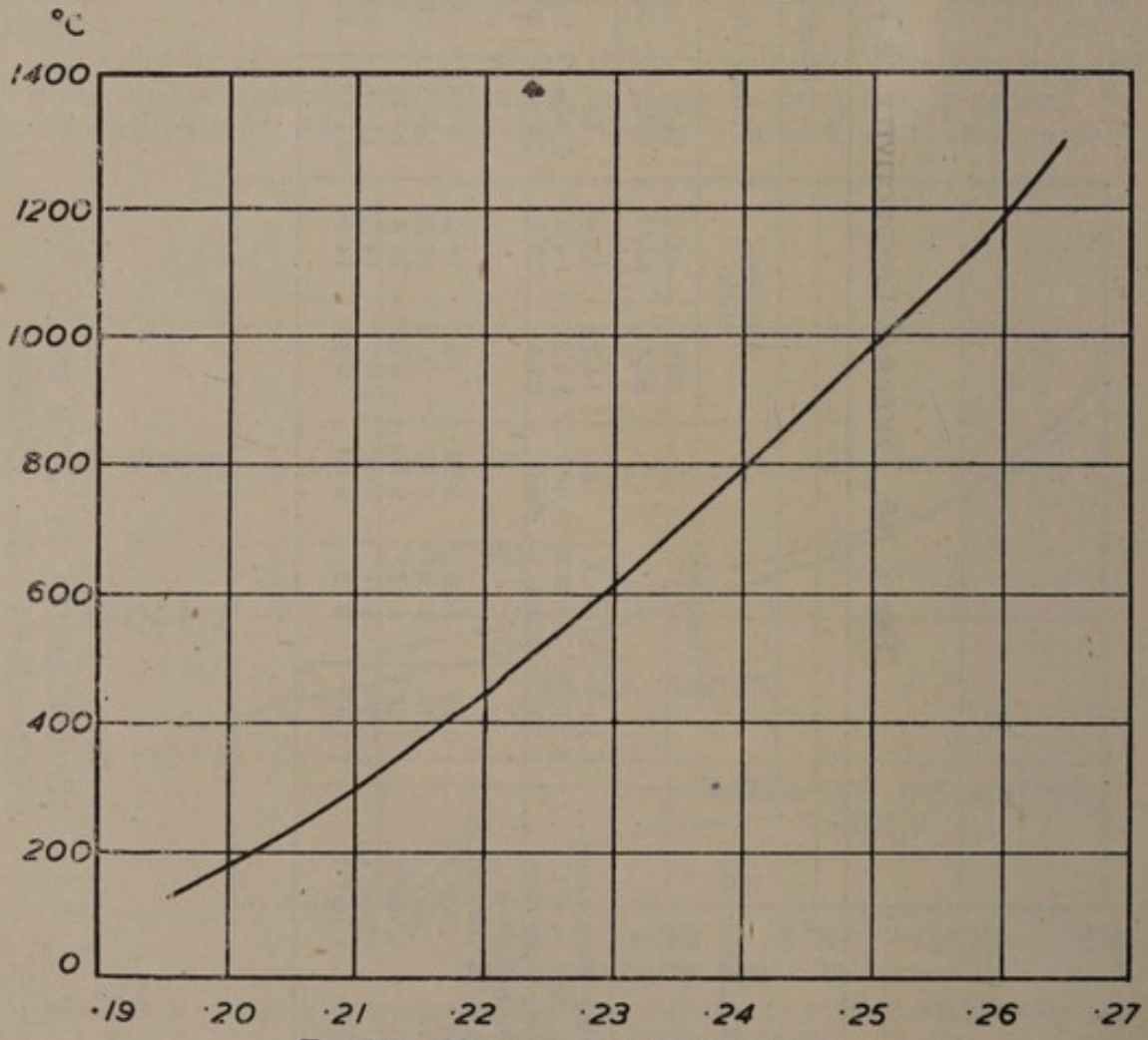


FIG. 187. Mean specific heat of firebricks.
(S. T. Wilson and A. D. Holdcroft.)

TABLE 106. MEAN VALUES OF THERMAL CONDUCTIVITY BETWEEN 25° AND t° C. IN B.TH.U. PER HOUR PER SQUARE FOOT PER °F. PER INCH THICKNESS *

Material / t =	100° C.	500° C.	1,000° C.
Corundum (fused Al_2O_3) ..	—	24·3 (between 650° C. and 1,250° C.	—
Sillimanite brick	11	12	12
Silicon carbide brick	—	—	55·5–69·5
Chrome brick	12 ± 3	12 ± 3	12 ± 3
Fireclay brick	4·5 ± 1·0	6·0 ± 1·0	8·0 ± 1·0
Magnesite brick (electrically sintered)		42·7	
Magnesite brick	45 ± 5	36 ± 5	28 ± 5
Silica brick	6	8	10

*Note that values of the thermal conductivity, given in this form for convenience, are twelve times those for consistent units (per foot thickness).

refractory at higher temperatures. However, a knowledge of the strength of a brick is useful as indicating its resistance to rough handling during transport, and in certain positions when abrasion resistance is important, at the top of a blast furnace stack for example, a brick of high crushing strength is usually sought.

With increasing temperature the mechanical properties of a refractory may either increase or decrease. In fireclay products, the crushing strength is generally found to increase with temperature up to about 1,000° C. when the bond starts to become viscous and the crushing strength becomes closely dependent on the duration of the stress. Thus at high temperatures a firebrick will withstand high loads for a short period, but will show subsidence under much lighter loads if they are maintained.

SPALLING RESISTANCE

An important property of refractories in which the mechanical strength and elasticity have a controlling influence is the spalling resistance.

Thermal spalling, as distinct from physical and mechanical spalling, is caused by the development of stresses within the body of a brick consequent on changes in the temperature gradient. Calculation of the magnitude of these stresses and their comparison with the ultimate strength of the material at the temperature in question, leads to the following formulæ :—

On sudden heating : $$\text{spalling tendency} \propto \frac{\text{Coefficient of expansion}}{\text{max. shearing strain} \times \sqrt{\text{diffusivity}}}.$$

On sudden cooling : $$\text{spalling tendency} \propto \frac{\text{Coefficient of expansion}}{\text{max. tensile strength} \times \sqrt{\text{diffusivity}}}.$$

Examination of these formulæ show that to resist spalling a brick should have a low coefficient of expansion, high strength and high thermal diffusivity. These requirements are fulfilled by silicon carbide, sillimanite and most firebricks. Silica bricks have a low spalling resistance below 600° C. owing to the high coefficient of expansion up to this temperature ; above 600° C. the expansion becomes small and uniform and the spalling resistance increases. Magnesite bricks have a fairly high diffusivity and strength, but also have a high coefficient of expansion ; in general, therefore, magnesite bricks fall between silica bricks and firebricks as regards their resistance to thermal shock.

A most important factor to be remembered is that almost without exception spalling is most liable to occur at comparatively low temperatures. At such

temperatures the bond is glassy and will not yield to the expansion stresses ; as the temperature rises the bond becomes more deformable and will take up some of the strain. It follows that spalling chiefly occurs during the warming up and cooling down of furnaces ; particular care must be taken during the warming and cooling of any furnace lined with silica or magnesite bricks.

Dolomite bricks should not be exposed to the furnace atmosphere, since with these products a peculiar type of spalling occurs on exposure to high temperatures. This applies to very high temperature iron melting furnaces, but not to reheating furnace hearths.

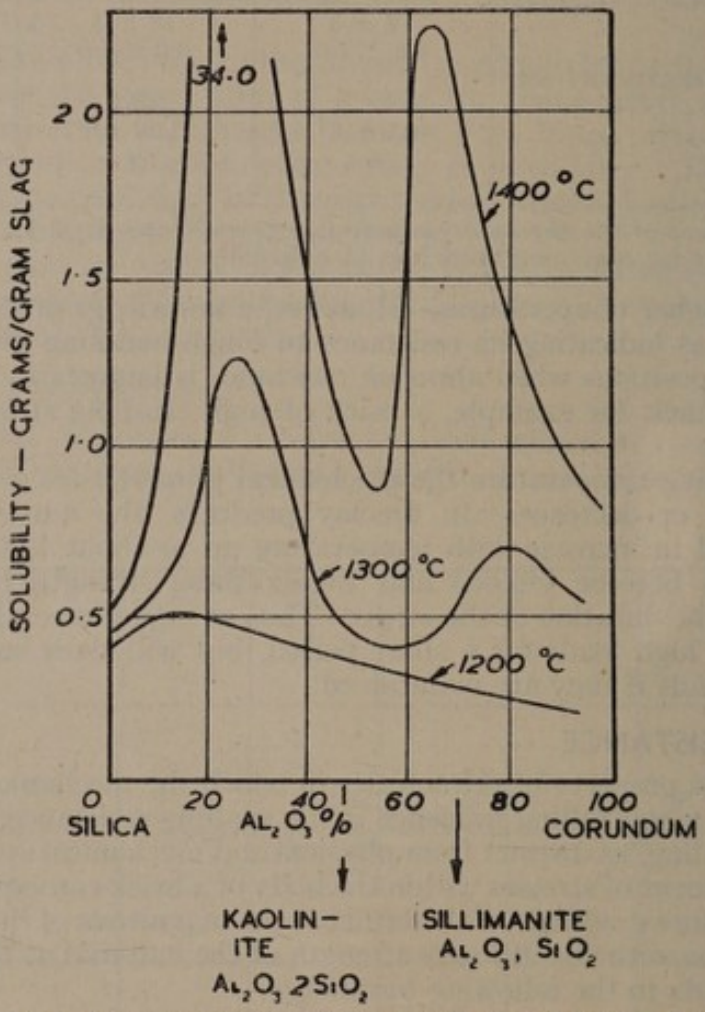


FIG. 188. Solubility of refractory materials in coal slag at 1,200°–1,400° C. Effect of temperature.
(H. R. Fehling, *J. Inst. Fuel*, XI, 454.)

If a furnace is to be used intermittently, firebricks, sillimanite, or silicon carbide bricks should be used wherever possible, the ultimate choice depending on the presence or absence of slag, and its nature, and on the cost.

THE ACTION OF SLAGS

It is probably unnecessary to emphasise the importance of the action of slags as a cause of refractory failure ; it has been estimated that this action influences the refractories in no less than 75 per cent. of industrial furnaces. The primary action of the slag is chemical solution of the brickwork, but ultimately a physical disruption of the slagged surface may take place. If the slag is in motion, or if it is carried as small particles by flame gases, erosion will occur in addition to solution.

Whichever type of slag action predominates in a given furnace, the life of the

lining will depend on the properties of the slag, the refractory, and the product of their interaction.

A most important factor is the fluidity and wetting power of the product of reaction. If a slag, though itself fluid at the temperature of operation, rapidly becomes viscous on taking brick into solution, then it may be anticipated that reaction will be slowed up, since further reaction can occur only by removal of, or diffusion through, the initial reacting layer. If, on the other hand, the product of reaction is itself fluid, reaction will probably be rapid. In so far as the fluidity is related to the degree of superheat above the liquid temperature, the probable fluidity of the product of reaction of a slag and a refractory can often be estimated by a consideration of the appropriate equilibrium diagram. Examples of the low melting-point compounds frequently formed as products of reaction are given in Table 107.

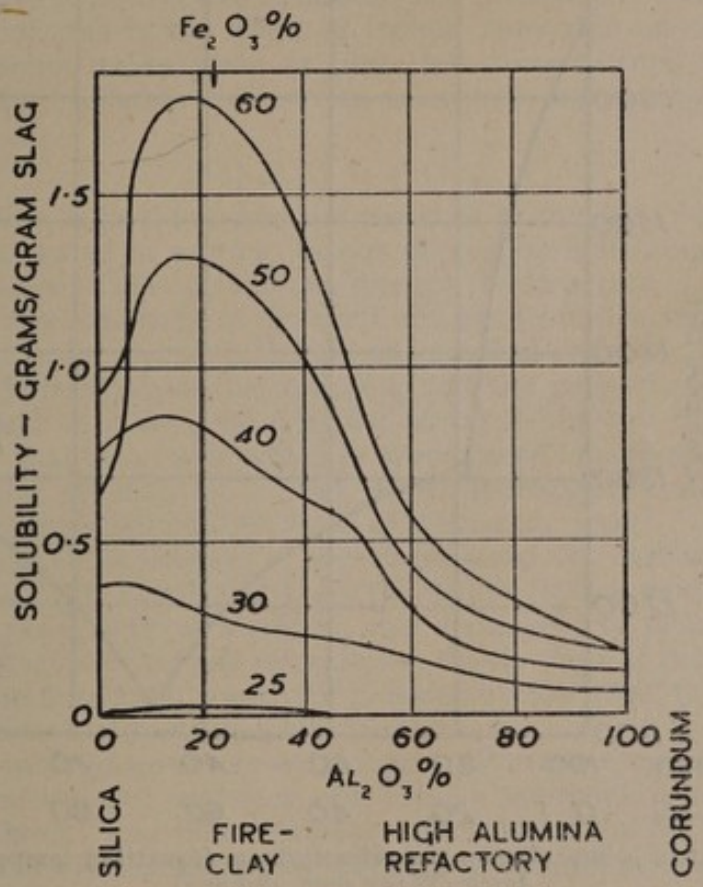


FIG. 189. Solubility of refractory materials in coal slag. Effect of iron oxide content.
(H. R. Fehling, *J. Inst. Fuel*, XI, 454.)

TABLE 107

Composition	Fusion temperature °C.
Eutectics between lime and silica	1,400–1,455
Lime-alumina-silica eutectics :—	
CaO, 23·3 ; Al_2O_3, 14·7 ; SiO_2, 62	1,170
CaO, 38 ; Al_2O_3, 20 ; SiO_2, 42	1,265
Lime-ferric oxide eutectic	1,203
Soda silica series : lowest eutectic	793
Ferrous oxide-silica series ; eutectic	1,180
Ferrous oxide-alumina-silica series :—	
Eutectic, FeO, 62 ; Al_2O_3, 3 ; SiO_2, 35	980
,, FeO, 68 ; Al_2O_3, 3 ; SiO_2, 29	1,002
,, FeO, 72 ; Al_2O_3, 3 ; SiO_2, 25	1,080

Coal ash is the one of most common corrodants of refractory materials. Typically, a coal ash consists of 24–45 per cent. SiO_2, 20–40 per cent. Al_2O_3, up to 30 per cent. Fe_2O_3, up to 10 per cent. CaO, up to 5 per cent. MgO, and 2–6 per cent. alkalies. Of the low-fusing constituents given in Table 108, compounds containing iron, lime and alkalies are most likely to arise from the action of coal ash.

Iron is particularly effective in promoting slagging under reducing conditions. The compounds of Fe_2O_3 with alumina and silica, in the absence of lime and alkalies, are not particularly fluid; when the Fe_2O_3 becomes reduced to FeO, however, as may occur in any furnace atmosphere under reducing conditions, reaction becomes more rapid and the product of the reaction has a low melting

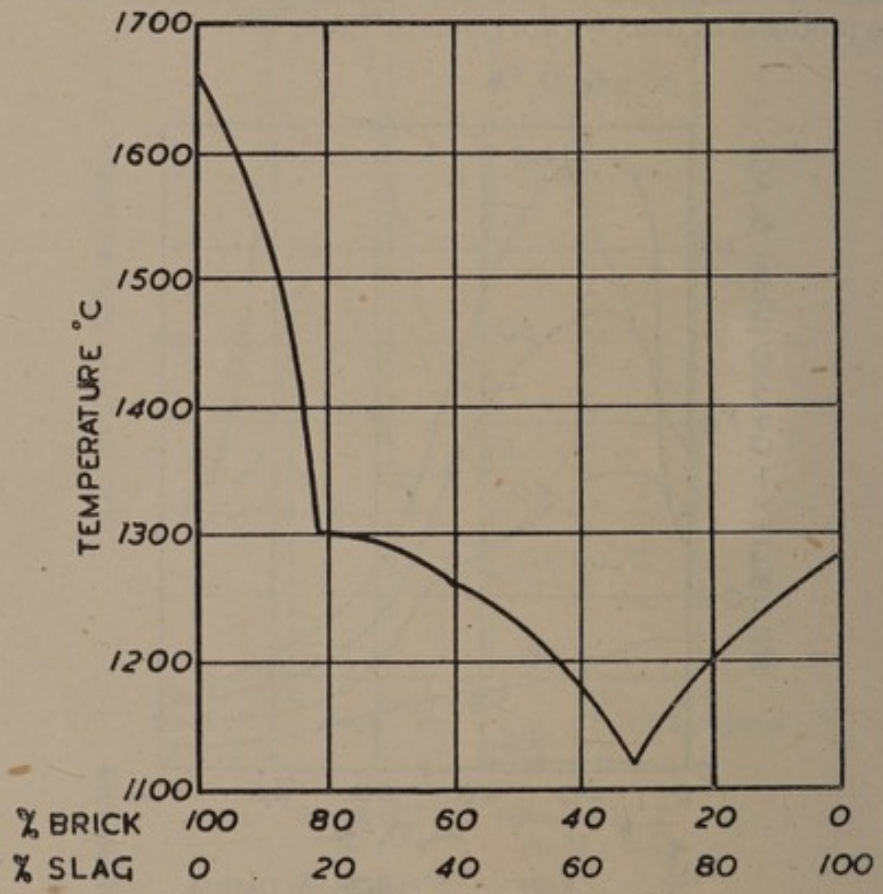


FIG. 190. Effect of a lime slag on the refractoriness (squatting temperature) of a fire-brick (33 per cent. alumina).

point. A coal ash containing important amounts of both lime and iron oxide, may prove particularly destructive towards a furnace lining. A salty coal is also very reactive, as is well illustrated by the action of such coals on coke oven linings and gas retorts.

The slagging action of iron compounds in a reducing atmosphere is illustrated in Figs. 188 and 189, which give the results of some recent experimental work (H. R. Fehling, *J. Inst. Fuel*, **11**, 453, 1938). These curves show the solubility of refractory material consisting of mixtures of silica and alumina in a coal ash at temperatures between 1,200° and 1,400° C., the ash having the composition: 45–50 per cent. SiO_2; 30–35 per cent. Al_2O_3; 15–20 per cent. Fe_2O_3; 5 per cent. CaO + MgO. From Fig. 189 will be noticed the increasing effect of higher percentages of iron, while in Fig. 188 it will be observed that increasing temperature has a very great effect in accelerating the corrosive action.

The effect of increasing temperature on the rate of slag attack is due to the decreased viscosity and to the increase in the rate of all chemical reactions at high temperatures. The following figures have been published showing this effect in a coal ash :—

Temperature (° C.) above softening point :—	0	50	200
Viscosity (poises)	1,000	200	40
Velocity of flow (centimetres per hour)	5·2	8·9	15·3

Fig. 190 shows the effect on fusion temperature of mixing a limey slag with a good quality firebrick.

THE ACTION OF GASES AND VAPOURS

Several of the gases and vapours likely to be present in a furnace atmosphere are known to have a destructive action on the brickwork. Perhaps the best known instance of this is the effect of carbon monoxide on certain fireclay bricks. The reaction takes place at quite low temperatures (450°–550° C.), and is due to the breakdown of carbon monoxide, into carbon and carbon dioxide, thus

$$2CO \rightleftharpoons C + CO_2.$$

This decomposition is facilitated in the presence of free iron oxide, so that if iron oxide is segregated in a brick, carbon deposition is liable to occur at the points of segregation. This carbon can disrupt the structure of the brick, and firebricks taken from the back of the shaft of a blast furnace, or from flues, are frequently found to be greyish in colour and cracked or friable.

Not all iron spots catalyse this reaction, and the proportion of iron oxide shown by chemical analysis is no criterion of the liability of the brick to disintegrate. Only that iron oxide which is uncombined is active, and a decision as to the probable stability of a brick in an atmosphere containing carbon monoxide can be made only on the basis of laboratory tests.

It has been found that dilution of carbon monoxide by carbon dioxide, as is usual in flue gases, considerably reduces the disintegrating tendency.

Hydrocarbon gases, such as methane, also decompose to yield carbon when brought into contact with certain refractories, chiefly those of the fireclay type ; with methane the maximum reactivity is found at 850°–900° C., but the temperature depends on the particular hydrocarbon.

The action of other gases is less well known. Water vapour has a damaging effect on basic refractories, and may also cause a reduction in the fusion temperature of firebricks and silica bricks ; accordingly careful and thorough drying out of all newly-built furnaces should be a routine procedure. Chlorine may be present in small amounts in a furnace atmosphere as a result of decomposition of the salt frequently present in coal ; the chlorine may attack iron compounds present in the furnace brickwork.

Alkalies, derived from the fuel or from the charge being heated, readily vaporise. In this form they can penetrate into the brickwork and react to form compounds of low melting point. Zinc, on the other hand, can vaporise and exert a physical action, since it may be deposited as metal in the pores and force the brick apart ; this effect can sometimes be detected in the iron blast furnace.

JOINTING CEMENTS

It is generally acknowledged that in many high temperature installations, the cement joints exert a considerable influence on the ultimate durability of the furnace. Consequently the properties of jointing cements demand rigid control.

There is a general consensus of opinion that the more important properties of refractory cements include the following: (1) refractoriness, (2) bonding strength, both dry and fired, (3) vitrification, (4) drying and firing shrinkages, (5) extent of reaction with the brickwork, (6) working properties, (7) screen analysis, (8) chemical analysis, (9) coefficient of thermal expansion, (10) resistance to slagging.

Several of these properties are evidently inter-related. Thus the bonding strength must be partially dependent on reaction with the brickwork. Good bonding strength is desirable, but excessive reaction must be avoided. The bonding strength also depends in some measure on the working properties; if the water is immediately taken up from a cement on applying it to a porous brick surface, there is less likelihood of the bond being firm than if the cement remains moist and can be worked into the surface pores on placing the next brick in position. The drying and firing shrinkage are evidently of considerable importance; if excessive, the joint will crack. The incorporation of too much plastic clay in a jointing cement is likely to cause trouble in this direction, yet the workability must be adequate. The relative amounts of clay and grog must evidently be carefully adjusted. The grading is also of importance.

The whole of any cement should pass a 40 B.S. sieve, both in the interests of thin joints and of easy working.

The chemical composition should in general be similar, but not necessarily identical, to that of the bricks; exceptions to this occur in the use of chrome cements for purposes other than the jointing or patching of chrome and chrome magnesite bricks. In silica cements similarity should go beyond chemical composition to mineralogical constitution; the need for this will be appreciated when it is remembered that a fired silica brick, well converted to tridymite and cristobalite, will have very different expansion characteristics from a cement made up from unfired quartzite and/or sand. The addition of a proportion of crushed, fired, silica brick to a silica cement has been found to increase the bonding strength quite considerably.

INSULATING REFRACTORIES

Thermal losses from high temperature installations may reach a high proportion of the total heat input, unless suitable insulation is provided. Of recent years increasing attention has been paid to the problems associated with the application of insulating materials to furnaces, kilns and similar plant, and also to the complementary problem of producing insulating materials to meet the requirements of furnace designers.

High-temperature insulating materials are here considered as those products suitable for use at temperatures above 600° C. This selected temperature is quite arbitrary. It eliminates products such as asbestos and slag wool, which are looked on rather as lagging material. Two methods of insulation are favoured. When the furnace is in use for extended periods, or when the conditions of temperature and atmosphere are very severe, a backing of insulating material is placed behind a dense refractory face. In other installations, where cleaner conditions or a lower temperature permit, and particularly in short-cycle intermittent furnaces, the insulating material can form the refractory lining. The commonest material for the first class of insulating brick is diatomaceous earth (diatomite), which can be used up to about 900° C.

In the second method of application an insulating firebrick, or, a better name, a refractory insulating brick, is used. Materials of this type are highly porous, refractory and have fair strength and spalling resistance; they can be used up to about 1,300° C. In general, they consist of fireclay, the high porosity being achieved either by the incorporation of a combustible in the raw mix, or by foaming, i.e. the production of innumerable small gas bubbles in the clay paste

before the latter solidifies. The former method is by far the more frequently adopted, owing to the greater manufacturing difficulties encountered in foaming.

Further information on insulating bricks is given later in this chapter.

THE SELECTION AND TESTING OF REFRACTORY MATERIALS

Correct selection of refractory materials can be made only on the basis of a full knowledge of the conditions of service and the properties of the available refractories. These properties can be evaluated fairly accurately in the laboratory. The testing methods which should be followed have been recently standardised (" Tentative Standard Methods for Testing Refractory Materials," B.R.R.A. Special Pub. No. 3, 1941). They consist of chemical analysis and the determination of true specific gravity, true and apparent porosity, bulk density, permeability, refractoriness and refractoriness-under-load, permanent linear change on reheating, cold crushing strength, reversible thermal expansion to 1,000° C., and resistance to carbon monoxide. Due attention should be given to the problem of sampling.

Testing methods are also available for determining the spalling resistance, the change in porosity and volume on reheating, and the reversible thermal expansion to 1,500° C., but these methods have not been standardised. Examination of a refractory by microscopical and X-ray methods may give valuable information on the constitution of a brick.

It is not necessary to carry out each of the above tests on any one brand of brick, but no one test will give all the information required. In testing a firebrick, for example, attention should be paid to the chemical composition, the refractoriness, and the refractoriness-under-load, the porosity and after-contraction ; if the firebrick were for use in the shaft of a blast furnace, say, the resistance to carbon monoxide and the crushing strength should also be determined. A knowledge of the permeability is useful if the brick is to be exposed to slags or noxious gases.

A slightly different series of tests would probably be used prior to the selection of a silica brick. Micro-examination and the determination of the true specific gravity are essential, since the mineralogical constitution is here of more importance than the chemical analysis, which can usually be omitted. The refractoriness-under-load is a useful check on quality, and the after-expansion should be determined if the brick contains much quartz (as indicated by the microscope or by a high specific gravity). The reversible thermal expansion should be known in order to make the correct allowance in the furnace structure for expansion of the brickwork.

In selecting basic refractories it is useful to know the chemical composition. For a magnesite or dolomite brick, further tests should be the refractoriness-under-load (the ordinary refractoriness of these products is invariably above 1,800° C., and need not be determined), the after-contraction, porosity and spalling resistance. A chrome or chrome magnesite brick should be tested in a similar manner to a magnesite product, but a further test should be added, namely, the determination of the bursting expansion when heated in contact with iron oxide ; this test is described in " Methods of Testing Basic Refractory Raw Materials and Products " (B.R.R.A. Special Pub. No. 1, 1940).

Turning to the definition of the service conditions, it is well known by all who are associated with the operation of any furnace just how many factors may influence the life of the lining. In many cases, a variation in these conditions, frequently accidental, may have a greater influence on the service obtained from the bricks than any variation in the properties of the bricks themselves. Some of the factors which should be considered are : the maximum temperature ; the load, especially the thrust in arches ; the nature of the slag, ash or gases ; the possibility of rapid temperature fluctuations ; the necessity

or otherwise for resistance to abrasion or impact ; the thermal properties and the standard of the building technique in particular in regard to the character of the joints and the jointing material.

A knowledge of these conditions, an appreciation of the general technology of refractory materials, and test data on available products, will serve as a foundation for scientific selection and ultimate economy both in refractories and fuel consumption.

As a general statement on selection, it may be noted that, in the sphere of fireclay products, the more aluminous materials are the more refractory, and, as a result of their chemical nature, are less reactive towards certain types of slag. Siliceous firebricks containing 75–92 per cent. SiO_2, on the other hand, show considerable volume stability at temperatures up to 1,350° C., or even higher, and, on this account, have a sphere of use in the carbonising and other industries. Silica bricks, characterised by high strength at elevated temperatures, are widely used in the arches of furnaces operating under severe conditions. However, the tendency of silica bricks to spall at low temperatures must be kept in mind when selecting this type of refractory for a particular usage ; since the abrupt expansion of silica occurs only at low temperatures, furnaces lined with this material may be subjected to wide temperature fluctuations above 600° C. without damage.

In positions demanding mechanical strength at high temperatures, but in which the slagging conditions preclude the use of silica, sillimanite refractories find a specialised use ; they are also widely employed in the glass industry.

Basic refractories, among which may be included magnesite, chrome magnesite and dolomite bricks, are used chiefly in the metallurgical industries. In basic steel making furnaces, for example, magnesite or dolomite bricks are used as a hearth lining and chrome and magnesite have found increasing use in the walls and ports.

INSULATING MATERIALS

The use of insulation to reduce heat transmission has been discussed in Chapter VIII. The rate at which heat is lost from a vessel or structure depends (other things being equal) upon the temperature of the exposed outer surface relative to the surroundings. Insulation is effected by providing a layer of material having a low heat conductivity between the internal hot surface of the vessel or furnace and the external surface, thus causing the temperature of the external surface to be materially reduced. As normally used, therefore, insulation reduces the escape of heat by keeping the outer surfaces cool. In refrigerating work, the insulation prevents the flow of heat from outside to the interior of the chamber.

The temperature to which the insulating material is raised in use is a matter of great importance. Insulating substances owe their property to a structure consisting of minute pores filled with air or gas which have in themselves a very low thermal conductivity. If the pores become partly filled up (as when the material becomes wet or compressed) or if the structure is disturbed (as when an insulating material settles into a solid block or is partly fused) the thermal conductivity is increased. Excessive heat affects all insulation adversely, but the temperatures to which the various insulating materials can be heated before this adverse effect occurs differ widely.

Clearly, therefore, the choice of an insulating material must depend upon the effectiveness with which it is required to operate and upon the temperature that it will withstand successfully.

It is usual to refer to the " insulation " of furnaces, and to the " lagging " of lower temperature plant such as steam pipes, because insulating material for

furnaces is frequently used in the form of a brick, whilst for low temperature work it is more often applied as a plastic mass when the appliance is hot.

CLASSIFICATION OF INSULATING MATERIALS

Insulators can be roughly divided into groups according to the temperature for which they can be used. A classification (J. S. F. Gard, *J. Inst. Fuel*, X, 223) is as follows :—

(1) *Low Temperature Range*—below 200° F.

Refrigeration and building construction, including :—

Refrigeration.
Transporters for ice, solid CO_2, ice cream, etc.
Cold storage.
Air conditioning.
Cool-water systems.
Hot-water systems.
Storage tanks.
Tank waggons.
Buildings.

The most commonly used materials are granulated cork, cork board, wood (ordinary and special), wood pulp, pulp boards, straw boards, grasses, charcoal, sponge rubber, mineral fibres, aluminium foil.

(2) *Medium Temperature Range*—200°–700° F.

Power station practice, steam raising and low temperature heating, including :—

Boilers.
Steam lines.
Steam accumulators.
Hot-air systems.
Drying plant.
Flue gas ducting.
Turbines.
Cylinders.
Tank waggons.

These most used are 85 per cent. magnesia-asbestos, other magnesia compositions, diatomaceous compositions, slag wool, spun glass, felted asbestos, bonded asbestos, asbestos mattresses, asbestos paper and mill-board in various manufactured forms, aluminium foil, other metal foils and sheets.

(3) *Moderately High Temperature Range*—600°–1,200° F.

Typical uses are superheated steam plant, ovens, stoves and diesel engine exhausts.

Here the choice is more limited and is restricted to asbestos (up to 850° F.), spun glass (up to 900° F.), kieselguhr, asbestos compositions, magnesia-kieselguhr-asbestos compositions, aluminium foil.

(4) *High Temperature Range*—1,000° F. upwards

Insulation for the highest temperature ranges applies generally to industrial furnace practice and carbonisation, typical applications being kilns, furnaces, gas-retort settings, regenerative stoves, blast furnaces, hot-blast mains, gas producers and coke ovens.

The choice of material is restricted generally to refractory insulating materials. These consist of two types, the highest grade of insulation usually being made from diatomaceous earth or other special material, and confined to external use, and the more recent type known as hot-face insulation, usually made from a refractory base, such as fireclay or silica. Hot-face insulators are used as

inside linings, being more robust and refractory than the former type, though of somewhat lower thermal insulation value.

The use of the insulating materials which fall under classes 3 and 4 has been discussed in Chapters XVIII and XIX. What is there said applies equally to boiler settings and furnaces.

CONDUCTIVITY OF INSULATORS

The value of the thermal conductivity of insulating materials is given in Table 108. This table is to be taken as a general guide only, since different specimens will show differences in thermal conductivity.

TABLE 108. ALL CONDUCTIVITIES, k, EXPRESSED AS B.TH.U. PER SQUARE FOOT PER HOUR PER °F. PER INCH THICKNESS

Material	0° C.	100° C.	400° C.	600° C.
Asbestos	1·05	1·35	—	1·65
Asbestos felt	—	0·47	0·67	—
Balsa wood	0·34	—	—	—
Balsam wool	0·25	—	—	—
Charcoal	0·35	—	—	—
Cork, granulated	0·30	—	—	—
Cork, slab	0·32	—	—	—
Cotton wool	0·29	—	—	—
Diatomaceous earth, high grade	—	—	—	0·7–0·65
Diatomaceous earth, low grade	—	—	—	1
Kieselguhr, loose	0·42	0·53	0·635	—
Kieselguhr, another specimen	0·56	—	—	—
85 per cent. magnesia	—	0·42	0·58	—
Spun glass	—	0·45	0·65	—
Expanded rubber	—	0·2–0·25	—	—
Rubber sponge	0·38	—	—	—
White dry sand	—	2·5	—	—
Sawdust	0·35	—	—	—
Hair felt	—	0·25–0·35	—	—
Yellow pine	—	1–1·15	—	—
Slag wool	0·29	0·35	—	1·33
Wool	2·6	—	—	—
Pipe coverings :—				
Plastic magnesia	—	0·40	0·64	0·92
Plastic asbestos	—	0·67	0·87	—
Glass fibre mattresses	—	0·33	0·7	—
Heavy materials	—	1·15	1·28	1·36

Insulating bricks over the mean range 0° C. to :—

	200° C.	400° C.	600° C.	800° C.
Insulating firebricks	1·74	1·89	2·03	2·18
Porous diatomaceous bricks	0·73	0·81	0·90	1·02
Solid diatomaceous bricks	1·07	1·2	1·36	1·45

INSULATING FIREBRICKS

When insulation is placed behind the refractory lining of a high temperature furnace, as was explained in Chapter XVIII, the interfacial temperature may rise beyond the value permissible for normal insulating materials. The best procedure is then to use a course of insulating firebrick through which the temperature gradient will be steep followed by a further course of normal insulating brick of lower conductivity, but not capable of withstanding equally

high temperatures. Table 109 gives typical figures for two grades of high temperature insulating material, with a firebrick for comparison.

TABLE 109. VALUES OF CONDUCTIVITY IN B.TH.U. PER SQUARE FOOT PER HOUR PER °F. PER INCH FOR SEVERAL MEAN TEMPERATURES

(The mean temperature is the arithmetic mean between the hot and cool face temperatures.)

Mean temp. °F.	Grade A brick	Grade B brick	Firebrick
300	0·84	—	6·23
500	1·02	1·64	6·69
600	1·085	1·75	6·92
700	1·14	1·86	7·15
800	1·18	1·94	7·38
900	1·20	2·03	7·61
1,000	1·22	2·11	7·84
1,100	1·23	2·18	8·07
1,200	1·24	2·24	8·30
1,300	1·25	2·30	8·53
1,400	1·26	2·36	8·76
1,500	—	2·41	8·99
1,600	—	2·46	9·22
Max. temp. at which material can be used :—			
°C.	870	1,250	—
°F.	1,600	2,280	—

The effect of using these bricks is shown in Fig. 191. In Fig. 191 (A) is shown the temperature gradient in a refractory wall of silica brick 13½ inches thick

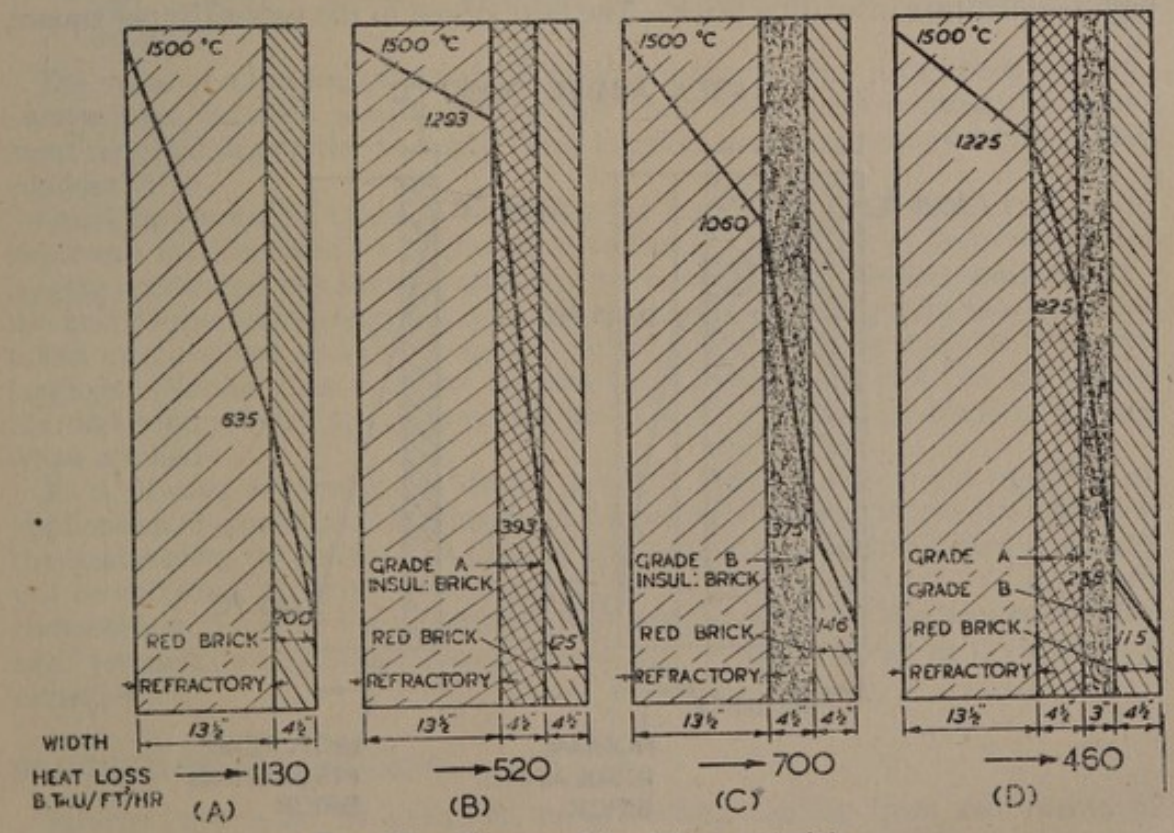


FIG. 191. The use of insulating materials.

backed by 4½ inches of red brick. In Fig. 191 (B) 4½ inches of grade A brick has been used between the red brick and the refractory. It will be noted that the interfacial temperature is above that which the insulation will stand. In Fig. 191 (C) there is the same arrangement as in Fig. 191 (B), but the grade A brick has been replaced by grade B. The result is not so satisfactory in reducing the temperature of the outer surface, but no part of the structure is overheated. Finally, in Fig. 191 (D) the insulating layer is divided into two parts using 4½ inches of grade B brick and 3 inches of grade A ; here the structure is sound and the heat conservation is satisfactory.

In addition to their use as intermediate courses as just shown, insulating firebricks are particularly useful as inside linings of intermittent furnaces (cf. Chapter XVIII). They are in this position limited to use in furnaces having an atmosphere free from corrosive dusts and slags.

The insulating firebrick increases the uniformity of temperature in a furnace for three reasons :—

(1) the heat radiation is high because of the high surface temperature ;
(2) the temperature drop along the path taken by the gases is proportional to the heat loss from the furnace walls ; by reducing the heat loss, the temperature drop is reduced ;
(3) the flow of heat parallel to the wall face and towards doors and other openings is reduced.

Hot-face insulating bricks are lighter than normal refractories, weighing perhaps one-quarter to one-third as much only. Moreover, the heat penetration into the brick is less, and the use of these bricks, as explained in Chapter XVIII, results in much less heat storage in the furnace walls. This feature of insulating firebricks is shown in Fig. 192. The shaded portion in this figure shows the extent to which the material of the wall is above 1,000° C. The weight of the wall for the full thickness shown, per square foot of superficial area, would be above 150 lb. for the composite wall and 30 lb. for the wall of high temperature insulating brick. The heat stored in the two walls per square

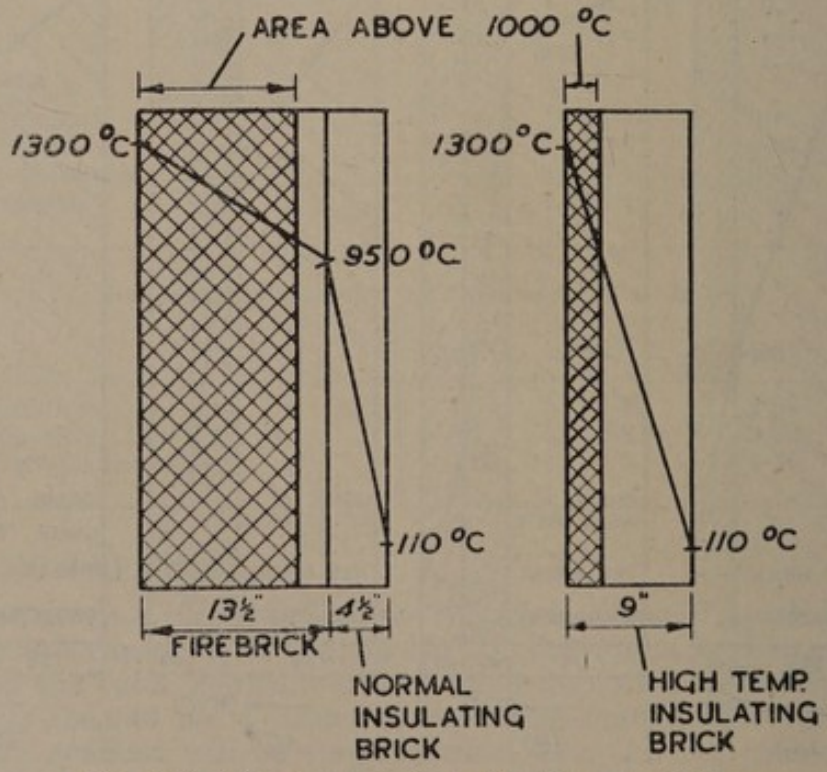


FIG. 192. The thermal effect of hot face insulation.

foot of superficial area would be approximately 75,000 B.Th.U. and 9,000 B.Th.U. respectively. The surface temperature is the same for the two walls, so that the external surface heat losses would be the same in the two instances. This comparison indicates the features of the new technique of hot-face insulation which has been described as " the most important contribution to greater furnace efficiency made in the last decade." It should be added that some of these bricks can be used up to 1,400° C.

The chief limitation of insulating firebrick is its lack of resistance to slags. The external structure being highly porous is readily penetrated and dissolved by fluid slag and although coatings have been tried, they have been of but little assistance since they cannot be made impervious. There have been instances of good service with very viscous slags, but in general these bricks can be used only under clean heat conditions.

A second limitation is the rather low resistance to abrasion which prevents these bricks from being used as a working hearth. For the same reason they cannot be recommended for positions where they will be scoured by dust-laden gases at high velocity, unless they are coated with a denser material. The resistance to heavy mechanical vibration is not so high as with firebrick. An example of this is forging furnaces adjacent to heavy hammers, for which it is advisable to use heavier and more rugged constructional materials.

LAGGING

The fundamental requirements of a lagging are :—

(*a*) The thermal conductivity must be low.
(*b*) It must be capable of adhering to the surface to which it is applied.
(*c*) When applied it must have a reasonable resistance to the weather, and to other agents causing deterioration.
(*d*) When required to operate intermittently, i.e. when the vessel is heated and cooled at frequent intervals, it should have a low heat storage capacity.

The value of a lagging material can only be assessed after it has been applied. Comparisons between samples in the raw state are useless because the treatment required in practice to apply these laggings may appreciably modify their conductivities.

Great care must be taken in installing the lagging in place since much of its value can be destroyed by improper installation. Clearly the first cost of the lagging material is not the sole criterion when considering the economic aspect ; the cost of the labour required to apply the lagging to the hot surface must be taken into account and this is usually appreciable and may vary for different laggings. Neither does it follow that the lagging material which has the lowest thermal conductivity as received will necessarily be the most efficient lagging when applied.

It is always advisable to obtain detailed instructions of the method of application of a particular lagging from the makers. Many factors can influence the heat-saving properties of a lagging and once it has been installed it should not be subjected to rough handling or other mechanical damage. A low-grade composition, however low its installation cost, is not generally economical for high temperature insulation, and should not be used except as a temporary expedient.

PROTECTION OF LAGGING

Special care must be exercised in protecting lagging from the effects of moisture and mechanical damage. From their general nature laggings are

porous, and consequently readily absorb moisture which greatly reduces their efficiency. In extreme conditions the heat loss arising from this cause may amount to ten times its value with dry lagging.

The cheapest method of protecting lagging is to wrap roof felting round it and to secure this with single strands of wire. There are several more expensive methods of protection, which seal the lagging from the atmosphere and at the same time provide it with a more adequate protection against mechanical damage. These are usually compounds of a bituminous nature.

PROPERTIES OF TYPICAL LAGGING MATERIALS

Slag Wool. This is a good insulant, and is comparatively cheap, but special care must be taken when installing it. It can be applied to hot surfaces having temperatures up to 600° C.

It is usually supplied by the makers in the form of a mattress 3 feet long. The width is such that it can just be wrapped once round the pipe. When the required thickness of the lagging is more than 2 inches, it is advisable to obtain this thickness by installing two layers of mattresses.

The mattresses are secured in position by tightly winding a wire round them. For pipes of less than 1 foot diameter, the distance between each turn of the wire should be of the order of 2 inches, but for larger pipes it is necessary to space the wire more closely. Care should be taken to ensure that the mattress is protected against moisture and mechanical damage, and that there are no large air spaces between it and the pipe. It is very important that a high grade of slag wool, free from impurities, should be used.

Magnesia. The cost of magnesia lagging is approximately the same as that of slag wool, but the installation is probably easier for large vessels and somewhat more difficult for small pipes. It is more rigid than slag wool, but should not be used on surfaces at a greater temperature than 300° C. This disadvantage can be overcome by interposing a thin layer of special cement between the magnesia and the hot surface. Magnesia lagging is obtained in the form of slabs, which are laid along the pipe and secured temporarily with strands of wire. A hard-setting cement-like compound is then applied to the outside, in order to secure the slabs permanently, and to fill the gaps between adjacent slabs.

Glass Wool. At temperatures below about 300° C. glass wool has a greater thermal resistance than slag wool or magnesia. It is more expensive and cannot be used for temperatures above 500° C.

It is obtained in the form of mattresses which can be secured in the same manner as slag wool, or in the form of rigid semi-circular sections, specially moulded to fit the pipe, which can be secured with metal bands.

Asbestos. There are many different types of asbestos used for lagging purposes, and consequently the available range of temperature varies widely. The cheaper types of asbestos usually have a higher thermal conductivity than either slag wool or magnesia and many of them are liable to absorb water. Furthermore, there are certain types of asbestos which can only be used for lagging at relatively low temperatures (of the order of 150°–250° C.). There are, however, grades of asbestos which can be used for higher temperatures (up to 600° C.).

Asbestos lagging is usually manufactured in the form of rigid semi-circular sections, specially moulded to fit the surface to which they can be secured with metal bands.

Temporary Forms of Lagging. Straw or felt is probably the cheapest lagging available and is useful when a temporary lagging is required. It can be wrapped round a pipe and secured with single strands of wire or string. It should not be

used for a surface hotter than 200° C. Exposure to the weather is likely to destroy its insulating properties.

LAGGING PIPES IN LOW PRESSURE, HOT WATER HEATING, AND HOT WATER SUPPLY SYSTEMS

Because of the comparatively low temperature met with in hot water central heating plants, it is too often erroneously assumed that lagging of pipe-work and vessels conveying or storing the heated water is not important. It is true, of course, that the higher the temperature the greater the loss from unlagged pipes, but even with comparatively low temperature systems normally used for central heating, lagging of pipes and boilers is an exceedingly good investment.

The economies of insulation for these lower temperatures can be shown by considering an installation comprising small pipes of 1 inch diameter, conveying hot water at an average temperature of 140° F. over 625 feet in total length, the surrounding air being at 65° F. If a lagging is used consisting of 85 per cent. magnesia of stated thickness, plus ½ inch of hard-setting composition, it will achieve a saving of 25,000 B.Th.U. per hour.

HEAT SAVING BY LAGGING STEAM PIPES

The magnitude of the losses which can take place from steam pipes may be indicated by an example. If a 5-inch steam pipe having a surface temperature of 200° C. is unlagged for a length of 200 feet, it may waste 9 lb. of coal per hour or 35 tons per year. If the same piping had a surface temperature of 360° C. (680° F.) the corresponding wastage of coal would be about 70 tons per year. As the temperature of a surface increases so the heat loss increases, and therefore the thickness of lagging or the efficiency of the insulation must be stepped up correspondingly.

It follows, therefore, that because a pipe is lagged it cannot always be assumed to be efficiently lagged. Many hot surfaces which are lagged are inadequately insulated and some preventable loss is taking place. Some indication of the thickness of lagging required in different temperatures and the heat saved by lagging is given in Table 110.

TABLE 110

Diam. of main	Temp. of surface to be lagged		Thickness of lagging	Heat loss/ft. length—B.Th.U./hr.		Efficiency of lagging	Heat lost/ft.² B.Th.U./hr.	
	° C.	° F.	In.	Unlagged	Lagged	Per cent.	Unlagged	Lagged
4 in.	80	176	1½	275	65	76·4	233	55
	160	320	2	885	126	85·7	750	107
	240	464	2¼	1,860	179	88·9	1,576	152
	320	608	2½	3,230	207	92·5	2,740	182
	400	752	3	5,220	282	94·4	4,420	240
8 in.	80	176	1½	520	110	79·0	233	50
	160	320	2	1,670	210	87·4	750	95
	240	464	2¼	3,500	300	91·5	1,576	135
	320	608	2½	6,100	392	93·5	2,740	177
	400	752	3	9,800	440	95·6	4,420	198

For boiler and steam work generally all hot surfaces should be lagged. This applies to valves, flanges, etc., as well as straight runs of pipe, to boiler fronts, storage tanks and so forth. Boiler settings should be insulated. Ancillary apparatus, valves, dampers, etc., should be placed so that there is no need to walk upon the insulation, e.g. upon the boiler tops, to reach them.

CONSTRUCTION

(*a*) Methods of building refractory structures.

Furnace bricklaying is a trade in itself and should be entrusted to those accustomed to the behaviour of brickwork under furnace conditions. Brickwork is essentially a mass made up of small individual units, and the method of placing these units and their relative position one to another are of considerable importance to the stability of the structure. The bricks are laid in a series of horizontal courses, and the problem is to find the best bond for the particular thickness of wall and the furnace conditions.

For ordinary work the standard size brick is $9 \times 4\frac{1}{2} \times 3$ inches, and the methods of bonding given below are based on this unit. The use of smaller bricks, e.g. $9 \times 4\frac{1}{2} \times 2\frac{1}{2}$ inches, increases the cost per cubic yard and also the cost of the bricklaying.

Stretcher Bond. This is applicable only to a $4\frac{1}{2}$-inch wall, and is mainly used in small furnace construction (Fig. 193).

Header Bond. This gives perfect bonding in a 9-inch wall, and is the best arrangement for high temperatures, especially if the load is high on the brickwork ; the load can be carried by the cooler ends of the brick where the temperature is lower (Fig. 194). For walls thicker than 9 inches the combination of some stretcher courses with the headers is necessary to get a tie or bond throughout the thickness of the wall. Header bond, however, with its advantages can still be retained in a $13\frac{1}{2}$-inch wall by using $9 \times 4\frac{1}{2} \times 3$-inch headers on the hot face and $13\frac{1}{2} \times 4\frac{1}{2} \times 3$-inch headers every third or fourth course as a tie. Alternatively, header bond can be used entirely for a $13\frac{1}{2}$-inch wall by using $13\frac{1}{2} \times 6 \times 3$-inch bricks.

English Bond. Alternate courses of headers and stretchers (Fig. 195). This is the most common bond for $13\frac{1}{2}$-inch walls, and is also used for thicker walls. It is necessary to use a soap or pup brick, $9 \times 3 \times 2\frac{1}{4}$ inches, in starting off certain courses to get the correct bond. The use of soaps eliminates cutting.

A common variation of the English bond is to have two or three header courses on the hot face and one stretcher (Fig. 196), and for most furnace work this is better than alternate header and stretcher.

Dutch Bond. This is similar to English bond (Fig. 197), but gives even better bonding. The alternate stretcher courses are not coincident and this makes it less likely to have several vertical joints coinciding in a long wall where the bricks may gradually run out of the bond.

Arches. Firebrick arches are usually bonded (Fig. 198). For building 9-inch arches where the conditions are severe and the end of the arch is exposed, as in the stoker arches of water-tube boilers, the use of a special bonder brick, $9 \times 6\frac{3}{4} \times 3$ inches, for breaking bond eliminates small cut bricks and gives a much stronger arch.

In firebrick arches up to 12 inches thick, one course is usually employed, but for an 18-inch arch, two rings of 9-inch bricks are more satisfactory.

Arches constructed of silica bricks are more often built in rings (Fig. 199). The thermal expansion of silica bricks is about 66 per cent. greater than that of firebricks, and most of it occurs suddenly at a low temperature, below 300° C. The building in separate rings may give greater flexibility in providing for the thermal expansion of the heating up of a large furnace.

Silica arches are often built dry, without the use of jointing cement. This practice probably originated in the difficulty formerly experienced in getting a satisfactory cement. Silica is a non-plastic material, and when fireclay is added to give plasticity, a eutectic mixture is formed with a much lower melting point and results in the cement being unsatisfactory. When an arch is built dry the bricks should fit closely, otherwise stresses are set up which

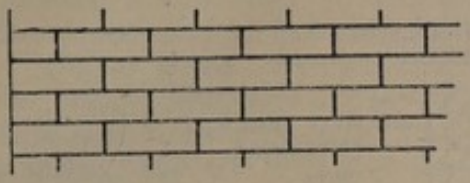


FIG. 193. Stretcher bond.

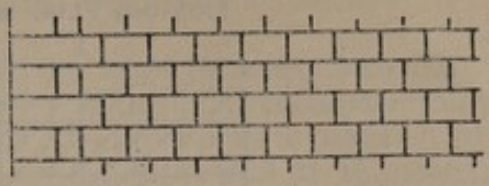


FIG. 194. Header bond.

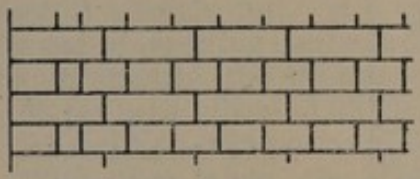


FIG. 195. English bond.

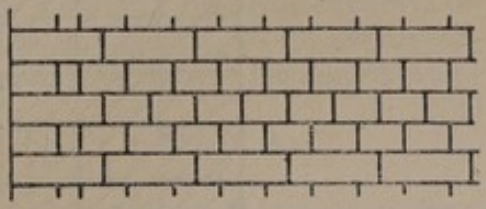


FIG. 196. Modified English bond.



FIG. 197. Dutch bond.

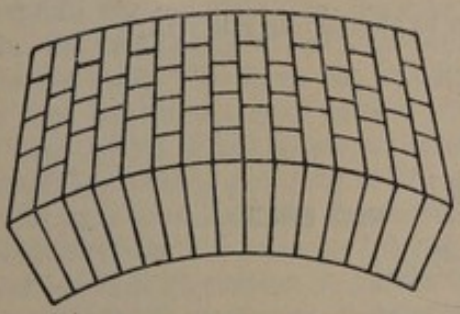


FIG. 198. Bonded arch.

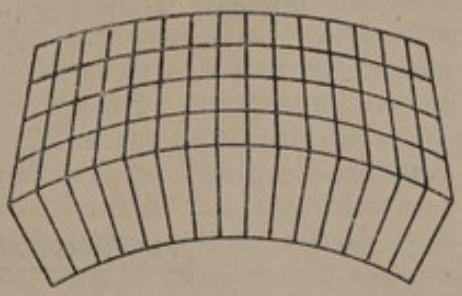
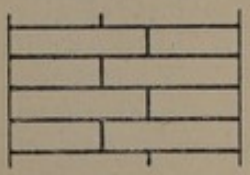
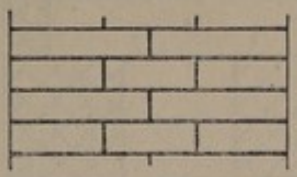


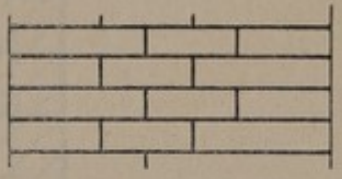
FIG. 199. Ring arch.



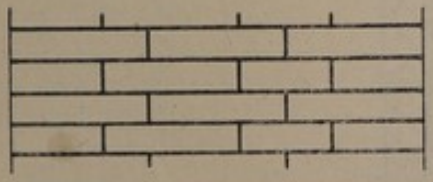
22½-inch wall.



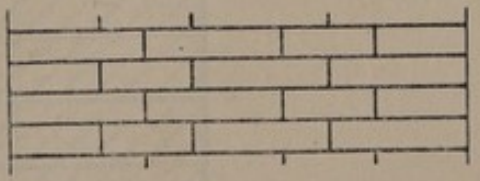
27-inch wall.



31½-inch wall.



40½-inch wall.



45-inch wall.

FIG. 200. Methods of bonding walls of various thicknesses

FIGS. 193–200. Methods of bonding bricks.

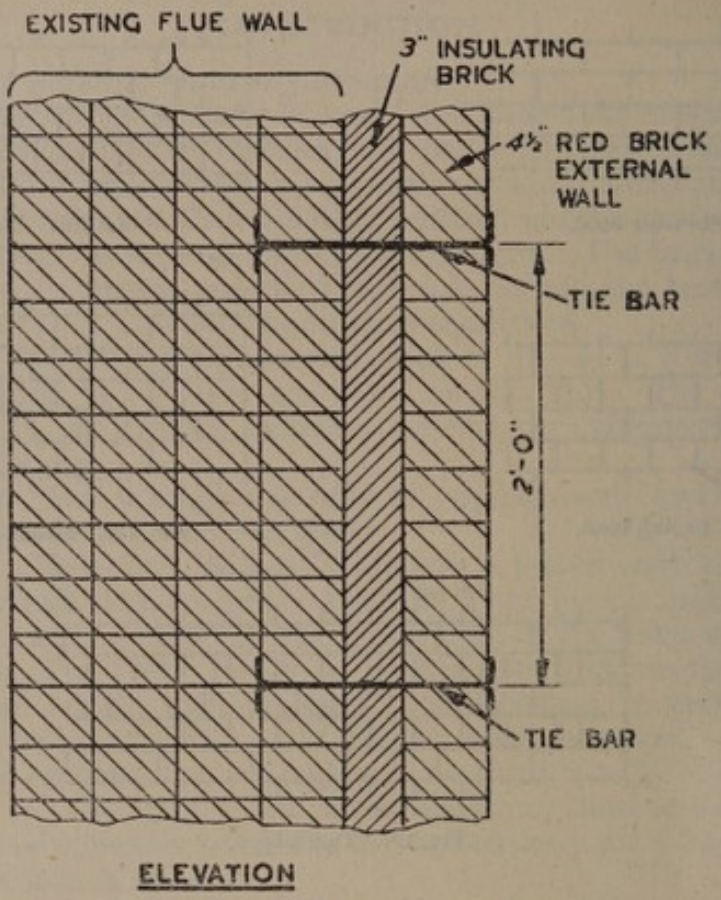


FIG. 201. Method of attaching insulation to an existing flue wall.

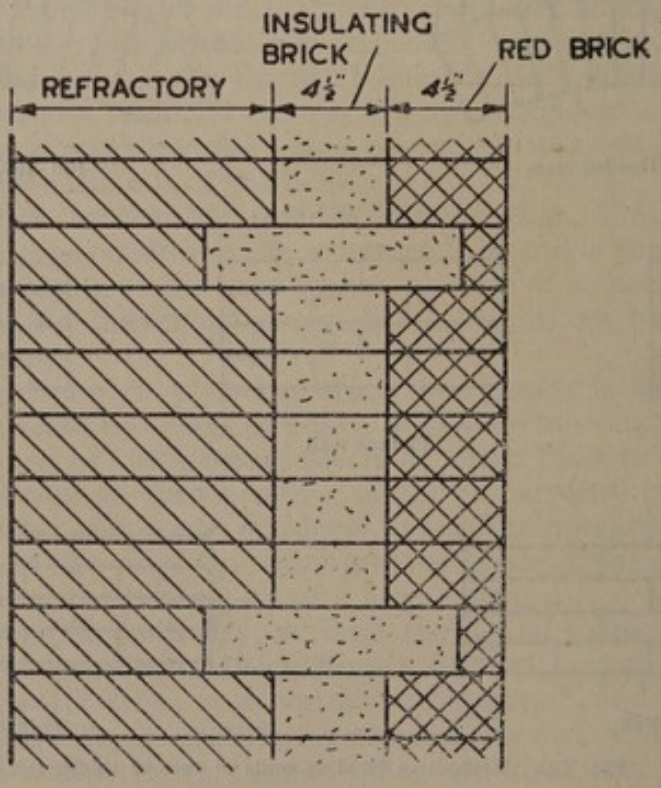


FIG. 202. Method of building in insulation.

may cause cracking. It is essential to adopt the ring construction to ensure the closest contact between the bricks.

Circular Walls. For building shaft furnaces, e.g. blast furnaces, lime kilns, etc., where conditions are severe and the thickness of the lining is considerable, improved bond is obtained by the adoption of 13½-inch bricks in addition to 9 inches, and the width of the bricks is then 6 inches instead of 4½ inches. The usual methods of bonding are shown in Fig. 200.

Allowance for Expansion. Experience has shown that in brickwork the individual bricks do not slide over each other, but that the wall expands in mass as a single unit. It is now customary to leave clear spaces for expansion instead of inserting cardboard or wood as was formerly done. Roughly speaking, in practice, the allowance required is :—

95 per cent. silica—1·25 per cent linear (3 inches on a 20-foot wall).

Firebricks—0·75 per cent. linear (1·8 inch on a 20-foot wall).

With silica bricks, the bulk of the expansion takes place at a temperature before the wood has had an opportunity to burn out. Thermal expansion must take place, and, if suitable provision is not made for it, enormous stresses are built up and serious damage may result.

Joints should always be kept as thin as possible no matter what method of bonding is adopted.

(*b*) *Construction of Insulating Walls.* Insulation of existing high temperature furnaces, as has been indicated previously in Chapter XVIII, should be undertaken only with expert advice on account of the increased temperature occasioned in the brickwork on the furnace, or hot, side.

Existing lower temperature furnaces, including boiler settings, can be insulated by the method shown in Fig. 201. A 4½-inch brick wall is built outside the existing one, leaving a suitable gap, 3 inches or 4½ inches between the old and the new walls in which the insulating bricks can be inserted as the wall is built. To keep the outside wall rigid, it is necessary to incorporate one or two buttresses, and further strength can be obtained by fixing a number of T-bars about 2 feet apart, as shown in Fig. 201. After the insulating slabs are fixed, periodical pointing of the external brickwork will be found unnecessary.

In new installations, insulating bricks should be built in the side walls and protected on the outside by red brick. One method of building a composite wall of this character is shown in Fig. 202. The tops of flues should also be covered with insulating bricks in the form of slabs 12 × 12 × 3 inches. This should be well tarred, as much damage is caused to the insulation by men walking about during repairs or to get at valves.

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CHAPTER XXIII

UTILISATION OF STEAM FOR PROCESS AND HEATING

The Advantages of steam—Heat transfer at constant temperature—High heat content—Water cheap and plentiful—Clean and odourless—Heat re-usable—Combined power and heating—Pressure and temperature—Latent heat—Sensible heat—Temperature control—Superheated steam—Wire drawing—Superheat and steam distribution—De-superheating—Heat and temperature—Heat transfer—Temperature difference—Heating surface—Heat resistant films—Air removal—Condensate film—Dirt and scale—Stagnant material—Steam circulation—High pressure hot water—Condensate removal and handling—Steam traps—Group trapping—Pipe draining—Vacuum draining—Condensate return—Lifting traps—Flash—Flash collection—Blow-down flash—Flash cooling—Re-use of latent heat—Multiple effect evaporation—Peak steam loads, causes and cures—Accumulators—Distillation—Reducing the work to be done—Reducing obvious losses.

I. INTRODUCTION

THE properties and general methods of utilisation of saturated and superheated steam have been discussed in Chapter VII. In this present chapter it is proposed to extend the information given and to apply it to industrial heating processes for which steam is used. In the generation of steam a great deal of fuel is consumed. In previous chapters the importance of securing a high efficiency in the operation of steam generation has been emphasised. Unless the steam is used efficiently, however, the care that is bestowed upon steam generation will be dissipated elsewhere in the works.

The utilisation of steam for general industrial purposes involves many different applications and a variety of processes in which some heat is inevitably lost. But although each industry has problems of application peculiar to its own processes of manufacture and the machinery concerned, there are basic considerations common to all which have to be met wherever steam is used for heating purposes.

In a factory the processes and their products are necessarily the primary consideration. Deficiencies in quality or rate of output are carefully watched and readily observable, whereas the efficiency of steam utilisation and its cost is often neglected. Even where the cost of steam raising in the boiler house comes under critical review, the waste in the subsequent use of the steam is often overlooked, despite its direct effect on the consumption of fuel.

Hence it is perhaps the exception rather than the rule to find in factories that the engineer who is responsible for the production of steam has jurisdiction over its utilisation. He merely comes under criticism if the steam supply is not ample, regardless of whether it is being used wastefully or economically. This is wrong, for, whatever may be the cause, a great and often easily avoidable waste of fuel is frequently associated with commercial processes.

The era of cheap coal which industry has enjoyed in this country for the last century has probably passed for ever, and higher prices should engender a greater incentive to use coal more effectively. The cost of fuel in the manufactured product must be proportionately higher post-war than pre-war, so that the effect of more efficient utilisation must stand industry in good stead for its post-war development.

The practical use of steam heat in the factory is so wide a subject that common principles only can be here discussed. Each industry has its own specialised methods and machinery, dependent upon the character and requirements of its products, and uses steam heat in a manner suited to its processes. Notwithstanding the great diversity of application and the specialised machinery used, the principles of steam heat utilisation are reasonably common to all, as are the possibilities of fuel economies based on those principles.

Of all known media for distributing and applying heat for manufacturing and industrial purposes, and also for the heating of buildings, steam is the most widely used. There are admittedly practical limitations to the temperatures that can be attained by the use of steam heat. In practice, however, apart from power production, steam is used for purposes which do not call for temperatures above 400° F.

II. THE ADVANTAGES OF STEAM

Steam is used for heating and process work as a carrier for heat: it is a method of conveying the heat from the coal which is being burned in the boiler furnace to the place where the heat is to be used.

Steam is chosen as the conveyor of heat for several reasons:—

(*a*) It gives up its heat at constant temperature.
(*b*) It has a very high heat content.
(*c*) It is generated from water which is cheap and plentiful.
(*d*) It is clean, odourless and tasteless.
(*e*) The heat in steam can often be used again and again.
(*f*) It can generate power and can then be used for heating.

These six points will now be briefly discussed. The first four points cannot be sharply divided—these qualities are interdependent.

(*a*) *Steam gives up its Heat at Constant Temperature*

This property is exceedingly important. It greatly simplifies plant design because counterflow arrangements are not needed. It gives complete control of heating operations, and enables a heating operation to be exactly repeated at any time. The reason is that when steam condenses by giving up its latent heat to the heating surface on to which it is fed, it does so at constant temperature, as explained in Chapter VII.

(*b*) *Steam has a very High Heat Content*

This means that a comparatively small pipe can carry a great amount of heat. A hot gas or a hot liquid can only carry much heat if at a high temperature, but high temperature may be quite unsuitable for some processes. Steam at low temperature contains about twenty-five times as much heat as the same weight of air or flue gas at the same temperature, and whereas the gas or air must drop in temperature to give up any heat, the steam can give up five-sixths of its heat without any drop in temperature. Water could be, and often is, used, but again it must suffer a temperature drop and to keep the drop reasonably small very large quantities of hot water must be circulated. This useful property arises from the high latent heat in the steam.

(*c*) *Steam is Generated from Water which is cheap and plentiful*

To compete with steam any competing material must be one which can be vaporised and condensed at a temperature approximating that of the boiling-point of water. There is no such material that approaches steam in the amount of latent heat it contains.

(*d*) *Steam is Clean, Odourless and Tasteless*

This is important in many industries.

(*e*) *The Heat in Steam can often be used again and again*

When steam is used for boiling or evaporating watery liquids most of the input heat is given off in the output vapour. A number of ways in which the heat in such vapour can be re-used are discussed in Section VII.

(f) Steam can Generate Power first and then be used for Heating

This subject involves considerations of entropy, and the calculation of heat drop and wetness at exhaust. It is impossible to do more than touch the fringe of the subject in this book, but some notes on the methods of calculation will be found in Appendix III.

Conditions are particularly favourable for employing this system where there is a steady demand for steam at a low or moderate pressure for factory heating or process work. The conditions are less favourable where the steam for heating is required at pressures approaching the normal working pressure of the boiler plant; for example, where heating steam at 100 lb. pressure is needed which must be generated from a Lancashire boiler.

Nevertheless, even when steam at high pressures is principally required, there is often a considerable demand for steam at low pressure for some purposes, e.g. factory space heating, and where steam is generated at full boiler pressure for power purposes and supplied by the same boiler plant for low pressure heating by passing it through reducing valves.

It has been shown in Chapter VII that with steam engines and turbines, the higher the initial pressure and the lower the exhaust pressure, the greater is the power produced from a given quantity of steam; hence the use of the condenser in power plants. In the most efficient power stations, however, most of the heat imparted to the steam is lost in the cooling water, the amount utilised by the prime mover being at most 35 per cent. This figure compares very badly with 80 per cent. utilisation of the heat in steam at 10 lb. pressure secured when it is used for heating purposes.

The principle of combined power production and heating is, broadly, that of discharging the engine or turbine exhaust steam to the heating system, which then virtually serves as a condenser.

The additional heat units required to raise the steam pressure from 10 lb. (gauge) to, say, 175 lb., amount to only 38 B.Th.U., or about 3¼ per cent. The use of that additional pressure for power production, however, leaves the bulk of the heat still in the steam if it exhausts at 10 lb. pressure, and this can be used for heating.

When the demand for power is in balance with the demand for heating steam, the heat utilised by this combined operation can well be 70 per cent. of the heat in the fuel used. As another indication that the power obtainable is fairly substantial it may be stated that with 10,000 lb. of steam per hour at 200 lb. pressure, dry, exhausting at 20 lb. gauge pressure, about 250 kW should be available, the figure being considerably increased if the steam is superheated.

The saving in any given conditions will depend upon the initial steam pressure, the proportion of the exhaust steam which can be used and the temperature and corresponding pressure at which it is required for the heating process. Obviously the bigger the difference between the two pressures, the greater will be the saving if all the exhaust steam can be used. Some typical examples from a Lancashire boiler capable of supplying 7,500 lb. of dry steam per hour are:—

Steam pressure lb./sq. in. gauge	Back pressure lb./sq. in. gauge	Temp. of exhaust ° F.	Kilowatts
95	5	228	150
115	10	240	150
150	15	250	150
200	30	274	150

The desirability of keeping the pressure of process heating steam down to the lowest effective pressure will be discussed later; in many factories a reduction of existing pressures would be economical in heat utilisation. But

the additional bulk of the steam at lower pressure has to be considered in existing pipe-lines and plant. Where the steam is used in pipe circuits for factory heating and similar purposes, the economy to be effected by combined power and heating might often justify the cost of installing a system of larger pipes to deal with the low pressure steam.

A calculation of the possible savings by this system is as follows :—

Assume a factory requires about 50,000 lb. steam per hour and needs about 1,200 kW of electricity.

The total heat in steam at 200 lb. per square inch absolute	1,198 B.Th.U.
Adiabatic heat drop (see Appendix III) from 200 lb. to 30 lb. per square inch absolute	143 ,,
Turbine efficiency—estimated	65 per cent.
Net heat drop	93 B.Th.U.
As 1 kilowatt hour is equivalent to 3,413 B.Th.U., the power generated per lb. steam will be	·0272 kilowatt hour
50,000 lb. steam per hour will give (with 94 per cent. generator efficiency)	1,280 kilowatts
With a net heat drop of 93 B.Th.U. the wetness of the exhaust steam will be	6·2 per cent.
50,000 lb., 200 lb. steam will yield at exhaust at 30 lb./sq. in. absolute	47,000 lb.

Comparisons can now be made. It will be assumed that 47,000 lb. of 30 lb. steam will meet the process needs. As 30 lb. steam will require 3 per cent. less heat to be put into it than into 200 lb. steam, it will be assumed that the evaporation per lb. of coal will be 3 per cent. better at the lower pressure.

Case 1. *Combined Power and Heating*

Steam raised at a pressure of	200 lb. per square inch absolute
Steam per hour	50,000 lb.
Coal burned at 8 lb. per lb. evap.	6,250 lb. per hour
Coal burned per annum (3 shifts—50 weeks)	19,530 tons

Case 2. *Process Steam Made, Power Bought*

Steam raised at pressure of	30 lb. per square inch absolute
Steam per hour for process only	47,000 lb.
Coal burned at 8·25 lb. per lb. evap.	5,690 lb. per hour
Coal burned in power station for 1,280 kilowatt hours at 1½ lb. per kilowatt hour	1,920 lb. per hour
Total coal burned per annum	23,750 tons

Case 3. *Process Steam and Power Generated in Factory in Separate Plants*

Process steam at 30 lb. Power steam at 200 lb. to condense at 26 inches vac.

Coal burned for process (as in Case 2)	5,690 lb. per hour
Adiabatic heat drop, 200 lb. to 26 inches	266 B.Th.U.
Net heat drop at 67 per cent. efficiency	180 B.Th.U.
Steam required per kilowatt hour	19 lb.
Total steam for 1,280 kilowatts	24,300 lb. per hour
Coal burned at 8 lb. per lb. evap.	3,040 lb. per hour
Total coal burned	8,730 lb. per hour
Total coal burned per annum	27,280 tons

Case 1 shows an annual saving of 7,750 tons of coal over Case 3 and 4,220 tons over Case 2.

It is not necessarily economical to generate power in a factory. It is only economical if most of the exhaust steam can be utilised. An arrangement which is very flexible and which is used in many factories is the pass-out system. Instead of exhausting the whole of the steam at process pressure, part only of the steam is withdrawn or passed out from the engine or turbine to suit process requirements, passing the remaining steam, to suit power requirements, through the low pressure end of the prime mover to the condenser.

III. STEAM PRESSURES AND TEMPERATURES

(*a*) LATENT HEAT AND SENSIBLE HEAT

The formation of steam from water is described in Chapter VII where sensible heat, latent heat and superheat are discussed. Fig. 203 shows in a striking manner the distribution of heat in low and moderate pressure steam. There are two properties that strike the eye immediately. The first is that the total heat content of steam rises relatively little with increasing pressure. The second is that the sensible heat (the heat left in the condensate when steam condenses) rises materially with increasing pressure. As a consequence of these properties it follows that latent heat decreases markedly with increase of pressure.

When steam is used inside a heating surface, coil, pipe or jacket, the original heat in the steam is split into two parts: (1) the latent heat which is transferred to the material being heated when the steam condenses; (2) the sensible heat which is retained in the condensate. The condensate generally gives up little or none of its sensible heat. The heat in the condensate at any particular pressure is greater than the heat in the condensate at lower pressure. If, therefore, the pressure on the condensate is reduced, the excess heat causes some heat to be generated as "flash" or "self evaporation." Unless means are provided for collecting this flash steam all the heat flashed off is lost when high pressure condensate is discharged from the trap into the condensate tank or "hot well."

Clearly the lower the pressure that can be used, the less heat will be given up in flash. Clearly also, from Fig. 203, the lower the pressure the greater the latent or heating heat.

This is universally true when steam is used on a heating surface. When steam is directly blown into the material being heated, the whole of the heat, latent and sensible, should be transferred to the product, and at moderate pressures the higher the pressure the greater is the total heat in each pound of steam. It will be shown, however, that this is very often not true in practice.

(*b*) TEMPERATURE CONTROL

In evaporators or boiling vessels the temperature of the boiling product is quite independent of the temperature of the heating steam. The boiling point of the product depends only on its own properties and on the pressure under which it is boiling. The water in a copper vat can be made to boil by means of a submerged coil carrying steam at 5 lb. per square inch pressure at 228° F., or it can be heated by a fire having a temperature of over 2,000° F. The water still boils at 212° F., but it will boil away much more rapidly when heat is supplied at high temperature than at low temperature.

When steam is used to heat a stationary product the temperature required fixes the pressure of the steam supply. Thus the vulcanisation of rubber or the

moulding of plastics demand certain definite temperatures which, if not reached or if appreciably exceeded, result in spoiled material.

When steam is used to heat a flowing product the temperature of the steam

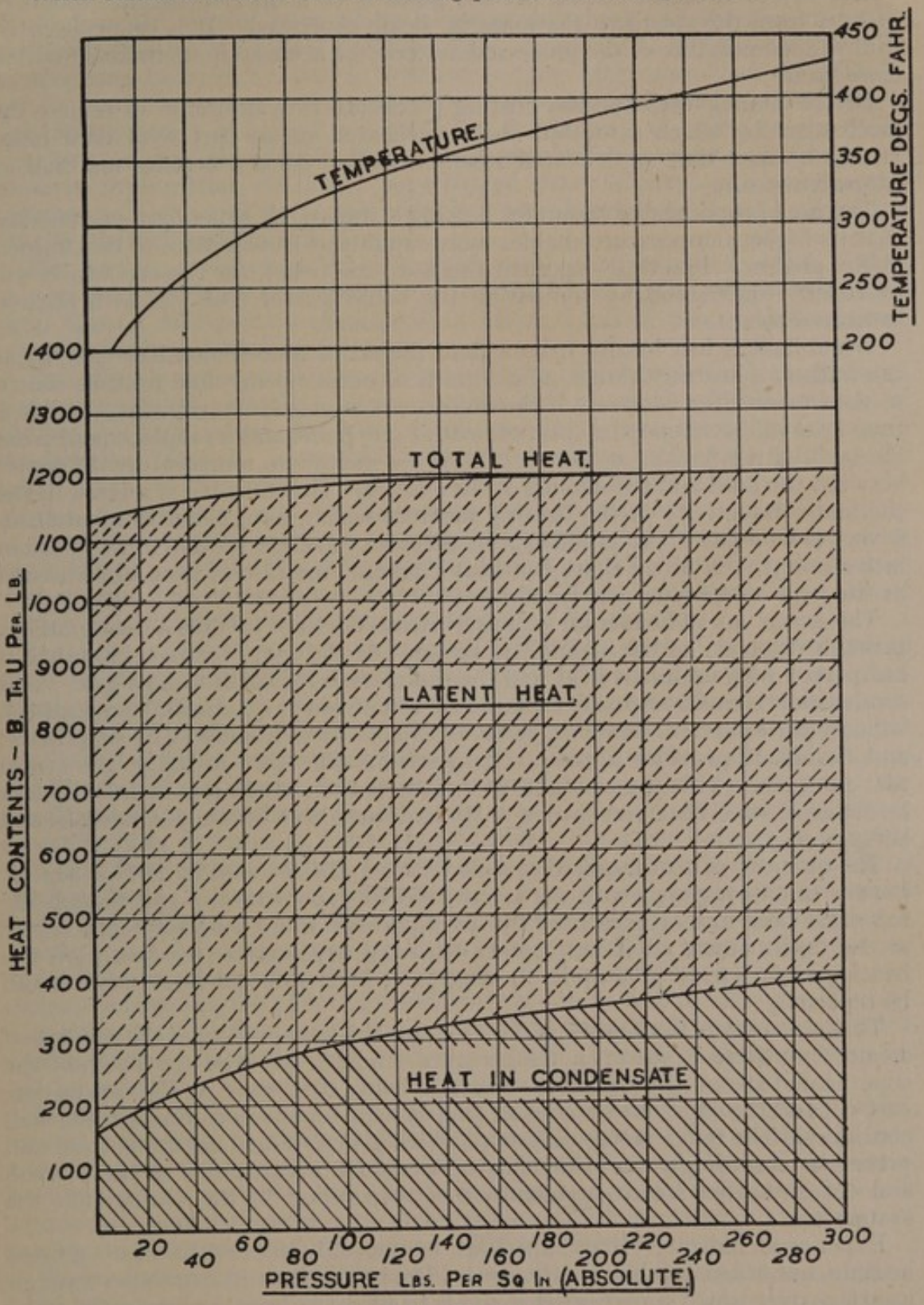

FIG. 203. Heat in steam.

depends on the rate of heat transfer, which varies greatly with the rate of flow, and on the amount of the heating surface area. These points are discussed in the next section.

(c) SUPERHEATED STEAM

The object of steam-heated process plant is to transfer a definite quantity of heat from the steam to the material being processed. It is often essential that the temperature of the processed material be accurately maintained within close limits.

Before dealing with specific heating processes, it is desirable to review the mechanism by which saturated and superheated steam part with their heat. It will be seen that superheated steam is not so good a heating material as saturated steam.

The use of superheated steam for heating is due to the belief, quite erroneous, that its higher temperature enables more rapid heat transference to be attained in the process. Practical experience refutes this belief, for reasons which will be readily understood by comparing the behaviour of true gas with that of saturated steam.

When heat is lost locally from a fluid (liquid or non-condensible gas), as at the walls of a heating system, a heat gradient occurs in the fluid from its centre to the transmitting surface. If the fluid is not in positive movement, there is a time interval between extraction of heat at one point and its replacement from the body of the fluid so as to re-establish the maximum temperature difference between the fluid and the outside of the system. This time lag is related to the thermal conductivity of the heating medium which, as previously mentioned, is very low for gases. In practice both diffusivity and convection must be taken into account, but where there is a heat gradient, the rate of heat transmission at the point of transfer is lowered.

The extraction of heat from saturated steam does not involve a heat gradient through the body of the steam. It remains at the same temperature until it has parted with its latent heat and then immediately ceases to be a gas. As it condenses it is simultaneously replaced by new steam at saturation temperature.

Superheated steam, however, behaves as a gas until it has lost its superheat ; and though it provides a greater temperature difference initially, the steam first in contact with the conducting surface of the system has to fall through its range of superheat and then give up its latent heat before it condenses and is replaced.

The quantity of heat required to superheat saturated steam by 200° or 300° F. is very small compared with the latent heat of vaporisation ; and though the few extra heat units represented by the superheat afford a higher temperature at the first moment of steam contact with the dry surface, the steam on the heating surface must give up its much greater bulk of latent heat before it can be replaced.

Thus, not only is the heating surface kept at the temperature of the condensed steam, but there is set up a temperature gradient between the body of the superheated steam and the steam which is being condensed at the conducting surface, and this gradient remains so long as heat transference through the wall continues. The temperature difference which is maintained between steam and process is accordingly the difference between the temperature of the process and the saturated steam temperature corresponding to the pressure in the system.

Experience indicates that superheat is not only useless in most process heating, but is usually disadvantageous. The rate of heat transmission through a surface over which superheated steam is passing has been demonstrated to be less than through a similar surface in contact with saturated steam at the same pressure for the reasons just given. Superheated steam supplied to one end of an extended coil, for example, and discharging as condensate at the other end, will part with less heat in the first half than in the second.

This may seem anomalous, since the difference of temperature between steam and process is greater in the first half ; but the fact remains, particularly when the process demands heat in considerable quantity. The explanation lies in the small quantity of heat released by lowering superheat and the low heat conductivity of gases (Chapters VIII and IX). A stream of hot gas can have its temperature lowered by 100° F. without parting with more than 30 B.Th.U. per lb., and if it is flowing across an area of heat-transmitting surface, this is the total quantity of heat it will release in the distance of travel necessary for that temperature drop, which distance is determined by the conductivity of the material transmitting the heat. Superheated steam is virtually a hot gas, and behaves as such until its temperature has been lowered to that of saturation for the pressure under which it exists.

The portion of an extended coil which receives superheated steam is thus subject to a much smaller quantity of released heat for a given temperature drop than is that portion of the coil in which a fall of two or three degrees releases the latent heat of the steam.

This effect is not so marked in the heating of a chamber to which superheated steam is supplied. Here the steam does not flow across any specific portion of the heating surface first, but forms a bulk of hot gas from the outside of which the heat is extracted. Immediately the superheat has been extracted from the outer envelope, that portion becomes saturated steam and condenses with the release of the latent heat. In the process the heat gradient set up in the body of steam causes a flow of heat by conduction, steadily lowering the temperature of the whole ; but the heat-transmitting surface remains essentially in contact with an envelope or stratum of saturated steam, from which the release of heat is high in quantity when the temperature falls a few degrees.

Certain experimenters have attempted to deduce a theory, based on limited evidence, that superheated steam has a greater heat-transmitting effect than saturated steam at the same pressure. A close examination of various tests with tubes through which steam flowed, however, indicates that support for the theory of higher value in superheat is only found within a narrow range of conditions. Variations in rate of heat transmission are closely related to the condition of the condensate deposited on the surface, i.e. whether it occurs as a regular film, or forms in drops with relatively dry areas between. A rough, irregular face tends to become covered all over with water which is retained by the roughness ; whereas on a smooth bright surface, and particularly one which has a trace of grease on it, the condensate tends to gather into large globules, having a considerable proportion of area between which is practically free from the insulating effect. Whatever the nature of the surface or character of condensate formation, the effect of velocity of the steam flow in removing it brings in another factor. Further, the velocity of the steam as it rushes towards the cooling surface to replace the condensed steam appears to have an influence on the heat flow.

In view of all the factors involved, it is quite possible to produce experimental conditions in which superheated steam shows a higher rate of heat transfer than saturated steam ; but the normal conditions of industrial heating circuits are such that this advantage is not realised. As previously mentioned, practical experience under some conditions shows a lower rate of heating with superheated steam than with saturated steam.

Process heating was divided in Chapter VII into the following broad groups :—

(1) *Evaporation.* By means of heating surface of :—
   (*a*) Calandria type, in which the heating surface is external and the liquor to be heated is inside the tubes, etc.
   (*b*) Coil or worm type, with heating agent inside the tubes and the liquor to be heated outside.

(2) *Raising the temperature of the processed substance*
  (*a*) By means of heating surface.
  (*b*) By direct contact.

(3) *Maintaining the temperature of the substance being processed*
  (*a*) By means of heating surface.
  (*b*) By direct contact.
    (1) To make up for losses by radiation, etc.
    (2) To replace heat absorbed in some endothermic process.

(4) *Distillation, by the use of open steam*

(1) In evaporation the temperature control is exercised, as already stated, by the physical properties of the process material and the pressure under which it is boiling. The only condition that the heating steam must fulfil is that it be sufficiently hot to give an adequate heat transmission through the heating surface. The higher the temperature of the steam side of the heating surface the more rapid will be the heat transfer.

There are two forms of evaporator heating surface—the calandria type and the coil or worm type. These types correspond exactly to the water-tube and fire-tube types of boiler respectively. The jacket type can usually be classed with the calandrias.

(1) (*a*) *Calandrias.* Process steam, however highly superheated, contains the bulk of its heat in the form of latent heat. From this it follows that most of the transfer of heat from steam to heating surface must take place at a constant temperature dependent on the pressure of the steam.

Superheated steam cannot exist in the presence of water. When the plant is operating the calandria tubes are covered with a film of condensate. If the steam fed into the calandria is superheated it immediately gives up its superheat by re-evaporating some of the condensate until it reaches saturation.

The thermal effect of using superheated steam in a calandria is, therefore, exactly the same as using saturated steam, provided that there is plenty of condensate present. From a practical point of view there are two differences. The first (beneficial) is that, as each pound of steam contains more heat, less steam and less condensate need be handled when using superheated steam than when using saturated steam. The second (harmful) is that there may be enormous temperature stresses at the flange attaching the steam pipe to the vessel. The pipe and flange are at superheat temperature. The vessel is at saturation temperature.

In general, therefore, the following conclusions apply to calandria vessels :—

(i) There is little thermal advantage in using superheated steam.
(ii) There is a mechanical disadvantage in using superheated steam.
(iii) If, for some reason, superheated steam is convenient, there is no reason, other than (ii) above, why it should not be used, and there is no need to de-superheat it.

(1) (*b*) *Coils or Worms.* In an evaporator fitted with coils or worms different conditions apply. The steam enters the coil and during its passage through the first part all it can do is to give up its superheat to the metal wall of the coil. There is no condensate present with which it can be de-superheated. Heat transfer, therefore, follows the behaviour of a dry gas, and takes place at lower rate than farther down the coil where the saturated steam is condensing. This lower heat transmission is partly offset by the fact that it takes place with greater temperature difference.

The conclusion can be drawn that in a coil evaporator superheated steam can be of no possible advantage and is probably undesirable.

(2) (*a*) *Raising the Temperature by Means of Heating Surface.* The temperature

of a substance can be raised by means of a heating surface by coils or by calandrias. The arguments relating to evaporators apply here equally but with an important addition. By the use of saturated steam a limit is set to the temperature to which the material can be raised. With superheated steam the temperature of the heating surface is no higher than with saturated steam during the temperature-raising process, but if, by negligence, the steam is left on too long, the temperature can rise, possibly dangerously, to that of the superheated steam.

Superheated steam is here clearly undesirable.

(2) (*b*) *Raising the Temperature by Direct Contact.* Many processes use steam blowers to raise the temperature of the process substance. There may sometimes be a legitimate use here for superheated steam. For a given water addition superheated steam will bring in more B.Th.U. and this may be important. On the other hand some parts of the material will be in contact with and be temporarily raised to the temperature of the superheated steam, with, possibly, detrimental results. In some instances it has been found that superheated steam blown into a liquid has insufficient time to de-superheat itself and condense, so that some breaks the surface and is wasted.

(3) (*a*) and (*b*) *Maintenance of Temperature.* The arguments in this connection are the same as those in (2) (*a*) and (*b*), except that the reasons for not using superheated steam are even stronger. Maintenance of temperature means maintenance of temperature and a plant that can produce overheating by the use of superheated steam is clearly undesirable.

From the foregoing it can be stated quite categorically that superheated steam for most process purposes is not only of no benefit but may be definitely undesirable.

(4) *Distillation.* This will be discussed in Section X of this chapter.

(*d*) WIREDRAWING

Suppose dry saturated steam at 160 lb. per square inch absolute is allowed to pass through a reducing valve into a low pressure main at 50 lb. per square inch absolute. Saturated steam at 160 lb. per square inch absolute contains 1,195 B.Th.U. per lb. of total heat. In expanding through the reducing valve the steam does no work, so it still holds those 1,195 B.Th.U. The superheated steam table shows that 50 lb. per square inch steam has a total heat of 1,195 B.Th.U. at a temperature of 320° F. The temperature of 50 lb. per square inch absolute saturated steam is 281° F., so that the reducing, expansion, " throttling " or " wiredrawing " has added 39° F. of superheat, although the actual temperature has fallen from 364° to 320° F.

When steam, particularly wet steam, expands through a valve that is cracked open, it is very apt to score the valve seat with grooves that look as if a wire had been drawn through the valve—hence the name " wiredrawing " for the expansion of steam that does no work.

This superheating by expansion only occurs to saturated steam below 450 lb. per square inch. At higher pressures there are other effects owing to changes in specific heat. For example, saturated steam at 750 lb. per square inch absolute blown through a reducing valve gets wetter down to 450 lb. per square inch absolute, and then gets drier until at about 230 lb. per square inch absolute it is again dry saturated steam. If it is still allowed to expand it superheats itself.

Steam used for direct heating in an injector or blower (see Chapter VII) often becomes superheated by expansion, and great losses can occur this way through superheated steam breaking the surface before becoming de-superheated.

It is clear, therefore, that a reducing valve will always give some superheat to dry saturated steam .

(e) SUPERHEAT AND STEAM DISTRIBUTION

In certain circumstances the superheat given to steam by reducing its pressure may be very useful. The exhaust steam from an engine, turbine or pump is often very wet. Steam is often wet, sometimes very wet, at the end of a long pipe line. If this wet steam goes into the heating surface of a piece of plant the extra condensate is a nuisance. It has little or no heating value, but it coats the heating surface with an additional water film, the extra water has to be handled by the trap, and the amount of flash steam is increased.

If such wet steam can be expanded as soon as possible on its journey to the heating process, the superheat due to expansion may be sufficient to render it dry when it reaches the vessel. This increases the amount of steam reaching the process plant and, as the steam is at a lower pressure, it will have a higher latent heat. There may, therefore, be a two-fold gain.

In a long steam main it is beneficial for the steam to be slightly superheated. Superheated steam loses heat to the pipe more slowly than saturated steam and the heat lost is only superheat. If saturated steam loses heat some of the steam condenses and this condensate is drawn off from the pipe by the draining system. Just that amount of steam is, therefore, lost. If high or moderate pressure steam is reduced in pressure at the vessel, the steam is superheated. Time will be lost if the steam is used in a heating surface. Steam will probably be lost if the steam is used for direct injection. If the pressure of distribution can be reduced by early expansion, the steam will be superheated in the steam pipes. It will lose less heat ; the heat lost will be superheat not steam ; less steam and/or time will be lost in the plant. Of course the steam mains must be adequate to carry the larger volume of steam.

(*f*) DE-SUPERHEATING

Superheated steam, as has been pointed out above, is almost always bad steam to use for heating. It may, therefore, be necessary to de-superheat it by passing it through a de-superheater which adds a spray of distilled water to the steam. The superheat gives itself up in evaporating some of the sprayed water. In the case considered at the beginning of (*d*) wiredrawing, the amount of superheating energy was 21 B.Th.U. This is to be removed by evaporating water in the de-superheater. The latent heat in 50 lb. per square inch absolute steam is 924 B.Th.U. per lb., so that every pound of steam de-superheated will evaporate $\frac{21}{924} = \cdot 023$ lb. of water. This increases the weight of saturated steam leaving the de-superheater by 2·3 per cent.

(*g*) HEAT AND TEMPERATURE

A point, somewhat elementary to the engineer, that is often overlooked by steam users, is the distinction between heat grade (or temperature) and heat quantity (thermal units). The temperature of saturated steam is dependent on its pressure—lb. per square inch. Heat quantity is dependent on weight of steam—lb. weight. Confusion often arises between 30 lb. of steam (weight) with 30 lb. steam (meaning 30 lb. per square inch pressure).

The temperature required in process heating is determined by the nature of the process or work to be done, some uses demanding a high temperature by reason of reactions that can be produced only at such temperatures. Others, again, require quantity of heat rather than high temperature. Many processes, such as distillation, require both a minimum temperature and a minimum

quantity of heat. It is here that many users fall into error, insisting on unnecessary temperature instead of increasing the quantity of heat transmitted.

As a simple instance designed only to indicate the difference between temperature and quantity of heat, an example may be taken of a plant in which it is desired to distil toluene from a second material boiling at a higher temperature. It is assumed for simplicity that no toluene condenses and runs back into the apparatus so that no additional heat is required on account of refluxing.

The still contains 2,000 lb. of the combined liquids having a specific heat in B.Th.U. per lb. per °F. of 0·43, and it is required to distil 1,500 lb. of toluene from the mixture, toluene having a latent heat of 156 B.Th.U. per lb. and a boiling-point of 231° F. (110·6° C.). The material of the still consists of steel weighing one ton and having a specific heat of 0·107. The whole of the apparatus and the liquid are initially at 60° F.

The theoretical heat required is as follows :—

	B.Th.U.
Heating the apparatus :—	
2,240 × (231–60) × 0·107	41,000.
Heating the liquids :—	
2,000 × (231–60) × 0·43	147,000
Distillation :—	
1,500 × 156	234,000
	422,000

This figure must be increased to allow for heat losses from the apparatus, and for this an allowance of 10 per cent. may be made. The final figure required is thus 464,000 B.Th.U.

The temperature required in the steam must not be less than that of the boiling point of the toluene and must obviously be a few degrees higher. Steam generated at 22 lb. per square inch absolute is at a temperature of 233° F. This will clearly be the minimum temperature of the steam.

The latent heat of steam at this pressure is 956 B.Th.U. per lb. and if the latent heat of steam only is used in the distillation and is completely used, the quantity of steam required will be 464,000/956 = 485 lb. Clearly, if a greater charge of the same oil is put into the still the heat required must be proportionately increased, but its temperature will not be affected.

If, on the other hand, an oil boiling at a higher temperature is distilled, the temperature and, therefore, the pressure of the steam must be increased in accordance with the temperature required ; the quantity of steam needed will depend upon the heat quantities required as shown by calculations similar to that already made in this example.

Admittedly, an increase in temperature will stimulate the flow of heat from steam to a cooler body and overcome the resistance offered by a restricted heating surface or low conductivity of the intervening medium. But this may be an unsatisfactory method of increasing the quantity of heat transmitted, and may be uneconomical. Instead, an adequate area of heat-transmitting surface should be provided. Section IV (*e*) (iv) contains further remarks on this subject.

An increase of steam temperature from 327·8° F. to 381·8° F. (or by 54°) involves increasing the pressure from 100 lb. to 200 lb. per square inch absolute, but the increase of thermal units in 1 lb. of steam is only 11·2 (from 1187·2 to 1198·4). Moreover, the heat units available as latent heat when the steam condenses are reduced, being 843 at 200 lb. pressure and 888·8 at 100 lb. pressure, as the quantity retained in the condensate is 355·4 as against 298·4.

IV. HEAT TRANSFER

The general principles of heat transfer are discussed in Chapter VIII. Only heat transfer as it affects steam will be dealt with here, and some account was given in the preceding section under " superheated steam " of the mechanism of the process.

The rate of heat transfer through a heating surface from the heating steam to the heated material depends on the following factors :—

(*a*) The temperature difference between the steam and the material.
(*b*) The temperature.
(*c*) The area of the heating surface.
(*d*) The thickness and material of the heating surface.
(*e*) The heat resistant films on the heating surface.
(*f*) The movement of the steam and the material.

These will now be discussed in turn, though some of them are largely interdependent.

(*a*) THE TEMPERATURE DIFFERENCE BETWEEN THE STEAM AND THE MATERIAL

The heat transmission can be taken for all practical purposes to vary directly with the temperature difference. This is one of the great limitations in the use of low pressure steam. If low pressure steam is available in quantity and adequate heat transfer cannot be secured, it will be necessary to try to improve some of the other factors that impede heat transfer.

(*b*) THE TEMPERATURE

The higher the temperature at which heat transfer takes place, the faster is the rate of heat transfer by radiation, though not by conduction and convection, for a given temperature difference. Figures given by different authorities vary so greatly that none will be given here.

(*c*) THE AREA OF THE HEATING SURFACE

At first sight it would seem obvious that a heating coil twice the size of another would transfer twice the heat. This is true up to a point. When a liquid is being heated the heat transfer will be slow unless the heated material be kept in brisk movement or circulation. In an endeavour to increase heat transfer by increasing the heating surface area, there is sometimes such a tangle of coils or pipes as to impede very seriously the movement of the material, so that heat transfer is actually reduced (cf. Chapter VIII).

(*d*) THE THICKNESS AND MATERIAL OF THE HEATING SURFACE

The conductivity of metals varies greatly, but the heat transfer through metal is so enormously greater than through the heat resistant films on the metal surfaces that for all practical purposes the metal of the heating surface can be ignored. Exceptions exist in special plant, such as glass-lined vessels, but here the glass is an additional heat-resistant film.

(*e*) THE HEAT-RESISTANT FILMS ON THE HEATING SURFACE

(i) Air film.
(ii) Condensate film.
(iii) Dirt or scale film.
(iv) Stagnant material film.

The heat flowing from steam in a steam-heated vessel to the material being heated has to pass through layers of resistance (see Fig. 204A).

Between the steam and the liquid to be heated there is first the resistance of

a film of air and gas in the steam space. This resistance is great because air is a very poor conductor of heat. Then follows a layer of water formed by condensation of the steam. This also creates a high resistance because water is a bad conductor of heat. The films are shown diagrammatically in Fig. 204A as separate layers, whereas in practice they would be present as a mixture. Next comes the metal wall of the vessel which is a good conductor and through which the heat will flow readily. Then there may be a film of dirt or scale. Finally, there is a stagnant film of material between the outer surface of the vessel and the material or liquid being heated. If the resistant films are thick the steam will have to be hotter than if the films were thin or absent, in order to give the necessary temperature gradient (Chapter VIII).

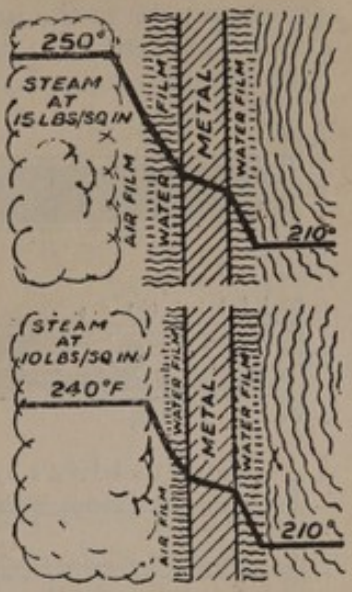

FIG. 204A. Heat-resistant films.

In the upper section of Fig. 204A steam is being supplied at a pressure of 15 lb. per square inch, and is at a temperature of 250° F. The final temperature of the liquid or material being heated is 210° F. In the lower section the resistant films of air and water have been reduced to half thickness. There is, therefore, a reduced temperature drop through the films. Because of this the final temperature of 210° F. can be maintained with a lower steam temperature and, therefore, a lower pressure, e.g. 10 lb. per square inch. Moreover, at this lower pressure more heat per lb. of steam is available. Alternatively, a steam pressure at 15 lb. per square inch can be maintained and a higher rate of heat flow is obtained ; this shortens the time necessary for the process.

The following figures will serve to explain the serious effect of these films which form on the sides of a vessel :—

	Conductivity at 200° F. B.Th.U./hr./ft.²/° F./ft.	Relative conductivity
Copper	222	1,000
Steel	26	117
Brick	0·8	3·6
Boiler scale	0·05–1·25	0·22–5·6
Water	0·37	1·67
Air	0·018	0·08

(i) AIR FILMS AND THEIR REMOVAL

Air fills all steam spaces when steam is off. When steam is turned on, air (and incondensable gases) and steam mix unless the air can escape. Steam generally has some air or gases mixed with it as it leaves the boiler. The air and gases come from the water fed into the boiler and are boiled off as steam is generated. The feed water should, therefore, be de-aerated before being fed to the boiler.

When the steam condenses on the inner surface of the heating or process vessel the air (which, of course, cannot condense), is deposited as a film, and this forms one of the insulating layers already mentioned.

The more air there is mixed with the steam the thicker will be the insulating film. The thicker the film the higher will be the steam pressure and temperature at which the plant will have to be worked to get the amount of heat needed from the steam through the film to the substance being heated.

If a vent is provided at any point in the steam space the air present will be forced out as soon as steam is turned on. If the vent is at a point remote from the steam inlet it will take longer for the steam to reach that point than if the vent were near to the inlet, and more air will have time to escape before steam reaches the vent.

All the air cannot be forced out of the vent however (because the steam is not like the piston of a pump), and some air will mix with the steam. This air and the air brought forward by the steam will accumulate unless it can be expelled.

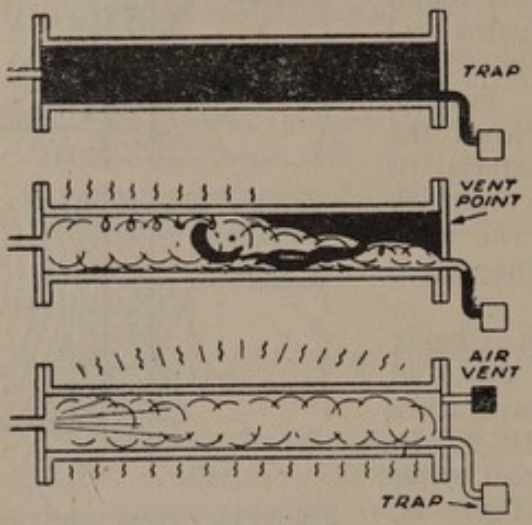

FIG 204B. Air venting point—pipes.

(*a*) *Position of Vents.* Fig. 204B shows a length of piping of, say, 6 inches bore. In the upper picture the pipe is full of air. In the second, steam has been turned on and some air has found its way through the drain trap; some has collected at the upper part of the outlet end of the pipe and some has begun to mix with the steam. Obviously there is need for an additional air venting point at the upper part of the pipe above the steam trap. The arrangement is shown at the bottom. The air has been able to find free outlet leaving the pipe filled with steam.

An additional air venting point is necessary where the trap is of the mechanical type without supplementary means of venting air. If, on the other hand, the trap draining the pipe is of the thermostatic type it will in all probability be quite capable of discharging all the air present in the pipe when steam is turned on and a separate air vent will not be necessary.

Fig. 204C shows two types of steam jacketed boiling pan. The upper row shows the simple fixed type with steam inlet and bottom condensate outlet; the lower row shows the tilting type with steam inlet through one trunnion and condensate outlet through the other trunnion.

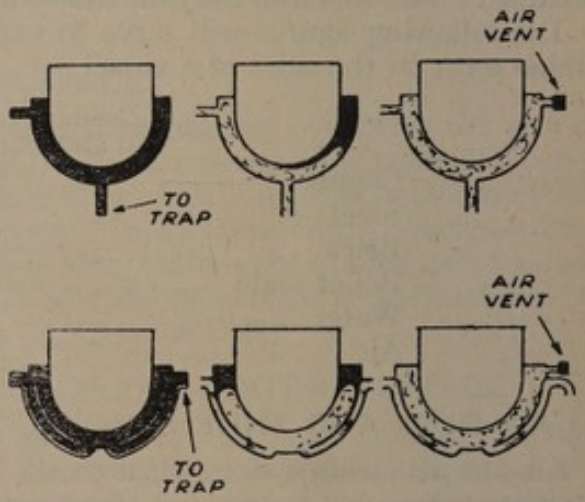

FIG. 204C. Air venting point—steam jacketted pans.

In the first section of the top row the jacket is seen to be full of air. In the second, steam has entered the jacket and although some air may have been discharged from the drain trap quite a lot is left and has accumulated at the upper part of the jacket, mainly at the side opposite the steam inlet. Here obviously is the most suitable venting point and on the right hand side an air vent is shown in position and the jacket clear of air.

The lower row shows steam first passing down the cored passage and then upwards into the jacket. The residual air collects at the upper part of the jacket round the periphery of the pan. The most suitable venting position will be at any point round the top of the jacket, as the example shows.

Fig. 204D shows a laundry calender with steam inlet and condensate outlet

at the bottom of the bed. The flow of steam will be in an upward direction and towards the draining end of the bed. Here again the best venting position is a point remote from the steam inlet. The most convenient places are the horns of the bed at the points shown by the two dots in the picture.

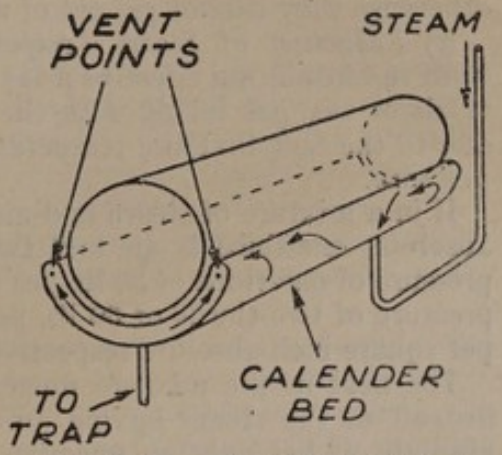

FIG. 204D. Air venting point—laundry calender beds.

Fig. 204E shows a less obvious problem. This represents a large rotary drying cylinder fed with steam through the left-hand trunnion. The point most remote from the steam inlet, as far as the steam flow is concerned, is here the point nearest the steam inlet.

(*b*) *Automatic Air Vents.* Having decided on the best place at which to arrange venting points, the most efficient means of releasing air from steam spaces must be considered. Hand cocks will release the air *if they are opened on starting up the plant.* But it is difficult to determine when all the air has been expelled, and if the cocks are not closed quickly steam will escape and be wasted. Again, unless the cocks are opened frequently during the period of operation air brought in by the steam will accumulate to the detriment of operation and steam consumption.

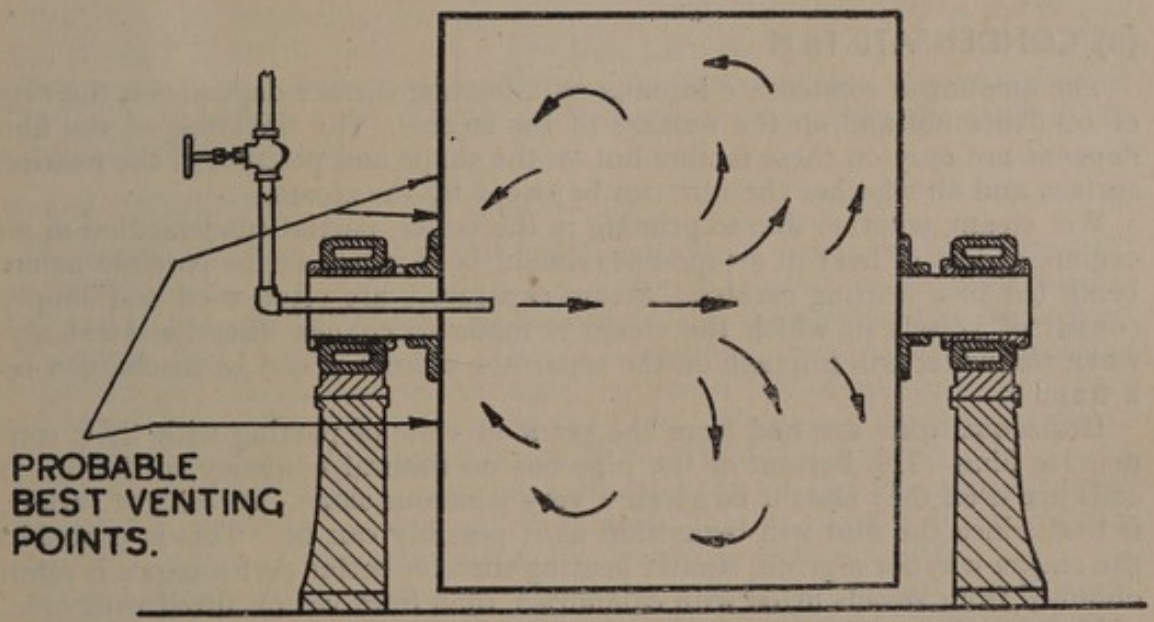

FIG. 204E. Air venting point—large rotary drying cylinder.

Probably the best way is to vent the steam space automatically. Air vents worked by thermostatic elements will release air and will prevent steam wastage. The valve of the vent is carried by an element which is contracted when cold, opening the valve. The initial cold air is expelled and the steam which follows heats up and expands the element, closes the valve and stops steam from escaping. When more air collects the element cools, opens the valve and the process is repeated.

When there are two or more steam mains at different pressures the various pieces of plant on the higher pressure can be conveniently vented into the lower pressure main. This only leaves the lowest pressure plant with the

necessity for care in venting by hand or automatically. The high pressure plant can be vented generously.

However venting is done the vents must be connected to the right place, otherwise they cannot get rid of all the air present.

(*c*) *Reduction of Steam Temperature through Admixture with Air.* Apart from the insulating effect of a layer of air in the heating surface, the presence of air or any gas mixed with the steam lowers the temperature of the steam due to the fact that the temperature of the steam is dependent on its partial pressure.

If in a mixture of steam and air at a total pressure of 30 lb. per square inch absolute, one-third is air and two-thirds steam, the air will exert a partial pressure of one-third of 30 lb. per square inch absolute, and the steam a partial pressure of two-thirds of 30 lb. per square inch absolute, i.e. 10 lb. and 20 lb. per square inch absolute respectively.

The heat in the mixture must, of course, come from the steam only, but instead of the steam having an apparent pressure of 30 lb. per square inch absolute, it has only an effective heating pressure of 20 lb. per square inch absolute.

Whilst the temperature of saturated steam at 30 lb. per square inch absolute is 250° F., at 20 lb. per square inch absolute the temperature is only 228° F. Thus the temperature of the mixture will be 228° F. and not 250° F. as would be expected from the total pressure reading.

In order to maintain output under such conditions either (*a*) a wastefully high pressure must be used in order to maintain good temperature conditions or (*b*) the air must be removed.

(ii) CONDENSATE FILM

The amount of condensate forming on a heating surface depends on the rate of condensation and on the wetness of the steam. The thickness of the film depends not only on these factors but on the shape and position of the heating surface and on whether the film can be swept by the steam.

Wet steam, whether due to priming in the boiler, partial condensation in an engine, or loss of heat in a pipe-line, should be dried as far as possible before being fed to a heating surface. Steam separators are often used and simply consist of vessels in which the steam is made to change direction suddenly, when the water will impinge on the separator whence it can be discharged by a trap.

Horizontal coils are bad from the point of view of parting with their condensate film. The bottom of the pipe has no natural tendency to drain. If coils are used they should be given a very generous slope. A vertical surface is best, since the film will be as thin as it possibly can be. This is probably the reason why for a given, similar heating surface, better performance is often obtained from vessels fitted with calandrias, than from vessels fitted with coils.

If the velocity of the steam can be kept high, the steam brushes the film and keeps it on the move, the brushing effect being particularly important in horizontal pipes.

The removal of condensate will be discussed in Section V.

(iii) DIRT OR SCALE FILM

Dirt and scale film are generally found on the liquor side of the heating surface. Air heaters collect dust and fluff which are excellent insulators. It is important, therefore, to clean air heaters regularly.

Many liquids deposit scale on the heating surfaces. The resultant lower rate of heat transmission can be offset by higher temperature differences with fuel-fired heaters, e.g. the steam boiler. But with steam heating the temperature

gradient is of necessity small, so that a small layer of scale has a disproportionate effect.

(iv) STAGNANT MATERIAL FILM

Most materials are bad conductors of heat, and unless the film in contact with the heating surface is constantly disturbed by quick flow or vigorous circulation, heat transmission will be very slow (cf. Chapters VIII and IX). It can be roughly assumed that a doubling of the speed of circulation or movement will increase the heat transfer by at least 25 per cent. In evaporators an adequate circulation can usually be obtained by natural means encouraged by good design, but in simple heating it may well be necessary to resort to mechanical circulation by pump or stirrer.

In evaporators, if the circulation of the process material is sluggish and there is considerable depth of liquid in the evaporator, it is possible for the lower layers of material to be raised to a temperature considerably in excess of the upper layer. This is due to the hydrostatic head of the liquid exerting a pressure on the liquid in the bottom of the vessel.

It might be thought that high temperature on the steam side of the heating surface would increase this danger, but with high temperature on the steam side the higher rate of heat transfer increases the circulation of the boiling substance due first to liquid convection currents and secondly to the mechanical agitation of the bubbles of steam acting on the liquid.

This increased circulation has beneficial effects. It greatly minimises the danger of bottom overheating because with violent circulation local stagnation is unlikely. By increasing the rate of flow over the heating surface the rate of heat transfer is much increased. The fact that the liquid is full of bubbles clearly reduces its mean specific gravity. A given depth of violently boiling liquid will, therefore, exert a lower hydrostatic pressure than the same depth which is boiling sedately.

(f) THE MOVEMENT OF THE STEAM

The beneficial effect of movement has already been discussed, but frequently the material cannot be moved at all, e.g. a moulding press, or the movement is unalterable, e.g. an air heater. Great benefit can then sometimes be secured by moving the steam rapidly. This is sometimes done by the deplorable method of just blowing steam through the vessel out of an open discharge, when the loss may be many hundred per cent. There is a method by which quite a brisk flow can be induced in the steam without any waste.

This can be done by an inlet valve constructed on the injector principle. The injector, however, must be so constructed that it has a positive injector action at any rate of steam flow, from full bore to a small amount.

When effective circulation is secured, all parts of the circuit are supplied equally with steam at the same temperature and pressure; the scrubbing action of the circulating steam helps to remove the condensate from the walls as it forms, and it is swept to the draining point and steam trap. The rate of heat transmission is increased by the reduction of the insulating film of water on the walls of the circuit; and accumulations of air and non-condensable gases are prevented, since these gases are constantly being swept along with the flowing steam.

More uniform heating is secured throughout the whole of the circuit, but no live steam is allowed to escape. Only condensate is discharged from the trap; and if the trap is located near the inlet side of the pump or injector, the condensate is sometimes discharged at a pressure somewhat below that of the system generally, by reason of the suction effect of the circulator.

The apparatus comprises a special circulating valve constructed on the

injector principle, through which the make-up steam is introduced to the heating system ; a steam separator which removes the condensate from the circulating steam ; and a steam trap for discharging the condensate. These are connected up in a completely closed circuit, so that the injected steam, in entering the circuit, induces a rapid flow from the return end back into the system, the return steam thus induced first passing through the separator where the condensate it carries is removed and delivered to the trap.

With the ordinary injector there is an optimum setting below or above which the efficiency falls rapidly. With a view to accommodating varying rates of heat transference (and hence steam consumption) in the heating system, the valve is designed to give a positive injector effect at any setting. Even at very low rates of delivery of make-up steam, a velocity is maintained that keeps the steam throughout the system in movement. Clearly to get a rapid circulation it would be desirable for the input steam to be at a considerably higher pressure than the circulating steam. This is often achieved automatically. Owing to the improved heat transfer a much lower pressure can often be carried in the circulating loop, thus giving the make-up steam a

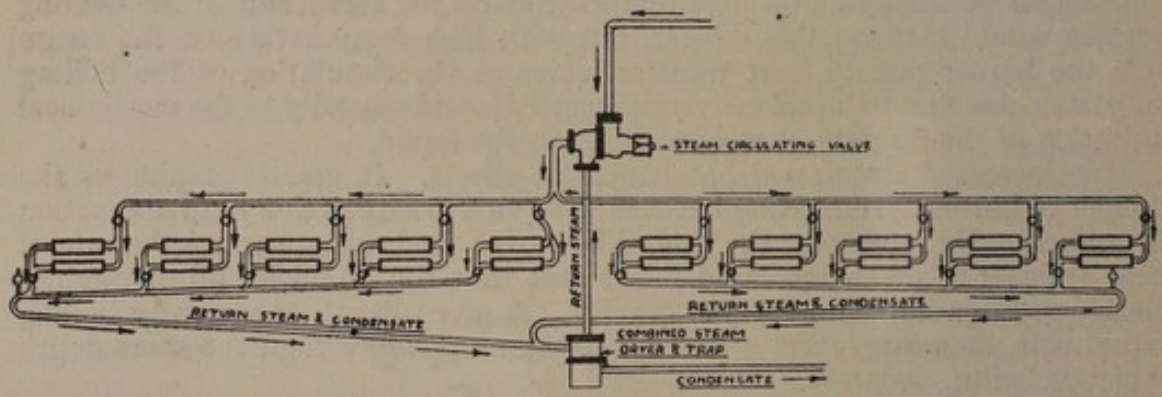

FIG. 205. Steam circulation applied to a battery of presses.

higher relative pressure. Fig. 205 shows the application of a steam circulator to a battery of presses.

Very Long Pipes and Other Exceptional Conditions. The path of the steam in a long heating pipe can fairly be regarded as the centre of the pipe, from which steam is being drawn by condensation on the walls. At the point of entry, the volume of steam passing being at its maximum, flow will occur close to the walls with the result that condensate forming there will be swept away and heat transmission maintained at its highest efficiency. At the half-way point along the system the velocity of the steam is proportionately reduced, with consequent less effect in stripping condensate from the walls. There is little or no flow at the end of the pipe.

For this reason, even though the temperature and pressure of the steam were uniform throughout the system, the heating effect at the early stages would be much greater than at the extreme end. In other words, the insulating effect of the condensate forming on the walls is countered less and less positively as distance from the point of inlet is increased. Nor can this variation in rate of heat transmission be entirely overcome by blowing steam from the discharge end of the system to create a flow. The inlet end still has the greater velocity of flow, since condensation progressively reduces the amount which arrives at the discharge point as steam.

Further, the pressure is not in fact uniform throughout the system. The initial pressure of the steam as it enters is constantly being lowered by condensation, and although fresh steam flows in to make good the decline the further the steam travels the lower becomes its pressure. It expands to fill the

space afforded by condensation. Since the make-up steam has to overcome the frictional resistance of the pipe and also supply loss of pressure by condensation all along its route, a gradual decline in pressure is inevitable.

Equal pressure throughout (and its concomitant of an equal amount of available heat in the steam) can be maintained only if a flow of steam is effected considerably in excess of the amount required to replace condensing steam. This is not possible in a normal condensing system. These difficulties can usually be met by dividing the long coil into two or more sections working in parallel. If this does not effect a cure, steam circulation may be found possible, but in many heating systems with very long pipes heat transfer is very slow—for example in oil storage tanks—and there may be insufficient input steam to induce circulation. The cure is probably to use very hot water in rapid circulation.

High Pressure Hot Water. Hot water is also of great benefit in certain other circumstances. In some moulding presses the amount of heat required is very small and the rate of heat transfer is low. Most of the input heat radiates from a machine that is difficult to lag properly. In such conditions the only possible method with steam heating is to blow through in order to get adequate production rate. Hot water may well effect a solution. Water can be used at the same temperature as steam by putting it under pressure. The advantages are that rapid circulation promotes heat transfer ; there are no traps or condensate return problems ; there is no possible loss of flash heat ; the water is in a closed circuit and no boiler feed water problems exist ; the pipes can go anywhere—there is no need to fear dips or pockets which might induce water-hammer with steam ; a hot water tank can be arranged in the system to give any desired degree of accumulator action, thus greatly ironing out peak demands on the boiler ; no boiler feed pump is needed.

Against these must be set the following : in order to avoid the danger of raising steam the water must be kept at a pressure considerably above its boiling pressure at the temperature in use ; the circulating pump may require considerable power ; the return pipe must be much larger than a condensate return line ; in order to give up its heat the water must suffer a temperature drop ; in order to get close temperature control the rate of circulation must be very rapid ; hot water cannot meet a peak demand nearly so well as steam.

Where heat transfer is rapid hot water cannot compete with steam, but in difficult sluggish heat transfer operations high pressure hot water may give an elegant solution.

For higher temperatures liquids other than water may be used, e.g. petroleum oils, diphenyl, melted lead.

V. CONDENSATE REMOVAL AND HANDLING

The condensed water film in pipes and heating surfaces, having been kept as thin as possible and encouraged to fall, or being brushed away, must be removed from the system without accumulating and without being accompanied by any steam.

Discharging Water from Steam Spaces. On many plants hand-operated drain cocks or cracked valves are still in use. They are opened wide when starting up so as to release the heavy initial load of condensate and then are supposed to be closed by the operator so that they are just releasing water at the rate at which it is being formed. Quite apart from the fact that most machine operators are far too busy to be overcareful in the setting of a drain cock, it is most difficult to set the cock so that it is just passing the condensate and not either causing water to be held back in the steam space or blowing steam. So if output is important the error is on the side of too much steam and the fuel waste is obvious.

Steam Traps. The method of discharging condensate now almost universally used is an automatically self-adjusting valve, known as a steam trap. Its function is to give automatic control of discharge without loss of live steam. Thus it is able to differentiate between steam and condensate.

There are two main groups of steam traps, each of which comprise a number of different designs :—

(1) *Thermostatic.* This type differentiates between steam and condensate by a temperature difference which operates a thermostatic element carrying a valve. As condensate and the steam that is condensing are at the same temperature, this type of trap clearly has limitations.

(2) *Mechanical.* This type differentiates between steam and condensate mechanically, generally by the action of a float or " bucket."

Correct Choice of Steam Trap. Careful thought should be given to the type and capacity of steam trap to be used and the position in which it is to be fitted. These are factors which have an important bearing on the improvement in plant efficiency and extent of fuel economy which may be obtained. Each of the many types and patterns has its specific purpose ; each has disadvantages as well as advantages.

THERMOSTATIC TRAPS—BALANCED PRESSURE TYPE (Fig. 206A).

Advantages. These traps are small in size but large in condensate handling capacity. The thermostatic element can be arranged to discharge the condensate at a relatively low temperature enabling some of the sensible heat in the condensate to be given up where the temperature difference between steam and the material being heated is large and the heating surface is generous. The valve is open when the trap is cold, so that it cannot freeze up when installed outside, unless there is a rise in the condensate pipe after the trap which would allow the water to run back and flood the trap when steam is off. Another advantage is that the discharge capacity of the trap is greatest when the amount of condensate is greatest (this is because the valve is wide open when starting up), so this type of trap never needs a by-pass to help it handle starting loads of condensate. Another advantage of the balanced pressure thermostatic trap is that it will automatically discharge any air which reaches it.

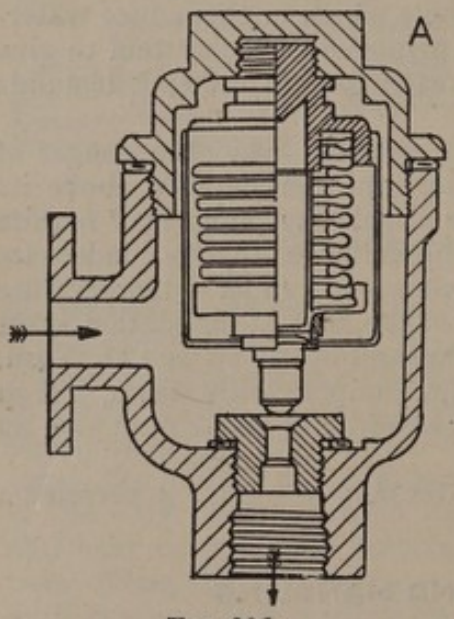

FIG. 206A.

Disadvantages. The thermostatic element of a balanced pressure trap is made of a flexible material ; because of this the element is liable to damage by waterhammer or by condensate which contains corrosive substances. This type of steam trap should not, therefore, be used under either of these conditions. If superheat is present it is never safe to use the balanced pressure thermostatic trap. This is because superheat, on account of its extra temperature above that of saturated steam at the same pressure, would create a pressure in the thermostatic element not balanced by the external pressure and it may be greater than the element is designed to stand.

THERMOSTATIC TRAPS—LIQUID EXPANSION TYPE (Fig. 206B).

Advantages. This type can be used on superheated steam and on higher

pressures (within the pressure range of the individual pattern). The trap will, if required, discharge condensate at a very low temperature. It is therefore particularly suitable for installations where the opportunity occurs of using a good deal of the sensible heat in the condensate before discharging. As in the balanced pressure thermostatic type, the liquid expansion thermostatic trap is open when cold and cannot freeze when installed outside (always providing that the discharge line from the trap does not rise). The discharge capacity of

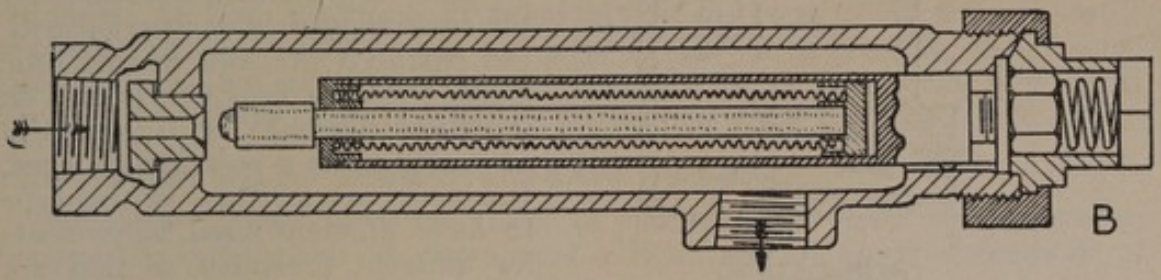

Fig. 206B.

the trap is greatest when the condensate is cool, as when starting up. This thermostatic trap is of the continuous discharge type, so it is quiet in operation. It is made adjustable for discharge temperatures and it will maintain the desired discharge temperature within reasonable limits. This trap is not affected by vibration, steam pressure pulsation or waterhammer.

Disadvantages. Sometimes it may be necessary to provide a by-pass to help this trap in coping with severe load variations. If the steam pressure or condensate volume at the trap varies widely and quickly the thermostatic element of this trap will not pick up the consequent temperature change quickly. It may, therefore, take some little time to adjust itself to new conditions. One other disadvantage is that the flexible tubing of the packless gland in this type can be destroyed by condensate containing corrosive substances.

THERMOSTATIC TRAPS—METALLIC EXPANSION TYPE

Advantages and Disadvantages. The principle of operation of this type is exactly similar to that of liquid expansion thermostatic traps except that valve movement is obtained by the expansion of a metal rod instead of by the expansion of oil. The traps have the same general characteristics, but because of the smaller movement per degree of temperature variation of a metal than of a liquid filling, the traps are not generally considered so flexible as the liquid filled type. They obviously do not suffer from waterhammer or corrosion and are much more robust. They are, however, generally of considerable size.

Thermostatic traps are most suitable where small amounts of condensate are to be handled, and where the load does not fluctuate. The valve is not very positively closed unless the trap is set to discharge cool condensate so that scoring of the seat may occur. In the event of failure they must be returned to the maker; they cannot generally be repaired by the works maintenance staff. Their principal advantage is their extremely small size.

MECHANICAL TRAPS—FLOAT TYPE (Fig. 206—C and D).

Advantages. Generally, this type is suitable whether condensate loads are heavy or light. It is not affected by wide and constant fluctuations of pressure. It is easy to install and usually all the working parts can be reached without taking the trap out of the line. As it removes condensate immediately, this type of trap is ideal for draining units where the rate of heat transfer is high for the heating surface available. Many patterns of float trap will automatically

discharge any air which reaches them so these will not air lock on starting up. It discharges condensate at steam pressure and temperature.

Disadvantages. The ball float (also the element of the thermostatic air release, if this is fitted) can be damaged by waterhammer or corrosive condensate. The float can leak. If the trap is fitted with a standard type of thermostatic air release it should not be used on superheated steam, for the reason given in connection with the balanced pressure thermostatic traps. When a float trap is used on superheated steam it is generally advisable to fit a separate air vent. It is necessary for traps of this type to have different sized valve seats for different pressures, so they are not self-adjusting to all pressures up to the maximum as are balanced pressure thermostatic traps. The reason is that the size of the discharge orifice of a float trap is governed by the power of the float and the steam pressure. As the power of the float is constant the higher the steam pressure the smaller must be the available discharge orifice. This last disadvantage is common to all kinds of mechanical traps.

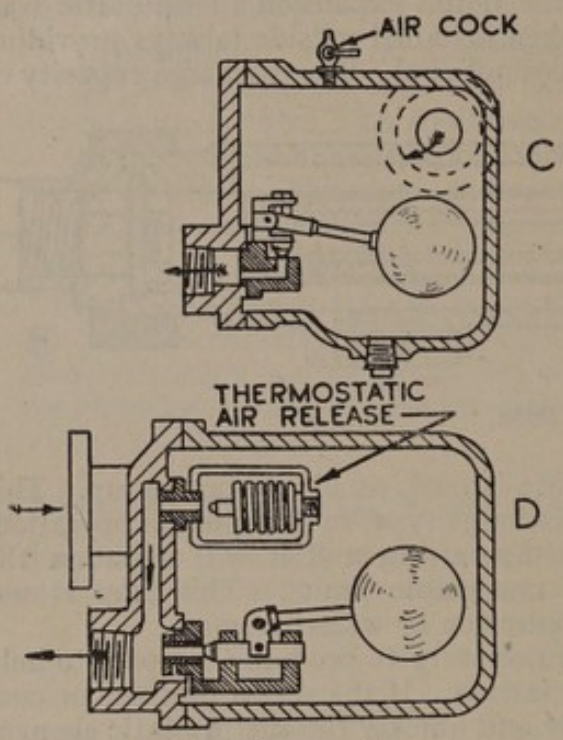

FIG. 206—C and D.

Float traps without thermostatic air vents may need to incorporate a manually operated valve to obviate air locking on starting up. This may lead to a potential source of steam wastage, according to the design. Float traps can be made for high pressure by the use of a counterweighted solid float, e.g. a solid aluminium ball.

MECHANICAL TRAPS—OPEN-TOP BUCKET TYPE (Fig. 206E).

Advantages. Generally these are very robust traps. They can be made for use on very high pressures and for superheated steam.

Disadvantages. No provision is made as a rule for air venting so this trap is liable to air lock. It is usually bulky. When fitted with air cocks or by-pass valves incorporated in the trap construction, the operation of these is manual, and as a result steam wastage is not uncommon.

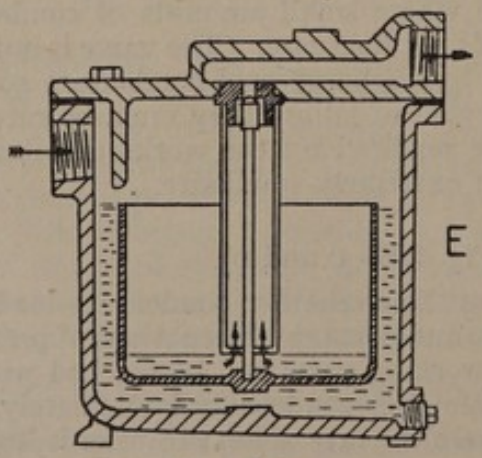

FIG. 206E.

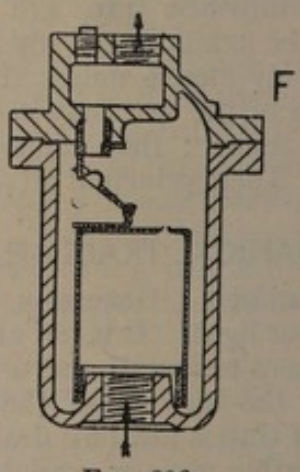

FIG. 206F.

MECHANICAL TRAPS—INVERTED BUCKET TYPE (Fig. 206F).

Advantages. This type can withstand waterhammer. It can be made of corrosion-resisting materials. It can be used on superheated steam and will discharge condensate as soon as it forms. Generally, the working parts of the trap are very simple and there is little to go wrong with it mechanically. It discharges condensate at steam temperature. On light loads it condenses steam within itself.

Disadvantages. It is wasteful of steam if not chosen exactly for the work to be done. It does not respond favourably to fluctuations of pressure or of condensate load. It discharges air only slowly. It is possible, under some circumstances, for an inverted bucket trap to lose its water seal, and if this happens it will blow steam. When installed outside the trap may freeze.

VERY LARGE RELAY OPERATED TRAPS (Fig. 206G).

For handling very large quantities of condensate a direct operating trap becomes so large as to be impractical. There are all kinds of methods of working a large vessel as a trap by means of a relay. Some of these methods are very elaborate. One of the simplest is shown in Fig. 206G.

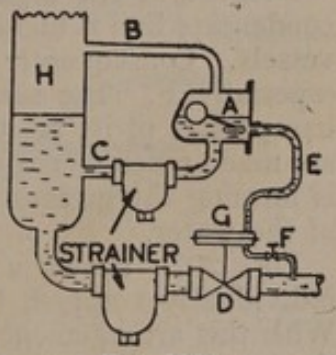

FIG. 206G.

A small continuous discharge float trap, A, is fitted in parallel with the main discharge pipe from the plant connected by balance pipes B and C. The float valve of the trap controls the flow of condensate from the trap outlet by pipe E to the upper side of a diaphragm G controlling the main valve D.

With no condensate in the pipe H the float valve of the pilot trap A is closed, and, there being no pressure on diaphram G, valve D is closed. When condensate collects in pipe H, float valve A is opened and pressure is put on diaphragm G opening the main valve. The amount of valve opening is controlled by the opening of float valve A which in turn is fixed by the rate of flow of condensate into pipe H.

A needle valve F allows pressure to leak continuously from the diaphragm chamber G so that when the condensate level in H falls and the float valve closes, pressure in the diaphragm chamber will quickly dissipate and allow valve E to close.

THE WRONG USE OF THE RIGHT STEAM TRAPS

The wrong use of steam traps has more to do with waterlogging and loss of

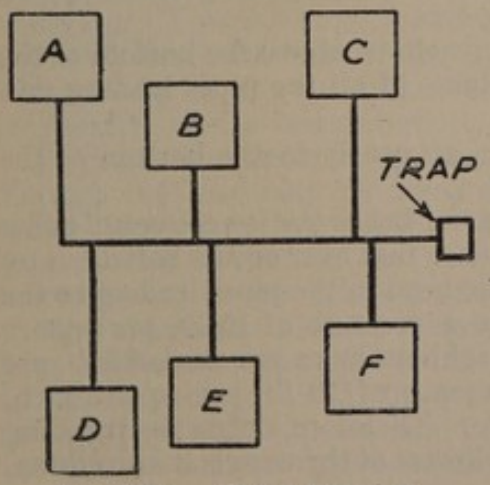

FIG. 207A. Group trapping.

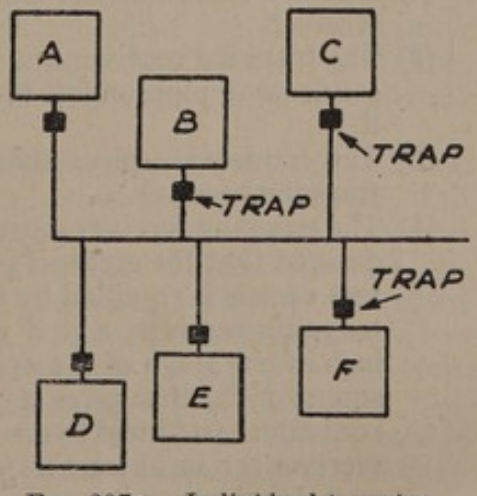

FIG. 207B. Individual trapping.

output efficiency than is generally realised. An example often occurs in what is known as group trapping. This is the practice of connecting a number of drains from steam-heated plant to a single steam trap.

(*a*) *Group Trapping.* Fig. 207A shows six steam-heated vessels (they may be boiling pans or laundry presses, or any other type of steam-heated equipment). The six vessels are identical in shape and they work at the same steam pressure. The drains from the six vessels are connected to a common condense pipe leading to a single steam trap. The trap is in good working order and its capacity is adequate for the combined condensate load of the six vessels.

To explain why waterlogging of the vessels occurs, suppose that vessel F is just starting up. It is condensing heavily and the pressure at the drain outlet is considerably below the steam pressure at the inlet. Vessel A is nearing the end of the process and as its temperature is high the condensation rate has dropped. Thus the pressure at its outlet is not much below the steam pressure at its inlet. Vessel C is midway through the process and the other vessels more or less the same.

Because of the high outlet pressure from A the pressure in the common condensate line is higher than the pressure at the outlet from any of the other vessels. Consequently there will be a tendency for these vessels to waterlog, especially F. The same conditions will occur whenever a vessel is started, irrespective of its position in relation to the common condense line. Matters are made much worse because of the fact that the vessel most recently started is forming the greatest load of condensate, but is least capable of getting rid of the water.

Fig. 207B shows a correct method of draining the vessels. Each is individually trapped, the traps discharging to the common condensate line. With this arrangement the condensate line is always at a pressure lower than the lowest outlet pressure of the vessels so that each can discharge its load of condensate quite freely through its own trap at all times, irrespective of the conditions in other vessels.

Fig. 207A and B shows that a single trap cannot be blindly substituted for a number of individual traps.

Group trapping must be very carefully designed if it is to give satisfactory results. When many traps in bad condition are replaced by one large trap in good condition, savings in steam have been made. But this is usually the wrong solution. There is only one arrangement where the use of one trapping receiver on several steam vessels is really justifiable. One trapping receiver can be used on a number of vessels if the following conditions are all satisfied :—

(1) Each vessel must have a separate condensate pipe right into the trapping receiver.
(2) The trapping receiver must have a water volume above the bottom of the condensate pipes equal to the total volume of all the pipes leading into it.
(3) The condensate pipes should, therefore, go nearly to the bottom of the trapping receiver.
(4) The trapping receiver must be situated so far below the lowest vessel being drained that the greatest pressure difference that ever occurs between any two vessels is equalled by the hydrostatic head in the pipes leading to the trapping receiver, e.g. if one vessel has a pressure of 15 lb. per square inch at one stage of the cycle and its neighbour has a pressure of 5 lb. per square inch, this gives a pressure difference of 10 lb. per square inch, equivalent to 23 feet hydrostatic pressure. Therefore, unless the trapping receiver can be at least 23 feet below the lowest of the vessels it is draining, waterlogging may occur.

Group trapping that does not fulfil these conditions may be wasteful by slowing down processing and possibly spoiling process material.

With insufficient hydrostatic head, water will build up in the pipes and then up into the steam heating space or coils in the vessel. This reduces output by causing the heating to be largely or wholly by hot water instead of steam.

(*b*) *Position of Traps.* Another frequent source of waterlogging, loss of output and excessive steam consumption is the habit of fitting steam traps in what seems to be the most convenient place rather than in the right place. Here are one or two examples.

Fig. 207C is a diagram of a laundry calender. The drain outlet from the bed is separated from the steam trap by a long length of horizontal pipe. In the first picture steam is off ; water is standing in the bed and in the bottom of the connecting pipe. In the second picture steam has been turned on and the water has been discharged through the trap. The bed, the pipe and the trap now are

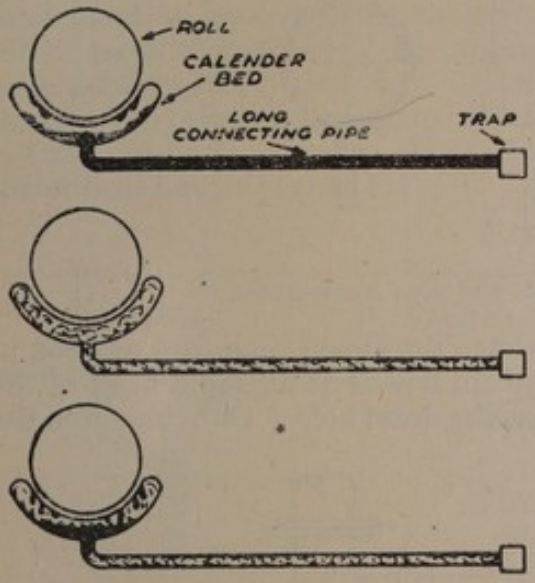

FIG. 207C. Steam locking—laundry calender.

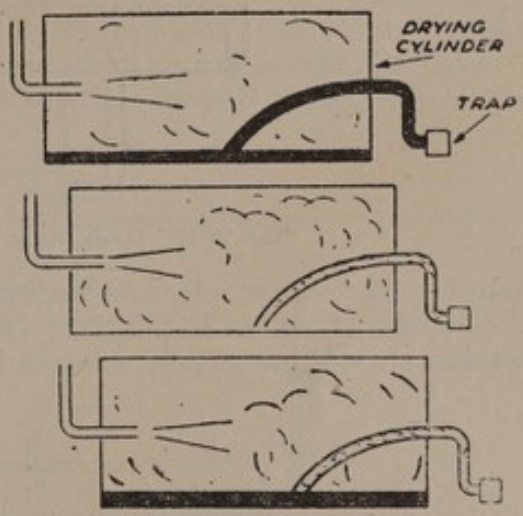

FIG. 207D. Steam locking—steam-heated drying cylinder.

full of steam. The trap is, of course, closed and will remain closed so long as steam is present in the pipe. But condensation is taking place in the bed and, as will be seen in the third picture, water is collecting in the base. It cannot be discharged because the trap is steam-locked. Until the steam in the pipe condenses the trap will remain closed and the calender bed gradually will waterlog. This will happen every time steam follows water into the connecting pipe. It is not the fault of the trap but of the long length of horizontal pipe. The remedy for this particular case of steam locking is to move the trap as near as possible to the bed outlet.

To put the trap near the exhaust outlet cannot always prevent steam locking. Though the trap may be fitted correctly the plant design may be such that steam locking is inevitable. The next example, a steam-heated cylinder (Fig. 207D), illustrates the point. Steam enters the cylinder through one trunnion and condensate is discharged through the other. The water is lifted to the exhaust trunnion by means of a dip pipe. In the first picture steam has been turned on and water is being discharged through the dip pipe and trap. In the second picture all the water has been discharged. The dip pipe, connecting pipe and trap are now full of steam. Condensation is taking place in the cylinder and, as is seen in the third picture, water is collecting at the base. It cannot get up the dip pipe to be discharged because of the steam locking.

Until the steam in the pipe condenses, the trap will remain closed and water will continue to build up in the cylinder. The condition is aggravated because the dip pipe is surrounded by steam and it will take so much longer for the steam inside the pipe to condense and allow the trap to open. Putting the trap right up against the exhaust trunnion does not prevent the trouble. Again, the only remedy is to use a trap which is fitted with an anti-steam locking device.

The function of the steam lock-release device mentioned in the last two examples is to dissipate the small quantity of locked steam. It will be noted that the locked steam is wasted because it is outside the steam space proper and is not, therefore, in a position to do useful work.

PIPE DRAINING

Unless steam pipes are properly drained the steam cannot reach the plant it is feeding as dry steam. If there is much undrained water in the pipe water-

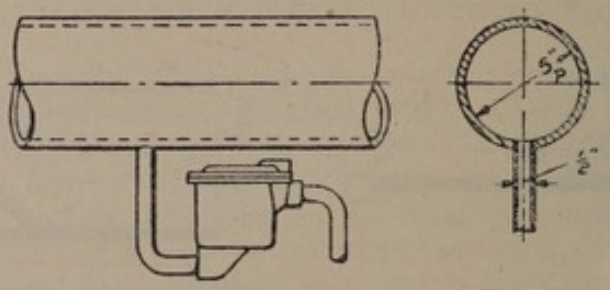

FIG. 207E. Collecting pockets and pipe connections.

hammer may occur. In hundreds of factories the steam pipes are drained in the manner shown in Fig. 207E. If the steam flow is at all rapid most of the condensate will jump right across or by-pass the drain hole., Only too often the

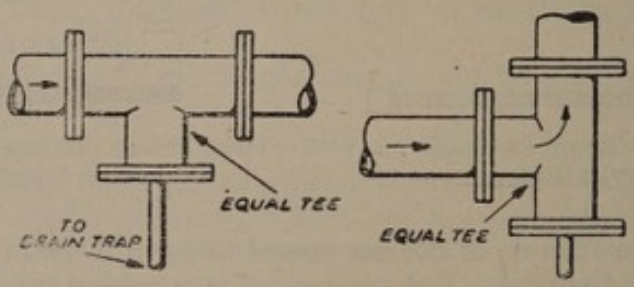

FIG. 207F. Collecting pockets and pipe connections.

drain pipe actually projects into the pipe, thus making quite sure that draining is not being done.

If the main is to be kept completely clear of condensate the correct method is to fit a full size Tee-piece or some other collecting pocket of adequate size into

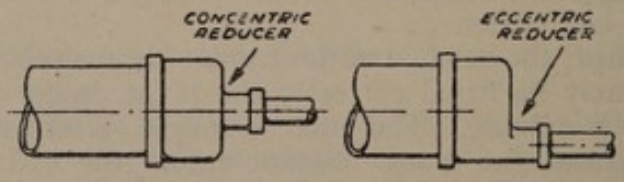

FIG. 207G. Collecting pockets and pipe connections.

which condensate will run. Once in the pocket the water will be clear of the steam flow and the trap will then be able to deal with it. Fig. 207F shows, on the left-hand side, a simple but effective collecting pocket of the kind described. The right-hand section shows a similar pocket fitted at a bend in the main.

The practice of fitting a concentric reducer at the end of a steam main for

drainage to the trap is quite common. This is shown in Fig. 207G. It will be obvious from this that the bottom of the main must always be waterlogged. The correct way to drain the main and so avoid waterhammer is to fit an eccentric reducer for drainage to the trap, as shown.

CONDENSATE REMOVAL FROM VESSELS WORKING UNDER VACUUM

The most common case is that of the condenser into which an engine or turbine exhausts. The Edwards air pump not only removes the air or gas from the condenser to maintain the vacuum, but withdraws the condensate as it is formed.

Where the evaporator or condenser that is to be drained is well above ground level, by far the simplest method is to use a " barometric leg " dipping into the water seal in an " atmospheric tank." There are certain precautions that must be taken with a barometric arrangement. The discharge from the atmospheric tank must be by overflow, leaving a minimum volume of water in the tank rather more than equal to the volume of the barometric pipe. Otherwise at a vacuum slightly higher than normal all the water might be drawn out of the tank up the pipe at starting up and the vacuum broken. The other precaution is that the barometric leg should have a generous height margin above the theoretical height (Table 111).

TABLE 111

Vacuum	Theoretical barometric head in water.	Recommended head.
5 in.	5·75 ft.	11 ft.
10 ,,	11·50 ,,	17 ,,
15 ,,	17·25 ,,	22 ,,
20 ,,	23·00 ,,	28 ,,
25 ,,	27·60 ,,	33 ,,

If local conditions do not permit of these heights it will be necessary to withdraw the condensate by means of an extraction pump or by means of a lifting trap (see later).

CONDENSATE RETURN

Having now withdrawn the condensate from the various pieces of plant and from all the distributing mains, the next step is to consider what should be done with it. The obvious thing is to return it to the boiler where, if it is uncontaminated, it will form pure distilled feed water.

If the condensate could be returned to the boiler without loss by flash it would not matter what pressure was used and there would be no great benefit in using low pressure steam. This is rarely possible.

If the whole of the flash can be collected there will be no heat loss from high pressure condensate and, again, theoretically there is no benefit from using low pressure steam. The recovery and use of flash is dealt with in the next section.

Condensate is generally returned to a sump tank, called a " hot-well," from which the boiler feed pump draws. As a pump will not lift or suck water that is at all hot because the reduction in pressure causes a flash of steam which breaks the pump suction, the name " hot-well " is somewhat misleading. If it is a well it cannot be hot. If it is hot it cannot be a well. Table 112 shows the suction lift or the head that the " hot-tank " should have above the boiler feed pump suction for various temperatures of feed water (cf. page 319).

TABLE 112

Temperature of feed water	Maximum suction lift	Minimum pressure head
130° F.	10 ft.	—
150° F.	7 ,,	—
170° F.	2 ,,	—
175° F.	— Level	—
190° F.	—	5 ft.
200° F.	—	10 ,,
210° F.	—	15 ,,
212° F.	—	17 ,,

Most traps will discharge condensate to a height above the trap outlet. This height depends on the pressure of the steam inside the trap driving the water out. It is thus possible to raise the condensate some 2 feet for every 1 lb. of pressure at the trap. This can be done sometimes, but it may lead to loss of output and sometimes it is dangerous.

If a heating surface is fed with 30 lb. steam and the trap discharges condensate into a tank 60 feet above, no condensate will be discharged until the pressure inside the heating surface has reached 30 lb. per square inch. At the beginning of the heating operation the heat transmission is generally so rapid that the pressure inside the heating surface is reduced by the brisk condensation. At such a time the amount of condensate to be discharged is a maximum, yet none of it can be discharged. A balance is reached by part of the heating surface becoming waterlogged and so reducing heat transmission that the pressure will rise sufficiently to eject the condensate. This may result in a great loss of time.

The drain traps of a steam main should not discharge above the level of the main. If the condensate is made to discharge above the level of the main it means that no condensate can be discharged until there is a pressure in the main. This sets up the perfect conditions for waterhammer.

It is, therefore, often necessary to run the condensate into a tank below the points being drained. The water from this sump or well can be raised to a high-level boiler feed tank by means of a pump or lifting trap.

LIFTING TRAPS

A lifting trap or return trap is, strictly, not a trap. It is a pistonless pump. Inside the trap body is a float which operates the valve gear. When the float is at the bottom water can run into the trap by gravity through a non-return valve. When the trap is full the float trips the valve, which opens a steam supply (at any suitable pressure) into the top of the trap. This steam drives the water out of the discharge pipe, through another non-return valve, up to the elevated tank. When the trap is nearly empty the float trips the valve gear, shuts off the steam supply and opens an exhaust valve. The exhaust steam can well be blown into the high-level tank to add extra heat to the feed water ; or the exhaust pipe can pass through a coil of pipe in the high- or low-level tanks. When the pressure in the trap has been sufficiently reduced the water can run in again and refill it. Clearly a lifting trap can be used to remove condensate from a vessel under vacuum provided the trap exhausts into the vacuum.

Where traps discharge a long way from the boiler it may not pay to return the condensate to the boiler, but an endeavour should be made to find a use for it. Condensate is pure distilled water and generally contains much heat. It is well worth taking some trouble to recover.

VI. FLASH

It has been explained in Chapter VII that when the pressure is reduced on hot water the water at once cools down to the temperature corresponding to boiling point at the lower pressure, giving up its surplus heat in a flash of steam.

THE COLLECTION AND USE OF FLASH STEAM

When condensate under pressure is released to the atmosphere it cannot exist as water at a temperature above 212° F. and its temperature is at once reduced to this figure. The excess heat above that in the water at 212° F. then is given off in the form of latent heat, forming what is known as " flash steam."

When the condensate is collected in a long return line which is under atmospheric pressure the flash steam will often condense, forming condensate at

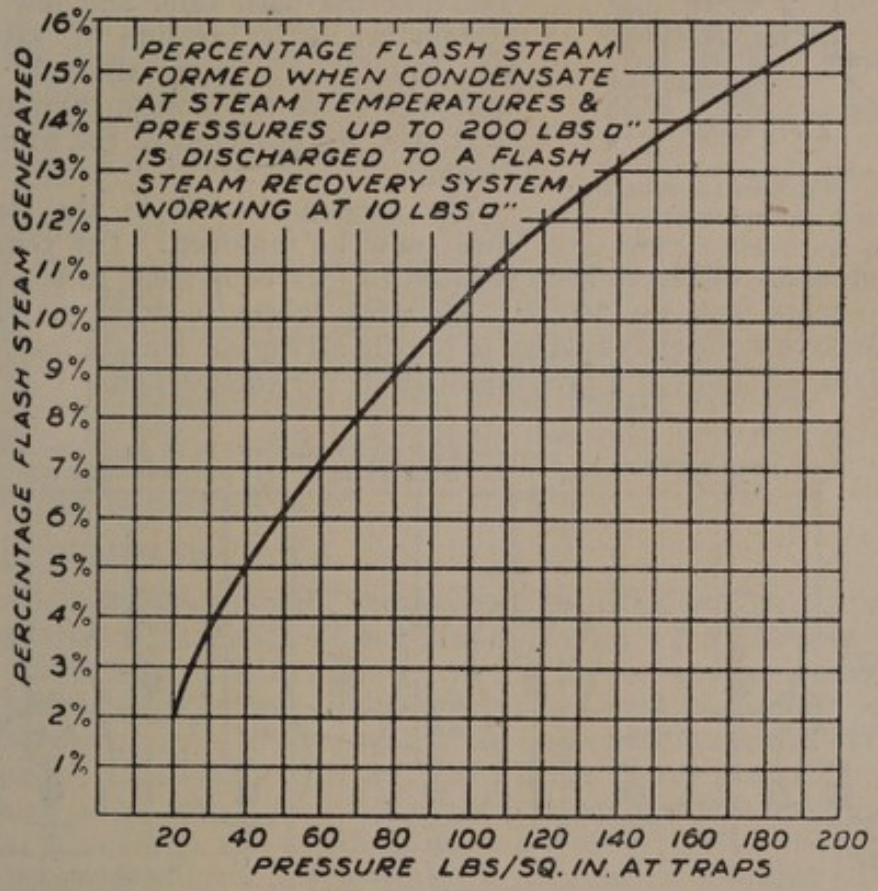

FIG. 208A. Amount of flash steam generated.

212° F. But the latent heat of the flash steam will have been lost, and this is a direct source of waste.

However, the condensate can be discharged to equipment working at a lower pressure and a lot of the flash steam recovered and put to good use. The curve (Fig. 208A) shows the amount of flash steam generated when condensate at different pressures is discharged to a heating or process system working at 10 lb. per square inch. For instance, condensate at a pressure of 200 lb. per square inch when discharged to this system will form flash steam amounting to 16 per cent.; condensate at a pressure of 80 lb. per square inch will, when discharged, form flash steam to the extent of 9 per cent.; even if it is discharged at a pressure of 40 lb. per square inch 5 per cent. flash steam is generated.

There are many types of plant on which flash steam can be used and thus

show a substantial saving of live steam. An example is given in Fig. 208B of an evaporator taking steam at a pressure of 80 lb. per square inch. The condensate from the evaporator heating surface is discharged through a steam trap into a flash tank (the nature of this tank will depend upon the installation and local working conditions ; it may mean no more than a length of large bore piping with the ends covered). From the top of this tank the connection is taken to the low pressure process or heating plant. Assuming that condensate is discharged from the evaporator heater at a temperature of 324° F. and that the flash tank and low pressure system are at a pressure of, say, 10 lb. per square inch, approximately 86 heat units per lb. of condensate supplied to the tank will be recovered in the form of flash steam.

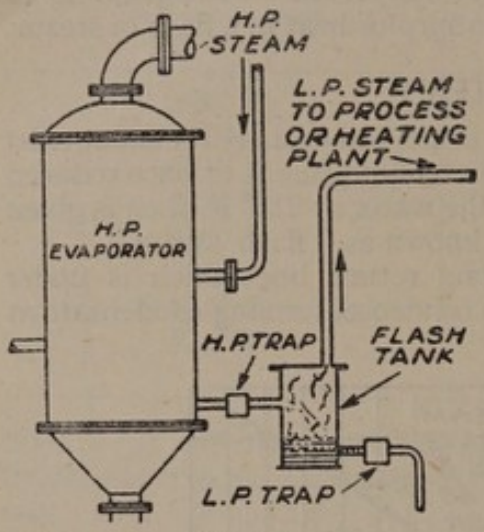

FIG. 208B. Flash steam recovery system—evaporator.

The condensate left in the flash tank must now be removed. The temperature of this condensate will have been reduced to that equivalent to a pressure of 10 lb. per square inch, i.e. 240° F. The trap chosen to drain the flash tank will need to have sufficient capacity to handle all the condensate from the high pressure plant, but now, at a pressure of 10 lb. per square inch. It is a matter

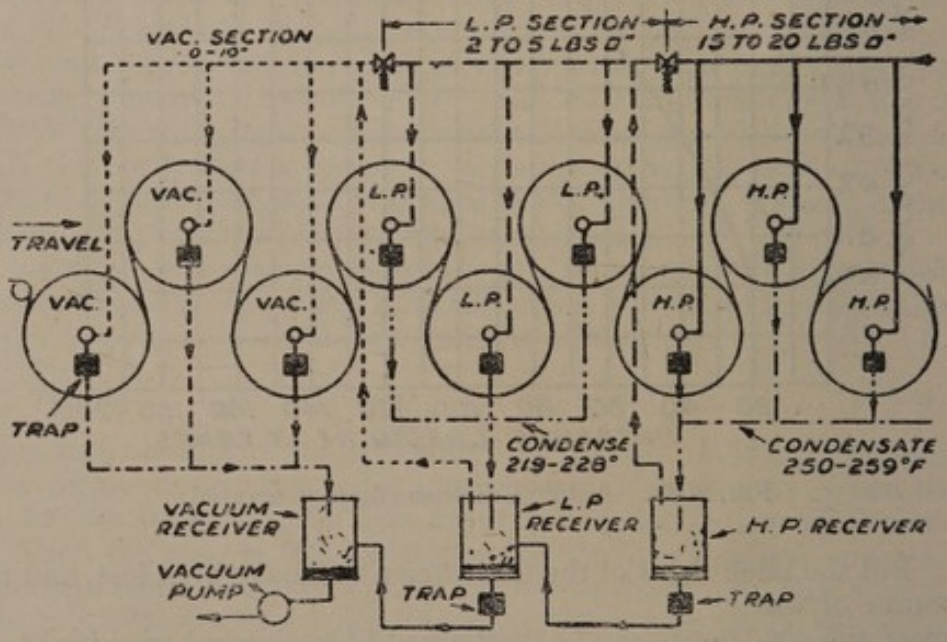

FIG. 208C. Flash steam recovery system—paper making machine.

of trap capacity and type but it is best to use a trap designed to discharge condensate as quickly as it collects. A relief valve on the flash tank is advisable.

Fairly extensive systems of flash steam recovery are in use in many industries. An example is the paper industry, where paper-making machines may be working at a pressure of from 5–20 lb. per square inch and quite near there may be air heating batteries working at pressures up to 150 lb. per square inch, or calender rolls working at even higher pressures.

On paper-making machines and other machines of a similar type the system

of flash steam recovery is carried a stage further than in the evaporator example, because generally the working pressure at one end of the machine is considerably lower than that at the other.

Fig. 208c shows a diagram of a paper-making machine ; the wet-end section of the cylinders is under vacuum ; the middle section takes steam at 2–5 lb. per square inch, and the dry-end section takes steam at a pressure of 15–20 lb. per square inch. Condensate from the drain traps at the high pressure end of the machine is collected in a flash recovery vessel which, as will be seen, is connected to the steam supply of the middle section of cylinders. The flash steam formed in the tank is utilised in this middle section. Again, the condensate from the traps draining this middle section is collected in another flash recovery vessel, this time connected to the vacuum section in which the flash steam is utilised.

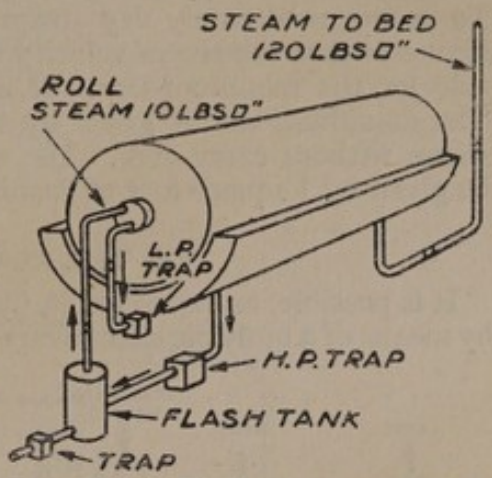

Fig. 208d. Flash steam recovery system—laundry calender or ironer.

Yet another example of flash steam recovery and a fairly simple example is given in Fig. 208d. It is a laundry ironer with a bed working at a pressure of 120 lb. per square inch condensing 600 lb. of steam per hour, and a roller working at 10 lb. per square inch. The amount of flash steam available from 600 lb. of condensate discharged at 350° F. is 71 lb. per hour, which is ample to supply the roller without any addition of live steam. This is a clear saving of more than 10 per cent. A relief valve must be fitted to the flash tank.

BLOW-DOWN FLASH

Boiler blow-down, especially from high pressure boilers, contains a great deal more heat than it can hold at atmospheric pressure. For example, each pound of blow-down from boilers working at 250 lb. per square inch will contain 200 B.Th.U. of surplus heat. This will cause a flash of over 20 per cent. If the blow-down amounts to 3 per cent. of the boiler feed the flash from blow-down will represent three-fifths of 1 per cent. of the steam generated. This may sound small (it is always dangerous to think in percentages) but the blow-down flash may represent a loss of 6 tons of coal in every 1,000 tons used. If continuous boiler blow-down can be used it is much easier to utilise the flash than if it comes in occasional short, sharp bursts. A contact heater is probably one of the best ways of absorbing such flash. The flash steam is conveyed to a large pipe down which is sprayed the boiler feed make-up water.

FLASH COOLING

Suppose there is a dilute process liquid at 200° F. and that the next process requires a temperature of 150° F. There may be a heat exchanger available. Another way of cooling is to spray the liquor into an empty vessel connected to a condenser and vacuum pump. If a 22-inch vacuum is maintained in the vessel, the steam table shows that the liquid must boil at 150° F. It will, therefore, flash off its surplus heat and reduce its temperature from 200° to 150° F. The steam table shows that each lb. of liquor must flash off 50 B.Th.U. The latent heat of steam at 22 inches vacuum is 1,008 B.Th.U., so that

$$\frac{50}{1008} = 0{\cdot}050 \text{ lb. of flash steam will be produced from each lb. of liquor.}$$

This effects a 5 per cent. concentration of the liquor, and, if the condenser

takes the form of a contact heater, process water can be heated with the flash vapour, or it can be used for space heating.

THE SIZE OF FLASH TANKS

It is difficult to lay down a general rule for the size that a flash tank should be. It must be large enough to permit the steam to separate from the water. To ensure moderately dry steam, it is desirable that it should be of such a diameter that the steam velocity does not exceed 10 feet a second. This is the rule for the minimum size. If it is too large it radiates heat unnecessarily. The maximum size is given by the area of flashing water that will give up steam without carry-over. The weight of steam in pounds per hour that can be given by 1 square foot of flashing water is three times the absolute pressure.

VII. THE RE-USE OF LATENT HEAT

It is possible, as explained in Chapter VII, to re-use the latent heat in steam by means of a multiple effect evaporator. This will be discussed later. In most

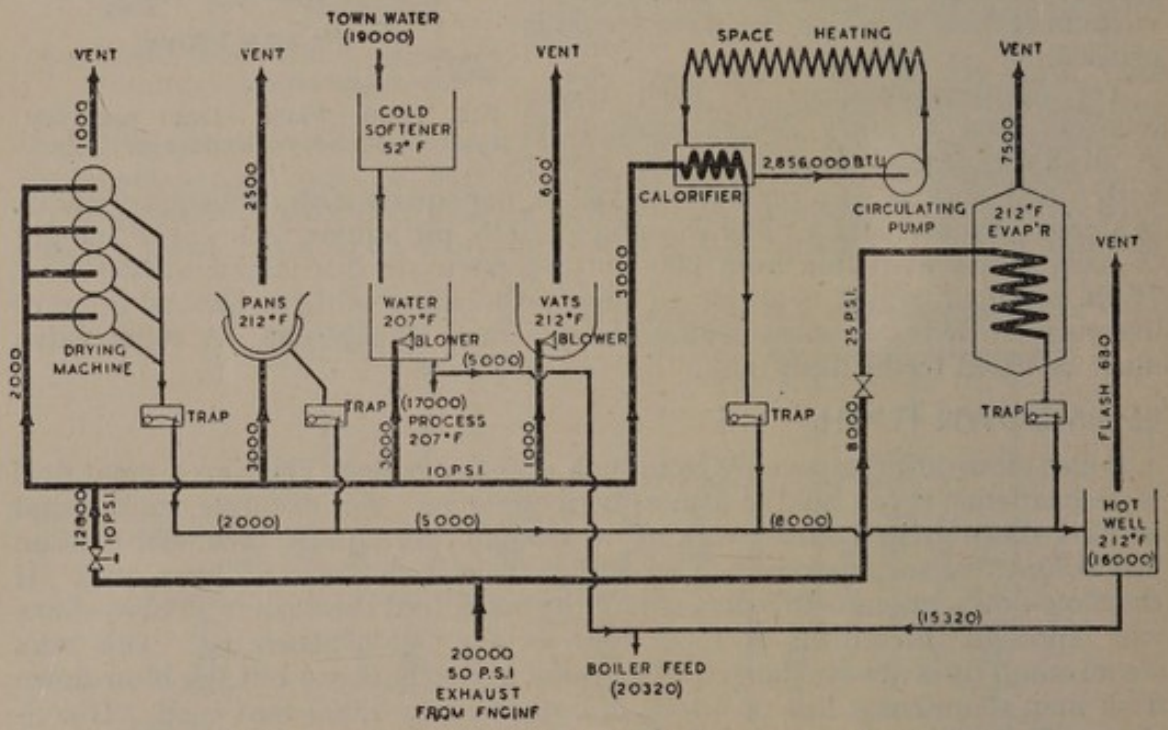

FIG. 209A. Steam layout (bad) of a hypothetical factory.

factories there is no application for a multiple effect evaporator, but the latent heat can often be used two or more times. If evaporation is part of the process so much the easier is the re-use of the latent heat.

The heat-using plant of a mythical factory is shown in Fig. 209A, which gives the approximate heat distribution in pounds of steam per hour. The figures in brackets are pounds of water. Only the steam and water circuits are shown in the figure. The pans are open-topped jacketed vessels. The vats are open-topped vessels supplied with steam by blower. The evaporator is supplied with steam at 25 lb. per square inch. The drying machine takes steam at 10 lb. per square inch.

An extensive space heating installation, for heating the factory and the warehouses, takes some 3,000 lb. of 10 lb. per square inch steam per hour. The steam goes into a calorifier through which is pumped the heating water. The boilers use the returned condensate and require, in addition, 500 gallons, or 5,000 lb. of make-up water per hour to compensate for the loss due to blow-

down for flash steam lost in the condensate hot-well and the steam used by the blowers. The factory requires 1,700 gallons, or 17,000 lb. of hot water per hour for process and cleaning. This water is heated to just below boiling point by open steam blowers using 3,000 lb. of 10 lb. per square inch steam per hour.

The factory is reorganised with the object of using the latent heat of the steam over again and thus effecting a great economy. The reorganisation is shown in Fig. 209B.

The evaporator had been designed for 50 lb. per square inch but seldom

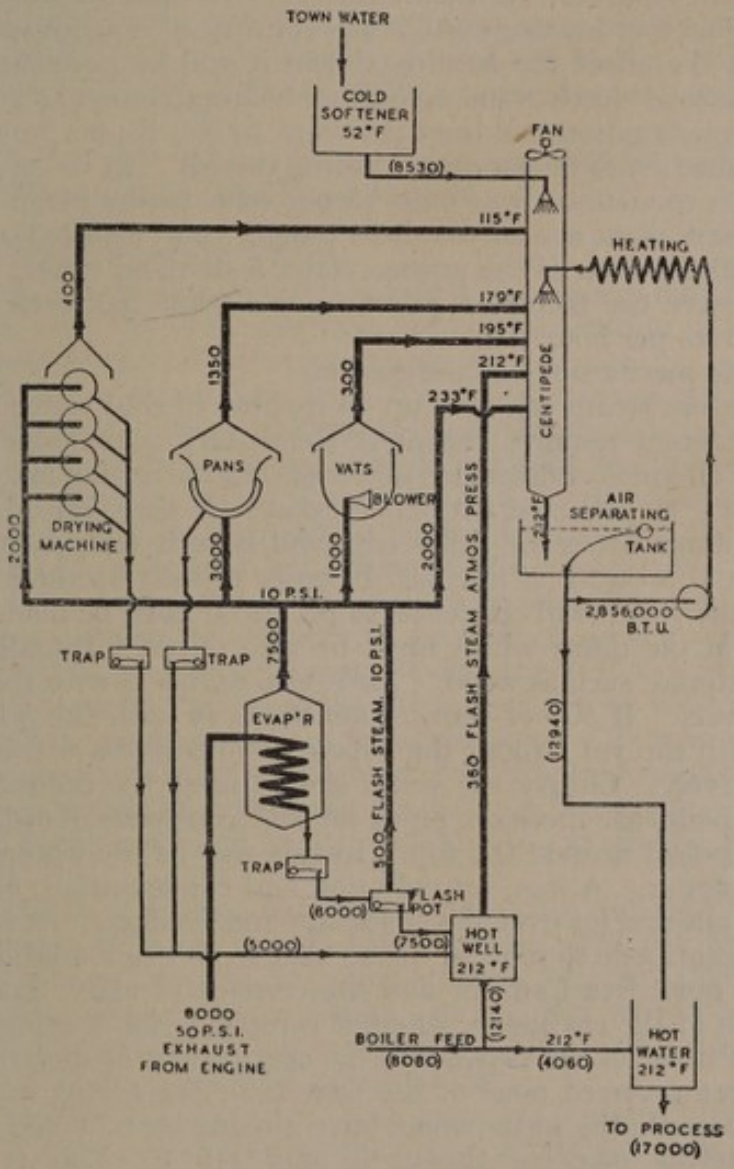

FIG. 209B. Steam layout (improved) of hypothetical factory.

carried more than 25 lb. per square inch. So it was turned into a pressure vessel using steam at 50 lb. per square inch, and boiling the product under 10 lb. per square inch instead of under atmospheric pressure. The vapour from the evaporator was piped into the 10 lb. per square inch main where it was more than sufficient to supply the drying machine, the pans and the vats.

The condensate from the evaporator was piped to a flash pot which breathed into the 10-lb. main, so that there was another source of 10-lb. steam in the event of the pans or vats making a sudden demand.

Hoods were fitted over the pans, vats and drying machine to collect the vapour coming off these vessels. As the hoods over the pans and vats must be movable

it is assumed that only about half the vapour is collected. A hood is fitted over the drying machine. It is difficult to make a good job of collecting the vapour from such a plant, so it is assumed that only two-fifths of this vapour is collected. These three lots of waste vapour, together with the excess 10 lb. steam and the flash steam from the hot-well are directed into a long vertical pipe 15 feet × 24 inches down which the cold water is sprayed. This piece of plant was christened the " centipede." It is a spray condenser.

The water leaving the bottom of the centipede goes into a tank where any entangled air can separate. It then either runs straight to process or is circulated round the factory heating system and returns to be again sprayed through the centipede. To effect the heating shown it will be necessary to circulate about 10–20 times as much water round the heating system as goes to process.

The steam consumption has been cut from 20,000 lb. per hour to 8,000 lb. per hour—a reduction of 60 per cent. Nor is this all. In the original arrangement the return condensate was only 75 per cent. of the boiler feed, whereas in the reorganised plant not only is there 100 per cent. distilled water for boiler feed but over 20 per cent. of the process water is distilled water. The cloud of vapour which gave the factory a picturesque skyline has been reduced from 12,000 to 2,000 lb. per hour.

The centipede merits some consideration.

Low temperature steam can give up all its heat to cold water by direct contact. Direct contact requires the most elementary plant which can be improvised out of all kinds of discarded material. In the drying machine much air must of necessity be collected with the vapour. If there are nine volumes of air and one volume of steam, the steam will have a temperature of 115° F. due to the partial pressure of the air. In order to heat up these nine volumes of air some of the steam will condense, so provision must be made to drain this condensate from the ducts which must be well tarred internally or made of non-rusting material, such as wood. Air is also drawn in with the vapour from the pans and vats. If 50 per cent. of air comes in with the pan vapour and 30 per cent. with the vat vapour the vapour temperatures will be 179° F. and 195° F. respectively. Clearly the water should meet the coolest vapour first. The various vapours are therefore piped into the centipede in order of temperature with the coolest nearest the top. In this way as the water gets hotter it meets hotter vapour. A loss, by radiation and condensation, of some 10 per cent. has been allowed for over the centipede and its ducts. A fan is fitted at the top of the centipede to remove the air and to induce a mild draught. The vapour that is quite free from air and therefore at or above 212° F., is piped into the bottom of the centipede, where it completes the heating.

This order of input of the vapours to the centipede is of the greatest importance. If it were reversed none of the heat from the drying machine vapour could be used because the water would have already been heated up to 190° F., whereas the drying machine vapour is only 115° F. This is an excellent example of cascade or stage heating.

Here is a table showing the improvement :—

	Steam used per hour. lb.	No. of boilers.	Town water galls./hr.	Money spent on coal and water per annum.
Original	20,000	3	1,900	£7,100
As reorganised ..	8,000	1–2	853	£2,870

In this factory all the incoming water is softened cold so that there is no trouble with scale in the centipede or in the heating lines. If hard water is not softened prior to entering the centipede two things must be done. The first is that the central heating system must be given its own separate centipede into

which the hard water never comes. As the water will gradually increase in quantity due to condensation, means must be provided for bleeding off the surplus into the hot water system. The other precaution is to provide means for guarding against the formation of great masses of scale in the process water centipede. One method is to hang chains in the centipede down which the water flows and on which scale can form. The chains can be periodically withdrawn and the scale beaten off.

It has now been shown how latent heat can be re-used by making an evaporator feed a low pressure main. This is a most effective way of re-using latent heat. It was first done by Rillieux in Louisiana in 1843—just 100 years ago.

It has been shown how flash can be collected and used in stages making the utmost use of its potential, and how the flash latent heat can be used more than once.

It has been shown how vapour badly adulterated with air can give up its heat to water if used in cascade with direct contact.

MULTIPLE EFFECT EVAPORATION

This has been briefly discussed in Chapter VII. It will now be dealt with in some detail in order to show that the correct application of simple thermal principles to sensible heating is of paramount importance.

In evaporation by steam heat only the latent heat is transferred to the product to be treated. This transferred heat is absorbed (*a*) as sensible heat to raise the product to its boiling point, and (*b*) as latent heat for vaporising water from the heated product. This water vapour from the product is at a lower pressure than the original steam, but it holds all the heat in the original steam less the sensible or water heat left behind in the condensate and any sensible heat that has been used to bring the product up to boiling point.

If now there is a use for this evaporated steam at its pressure and temperature, it can be used in the same way as steam newly generated in a boiler. It can be used for process-, water-, or space-heating ; or it can be used for evaporating water from another product or from the same product from which it sprang.

Such a second evaporating process must obviously be done at a pressure lower than the first evaporation and again the heat in the steam will divide itself into (1) latent heat transferred to the product, and (2) sensible heat retained in the condensate.

Clearly, provided pressures and temperatures are suitable, this process of passing the latent heat on and on can be done until the temperature and pressure have become too low for practical heat transfer. But at each stage the sensible heat is retained in the condensate. It will be seen below that much of this condensate heat can be recovered as flash and put back into the cycle, but it forms one of the practical limiting factors.

This successive use of latent heat for evaporation is called the Multiple Effect Principle. Evaporation in Single and Triple Effect will now be considered. For simplicity certain important assumptions will be made :—

(*a*) The plant will be considered to be 100 per cent. efficient ; i.e. no radiation or other losses will be allowed for.

(*b*) The processed material will be considered to have the physical properties of water. This eliminates complications introduced by varying specific heat, conductivity, viscosity and boiling point elevation.

(*c*) Only the liquid that is to be evaporated will be considered, not the concentrated result of evaporation.

(*d*) It is assumed that the object is to evaporate 1 lb. of water.

(*e*) It is assumed that this water enters the plant at 60° F.

(*f*) In Fig. 210A, B and C round brackets are used to indicate (Total Heat) and square brackets to indicate [Latent Heat].

SINGLE EFFECT

Fig. 210A shows straight evaporation in single effect under atmospheric pressure. The quantities and heat distribution are shown. Each top figure is the weight of steam or liquid in pounds. The next figures are gauge pressures and temperatures ; p.s.i. signifies lb. per square inch. The figures in brackets are heat contents in B.Th.U. In this first example the method of ascertaining the heat and weight distribution will be described in detail.

The heat input is in the form of saturated steam at 69 lb. per square inch absolute or approximately 54 lb. per square inch gauge. (This pressure has been selected because the temperature drop between this and atmospheric

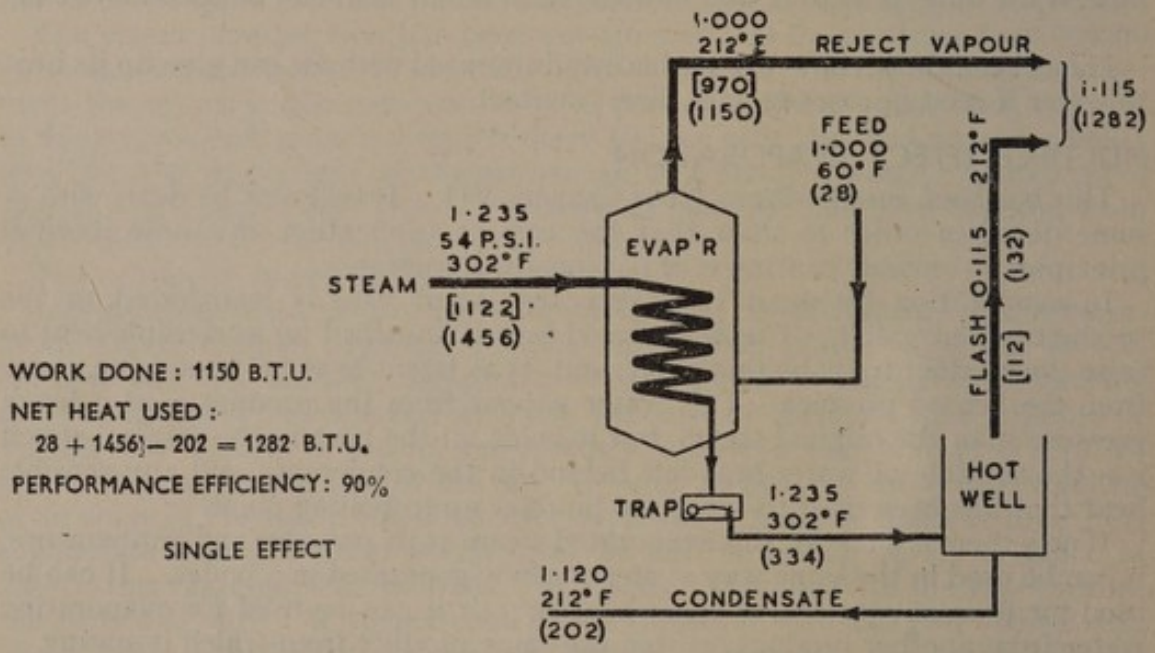

FIG. 210A. Single effect evaporator.

pressure is divisible by three into a round number.) One pound of steam at 54 lb. per square inch has a temperature of 302° F., contains [909] B.Th.U. of latent heat and (1,180) B.Th.U. of total heat. During heat transfer a constant pressure exists inside the heating surface (shown in the diagram as a coil) so that, if the coil is perfectly drained, only the latent heat in the heating steam is transferred to the liquid being evaporated. Now, 1 lb. of water is to be evaporated and it enters the plant at 60° F. It must first be raised to 212° F. requiring 152 B.Th.U. It then needs [970] B.Th.U. to evaporate it—a total of (1,122) B.Th.U. As only the latent heat of the 54 lb. per square inch steam is used it will be necessary to bring in [1,122] B.Th.U. of latent heat. As the latent heat of 1 lb. of 54 lb. per square inch steam is [909] it will be necessary to bring in

$$\frac{1,122}{909} = 1{\cdot}235 \text{ lb. of steam.}$$

This will contain $1{\cdot}235 \times 1,180 = (1,456)$ B.Th.U. of total heat. The condensate passing through the trap will contain the total heat less the latent heat which has been given up, $1,456 - 1,122 = 334$ B.Th.U. When this condensate reaches the hot-well or condensate tank, at atmospheric pressure, the excess heat will cause a flash of steam. One pound of water at 212° F. contains (180) B.Th.U., so that 1·235 lb. of water at 212° F. will contain (222) B.Th.U. The surplus heat in the condensate is, therefore, $334 - 222 = [112]$ B.Th.U. To vaporise

1 lb. of water at 212° F. takes [970] B.Th.U. So [112] B.Th.U. will vaporise

$$\frac{112}{970} = 0{\cdot}115 \text{ lb.}$$

Steam at 212° F. contains (1,150) B.Th.U. of total heat per lb., so that 0·115 lb. will carry away $1{,}150 \times 0{\cdot}115 = (132)$ B.Th.U. The condensate returning to the process weighs $1{\cdot}235 - 0{\cdot}115 = 1{\cdot}120$ lb. and contains $334 - 132 = (202)$ B.Th.U.

The work to be done in the evaporator is the transference of the energy needed to heat the liquid to 212° F. and then to vaporise it at atmospheric pressure. This requires (1,150) B.Th.U., of which (28) is already present in the feed. The heat taken in in the steam was (1,456) B.Th.U. The returned condensate contained (202) B.Th.U. so that the net heat used was $1{,}456 - 202 = 1{,}254$ B.Th.U. plus the heat in the feed, which tallies with the heat rejected, $1{,}254 + 28 = 1{,}282 = 1{,}150 + 132$. So the efficiency can be said to be

$$\frac{1{,}150 \times 100}{1{,}282} = 90 \text{ per cent.}$$

MULTIPLE EFFECT

Latent heat can be used more than once for evaporation by closing in the evaporator and taking the vapour, previously rejected, as heating medium for use in a second evaporator. If the second vessel is to boil under atmospheric pressure the steam supplied to it as vapour off the first vessel must be under pressure. The two vessels will reach an equilibrium automatically. Assume the second vessel is open to the atmosphere, then, when steam is first admitted to the first vessel's heating surface, boiling will be vigorous in the first body and non-existent in the second body. Pressure will build up in the first body and will thus reduce the temperature drop between the input steam and the output vapour and the evaporation will slow down. At the same time a pressure and temperature difference is building up between the heating vapour and the liquid in the second vessel and evaporation will start with, of course, condensation of the first-effect vapour on the second-effect heating surface.

If the rate of evaporation should rise in the first vessel its body pressure will rise and automatically reduce its rate of evaporation with an equivalent temporary increase in the rate of evaporation in the second vessel. Thus each vessel controls the rate of evaporation in the other and the self-regulation is perfect.

In the examples immediately following it is assumed that equilibrium conditions will be such that there will be an equal temperature drop across each vessel. This is not necessarily so in practice. It depends on what is called the coefficient of heat transfer and the sizes of the heating surfaces. The higher the temperature the higher the rate of heat transfer. The more concentrated the liquid being evaporated the lower the rate of heat transfer. In some quadruple effect evaporators the temperature drop in equilibrium in the last effect is nearly four times what it is in the first effect. This assumption of equal temperature drops is quite fair for illustrating the principle and for driving home the technique because the various heating surfaces can be so dimensioned as to secure any temperature drop desired in any effect.

TRIPLE EFFECT

Fig. 210B shows a triple effect arrangement. The heat distribution throughout is shown. This has been ascertained by starting with 1 lb. of 54 lb. per square inch steam, working through the plant and then reducing all quantities

to give 1 lb. of evaporation. The total temperature drop is still 302 — 212 = 90° F. which must now be split into three drops of 30° F., giving a temperature and pressure of 272° F. and 28·5 lb. per square inch gauge in the first body and 242° F. and 11 lb. per square inch in the second body. The total steam input has been reduced to 0·500 lb. ; a saving of 60 per cent. on single effect. The net heat used is 413 B.Th.U. ; a saving of 68 per cent. on single effect. The performance efficiency is 278 per cent.

It will be noticed that from a net heat-usage point of view triple effect is three times as good as single effect. From a steam consumption point of view, however, triple effect falls well short of three times as good as single effect. The reason for this apparent discrepancy is that the figures of steam consumption do not take into account the heat retained by and returned with the condensate.

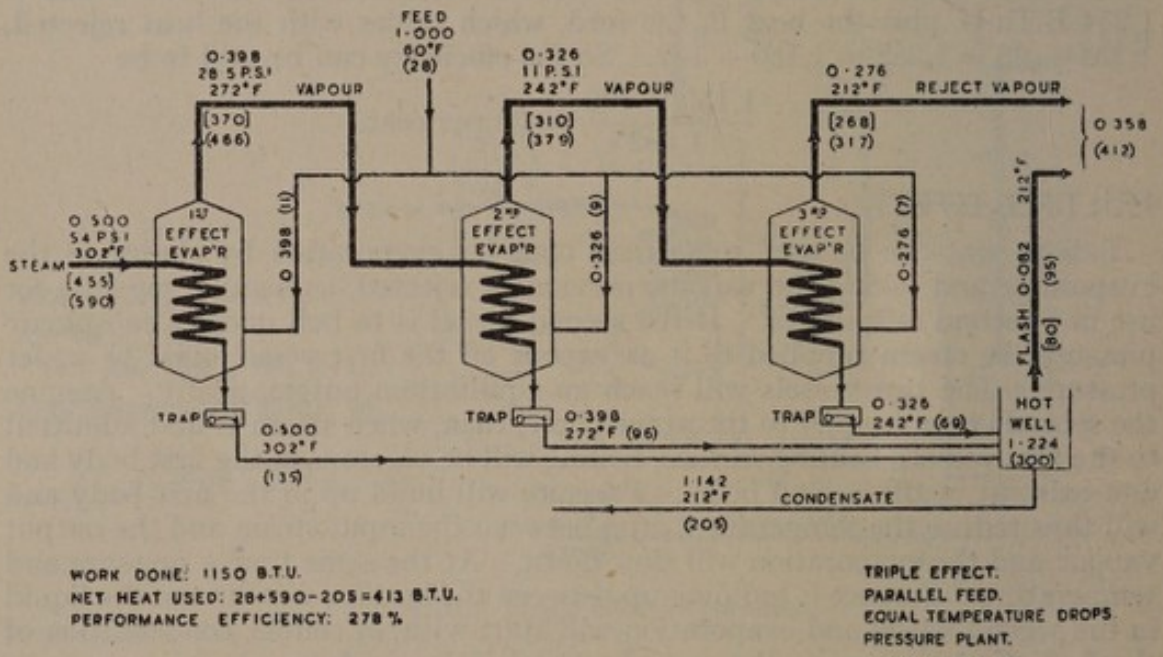

FIG. 210B. Triple effect evaporator—parallel feed.

PERFORMANCE EFFICIENCY

Can a plant be said to be more than 100 per cent. efficient ? The efficiencies of different plants are compared by expressing the energy made available for useful purposes as a percentage of the net heat or energy input.

Thus the efficiency of a turbo-generator is $\frac{\text{Electrical power generated}}{\text{Heat in steam} - \text{Heat in condensate}}$.

The efficiency of a boiler is expressed by $\frac{\text{Heat in steam} - \text{Heat in feed water}}{\text{Heat in coal}}$

It is rational to express the efficiency of an evaporator as

$$\frac{\text{Energy needed to heat and evaporate liquid}}{\text{Net heat supplied to plant}}.$$

Of course the true thermodynamical efficiency of an evaporator must be zero because all the heat in the evaporated vapour is rejected, but to use a yard-stick which always measures zero is absurd. As an example a plant producing distilled water may be taken, and it may be assumed that it has up-to-date coal-fired stills. The management will certainly measure the efficiency of their stills as if they were steam boilers (as indeed they are), and the efficiency may well exceed 80 per cent. If these stills are converted into double effect vessels

it is quite obvious that the efficiency will have been approximately doubled and must exceed 100 per cent. For better or worse, therefore, this method of expressing performance efficiency is used here.

CONDENSATE

There is another benefit apart from heat economy in the use of multiple effect, namely, the extra proportion of condensate produced and available for boiler feed or process purposes. The condensate relative to the steam used is shown below :—

	Steam used.	Condensate.	Excess condensate.
Single effect ..	1·235	1·120	— 10 per cent.
Double effect ..	0·680	1·136	+ 67 ,, ,,
Triple effect ..	0·500	1·142	+ 128 ,, ,,

The condensate from the first vessel is pure distilled water equal in weight to the input steam so that apart from loss by flash it can look after the boiler feed. The condensate from the other vessels is distilled water, possibly slightly contaminated by the solute in the solution being evaporated. This contamination may preclude its use as boiler make-up but almost certainly is no bar to its use for process purposes.

There are many industries where distilled water for process would be very valuable. These industries may deplore their lack of distilled water, yet their plants may contain many reducing valves. A reducing valve can always be replaced by an evaporator which can produce distilled water for practically no cost other than capital interest. In many factories steam is raised at 60 lb. per square inch and used at 10 lb. per square inch. The corresponding drop is 68° F. This is ample drop for a quadruple effect still which will produce distilled water of about three times the weight of steam passed through it.

LOSSES

The loss by radiation and convection from a well-lagged evaporator is very small. There are few published convection and radiation loss figures. The most frequently quoted are those taken by Kerr referring to a multiple effect sugar liquor evaporator. They are :—

No. of effects	Per cent. initial steam lost by radiation	
	Unlagged	Well lagged
2	1·06	0·26
3	4·20	1·05
4	9·80	2·70

It is important that the first effect should be well lagged because any heat lost in the first vessel is lost in all vessels. The lagging can be progressively less elaborate towards the final effect.

Another loss is caused by the necessity for venting the heating surfaces to remove air or other incondensable gases. This is generally done by a small vent blowing from one heating surface to the next, simply blowing into the vapour space. This loss, unlike radiation loss, is not a multiple effect loss. Any steam blown over with the air only loses one effect. But in an endeavour to make sure of adequate venting a great amount of steam is often wasted this way. It may be better to vent the heating surface by means of a thermostatic air vent device if the pressure inside the heating surface is above atmospheric pressure.

A very rough guide as to the probable percentage losses from all causes in a well-designed well-lagged plant is obtained by multiplying 0·3 by the square of the number of effects.

FEED ARRANGEMENTS

In Fig. 210B each vessel is fed with its quota of liquor. This is called parallel feed and is seldom used for evaporation. If the feed is all put into one vessel and passed in succession to the others it is called " forward " or " backward " feed depending on whether it is in the same direction as or against the flow of heat. If these different feed arrangements are worked out in detail they show the following results :—

Feed	Net heat used	Performance efficiency
Parallel	340 B.Th.U.	328 per cent.
Backward..	334 ,,	344 ,, ,,
Forward	488 ,,	236 ,, ,,

Backward feed is most efficient because all the feed is preheated by second effect vapour, whereas all the feed is preheated by virgin steam in forward feed.

CHOICE OF FEED METHOD

It is unfortunate that other considerations frequently compel the adoption of forward feed. For example, sugar liquors can tolerate fairly high temperature

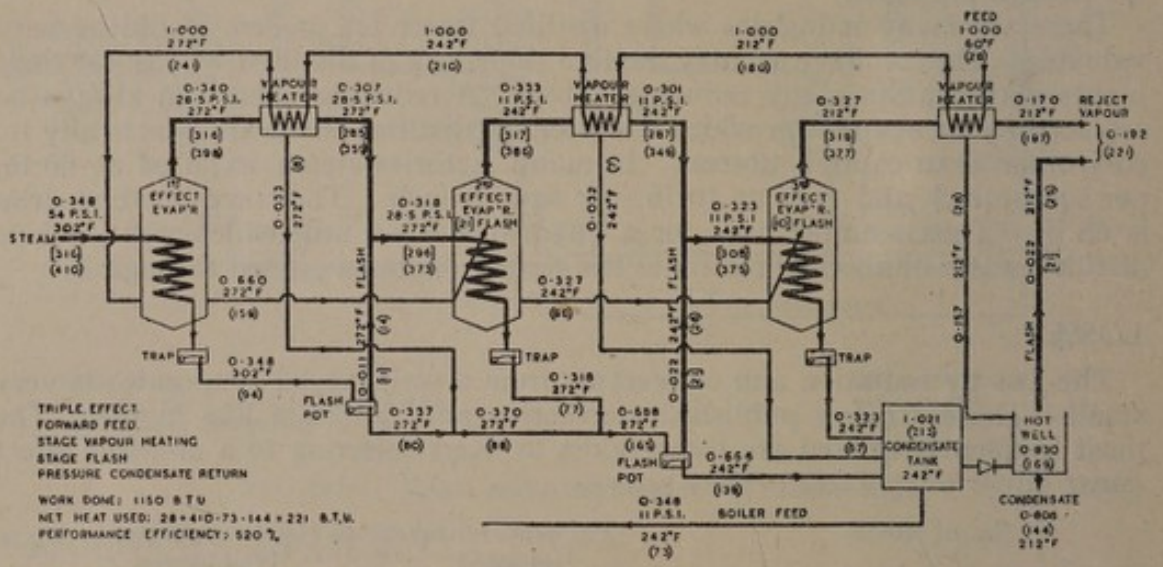

FIG. 210C. Triple effect evaporator—forward feed.

for a short time in dilute solution, but they rapidly deteriorate if raised to high temperature in concentrated solution. Almost every multiple effect sugar liquor evaporator works on forward feed for this reason. The other great practical benefit of forward feed is that only one feed or extraction pump is required as the feed flows naturally from vessel to vessel by reason of the pressure difference. In backward feed a pump is needed for each effect.

The difference in the feed heating in forward and backward feed gives the clue to the logical development of the technique. It is also desirable to make use of the flash from the condensate. The result is shown in Fig. 210C.

It is clearly impossible to work out the heat distribution by simple arithmetic. It is necessary to take in 1 lb. of feed and x lb. of steam. A long, tiresome, but quite straightforward analysis will give the solution.

The steam consumption has been reduced to 0·348 lb., and the net heat usage to 221 B.Th.U. giving a performance efficiency of 520 per cent.

This result is so remarkable that it merits detailed consideration. Plain forward feed gave a performance efficiency of 236 per cent. This has been more than doubled by the simplest of technique and comparatively simple plant.

It must be noted that most of the increase in performance is obtained by simple modifications of the sensible heating and is more than the increase obtained by going from single to double effect. This is the real lesson to be learnt.

The gain in sensible heating is due to the fact that in the example in Fig. 210c, the feed is introduced at 60° F. In practice, of course, the feed enters the process hot from a previous process. All too often this preliminary heating has been effected by live steam derived from the boiler. This practice is wrong, and whatever may be the method of heating in the previous process, the feed could have been passed to that process through the evaporator vapour heaters. This marriage of unrelated processes is one of the most fruitful sources of thermal economy.

NUMBER OF EFFECTS

The chief limit to the number of effects is that imposed by the elevation of the boiling point of concentrated solutions. Thus a 70 per cent. sugar solution has a boiling point elevation of approximately 9° F. ; a 70 per cent. caustic potash (KOH) solution is some 200° F. There must be a temperature drop across the heating surface, but the vapour coming off the solution is not usable in the next effect at the elevated boiling point corresponding to the pressure under which the solution is boiling. This sets a practical limit to the number of effects that can be operated for a given total t mperature drop. The limit in the sugar industry is about six effects, while for evaporating caustic it is seldom possible to use more than two effects.

The other limit is the necessity for maintaining a vigorous circulation. Good circulation greatly increases the rate of heat transfer and hence the rate of evaporation. It also prevents overheating which can occur in the bottom of an evaporator where the hydrostatic head of the liquid raises the boiling point with possible damage especially to organic products. In a water still, boiling point elevation is entirely absent. There is no increase of viscosity to slow up circulation and there is no reason why a water still should not operate with temperature drops of 5° F. Temperature drops of 9° F. or 10° F. are not uncommon in the sugar industry where boiling point elevation, viscosity and damage due to stagnant overheating of an organic product are all present dangers.

LIMITATIONS

The conditions which have governed the foregoing examples are somewhat idealised. In practice the efficiencies given for multiple effect working will never be reached for a variety of reasons ; for example, the liquids handled industrially are never pure water and only a certain percentage of the liquor is evaporated as water vapour. Even when making distilled water some of the original feed containing the unwanted impurities is discharged as blow-down. In consequence slightly more vapour must be used for heating the feed and slightly more steam per pound of water evaporated than the ideal figures indicate.

Many industrial liquors have scaling properties. This results in it being necessary to keep the temperature drop across each effect above a certain minimum in order to obtain a reasonable output from a dirty, scaled plant. This immediately limits the number of effects that can be employed—in extreme conditions only single effect is possible.

It is not usually possible to heat the feed in vapour preheaters to the same temperature as the heating vapour ; the liquor temperature will generally be a few degrees below the vapour temperature. Another limitation is imposed if the liquors are corrosive. For example, it may be necessary to construct the plant of nickel or stainless steel. It must be realised that a double effect is

twice as large as a single effect—" multiple effecting " exactly multiplies the size of plant. Financial considerations may be such as to preclude the use of multiple effect. The vapour also might be corrosive which would not only damage the plant but give unsatisfactory condensate. These are some of the limitations that may await the uninitiated. If such conditions are likely to be present, expert advice should be sought.

THERMO-COMPRESSION

There is another method of re-using latent heat which, although of limited application must not be overlooked ; this is the boosting of low pressure vapour, or exhaust steam, up to a slightly higher pressure by mixing it with high pressure steam in a suitable injector. The resulting mixed steam has a pressure higher than the original vapour and can therefore be used again. This is called thermo-compression and can be done with great efficiency when the right conditions exist. These conditions are that the steam or vapour to be boosted should be at or about atmospheric pressure, and that only a small increase of pressure should be attempted.

Suppose an evaporator is taking steam at 5 lb. per square inch and is evaporating under atmospheric pressure, then 3 lb. of vapour at atmospheric pressure can be brought back to the injector, where 2 lb. of steam at 150 lb. per square inch will boost the 5 lb. of mixture to 5 lb. per square inch. In other words, where 5 lb. of vapour is being evaporated, 3 lb. can circulate indefinitely. Two lb. of 150 lb. per square inch are used and 2 lb. of atmospheric vapour are rejected to do 5 lb. of evaporation, which is not very different from the performance of a triple effect evaporator. Under such circumstances plant that costs little more than single effect cost has nearly the same efficiency as a triple effect plant at nearly three times the capital cost.

Apart from the narrow limits of vapour pressure within which thermo-compression is applicable the great disadvantage is that high pressure steam must be used.

It is sometimes economical to re-compress steam by means of a mechanical compressor-turboblower. This is particularly applicable where steam is costly and power very cheap. It is seldom economic in Britain.

VIII. PEAK STEAM LOADS

In most plants the demand for steam fluctuates widely due to the large number of individual consumers and the intermittent nature of the different processes. In a dye works, for example, the dye vats are many, they vary in size and are operated more or less haphazardly. Apart from this the dyeing operation is carried out in two stages each demanding steam at a different rate over a varying period of time. Shift working, meal periods and overnight shut-down add to the complexity. In a steel works, as the rolling mills and steam presses create short sharp steam demands far in excess of the average, these are superimposed upon slow variations. The conditions are similar in mines and collieries due to the intermittent operation of the winding engines.

The Nature of the Steam Demand in Various Plants. Fig. 211 shows at A, B, C and D the total boiler load in typical industrial plants as obtained by steam metering. From the charts it will be seen that the peaks usually develop and fall off with considerable rapidity, they may rise to double the average demand or more and may last anything from a few seconds to an hour or longer, they have no characteristic shape, they change from hour to hour and from day to day, they occur at odd intervals and are, in most cases, impossible to predict. They have, however, one thing in common—they impede production and increase the heat consumption per unit of output.

Peak Loads in the Boiler House. No steam raising plant, of whatever type,

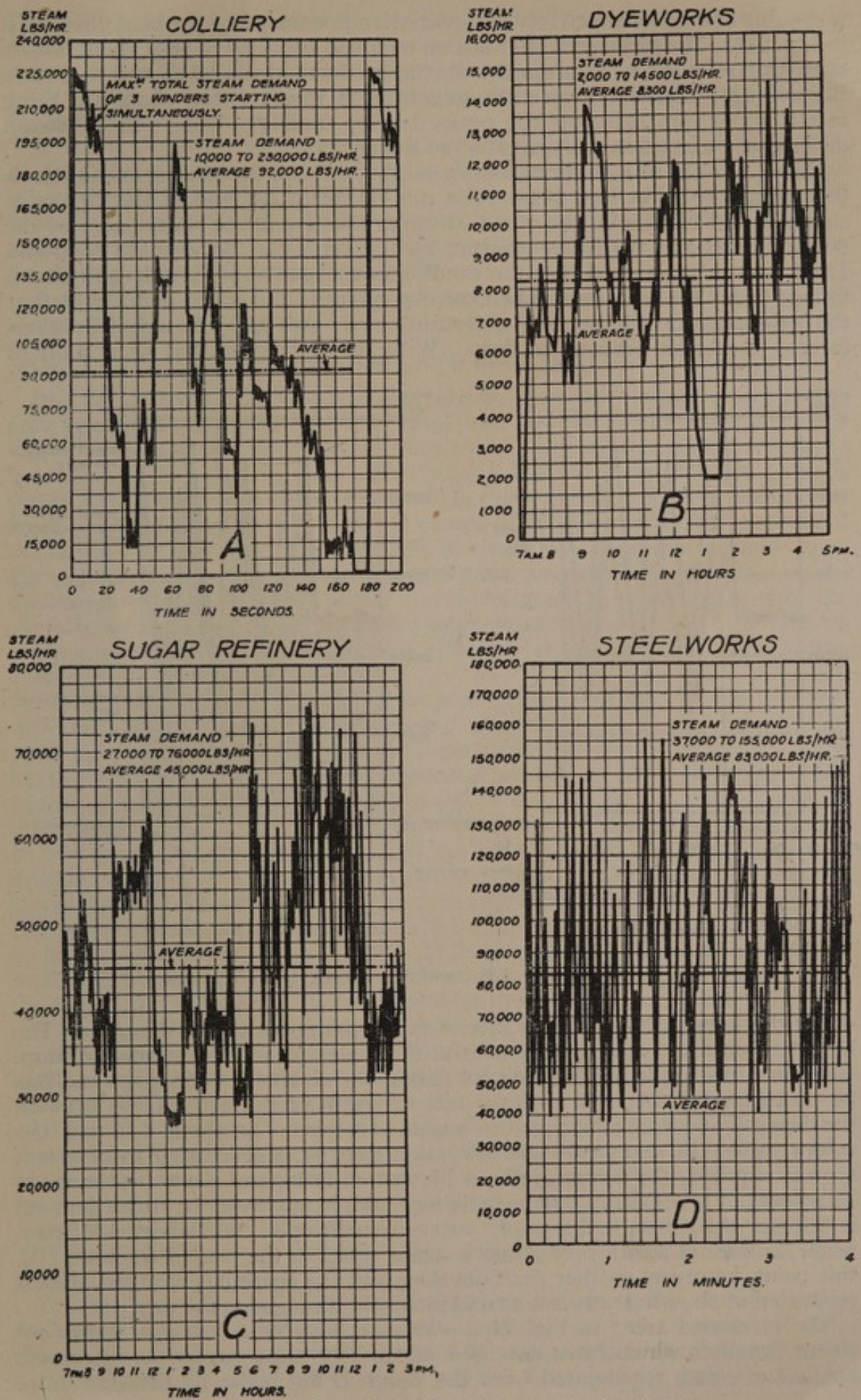

FIG. 211. Examples of fluctuating steam demands.

is capable of dealing effectively with variation in demand of the kind illustrated in Fig. 211. The ordinary boiler has an overload capacity of only about 25 per cent. to 30 per cent., whereas the maximum demand may well run to double the average. Perhaps even more important is the fact that change in firing rate is a slow process while the peaks develop rapidly. Valleys in the steam demand are almost as embarrassing as the peaks.

What happens in a boiler house where the load is a fluctuating one is shown diagrammatically in Fig. 212A. The rate of steam demand of the factory is shown by the full line A, the rate of steam production in the boiler house by the dotted line B and the boiler pressure by the curve C. Between 12 noon and 12.45 p.m. it is assumed that the conditions are reasonably steady and that a rate of firing has been established corresponding to the steam demand so that the boiler pressure is reasonably well maintained. At 12.50 p.m. a peak develops causing the boiler pressure to drop. With this indication that he is short of

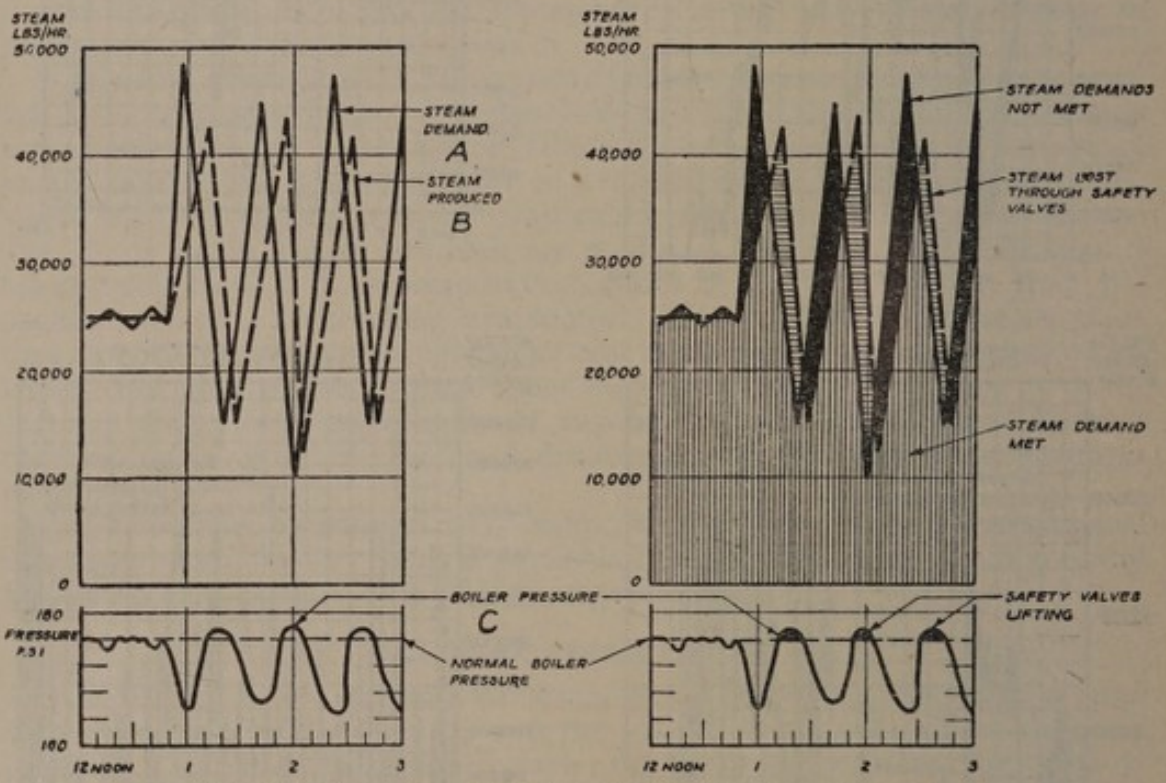

FIG. 212 (A and B). Effects of fluctuating steam demands.

steam the fireman increases his rate of firing and opens up his dampers. The boiler is, however, slow to respond and the boiler pressure continues to drop. Meantime, the peak has reached its maximum at about 12.55 p.m. and has begun to fall off. At 1.05 p.m. the rate of steam production catches up with the demand and the boiler pressure begins to recover. At about 1.15 p.m. the safety valves lift and, with this indication that he is generating more steam than is required, the fireman reduces his rate of firing and closes his dampers. This does not take effect for some little time and, in any case, the steam demand is falling. As a result, the safety valves remain open until about 1.20 p.m. when the rate of steam production is again equal to the rate of demand. At this point, however, another peak develops and the chase begins again, with a repetition of the whole process at odd intervals throughout the day.

The blackened areas in Fig. 212B—which is Fig. 212A redrawn—represent steam demands which have not been adequately met—except for the small amount of steam regenerated from the water in the boiler by pressure drop.

They also represent loss of output due to the inability of the boiler plant to meet the peaks. The horizontal cross-hatched areas show the amount of surplus steam blown to the atmosphere through the safety valves, while the vertical cross-hatched areas show the proportion of the total steam demand which is met.

In the boiler house maximum efficiency can only be attained under steady load and with proper attention paid to combustion and draught control. Where the steam demand is fluctuating, frequent changes in firing rate have to be made against a rising or falling pressure gauge, leaving little time or opportunity for controlling combustion. Safety valve loss has to be added. This is far more important than is usually assumed.

Peaks and valleys in steam demand, moreover, give rise to steam pressure variations which result in loss of output and further heat loss in the factory.

FLUCTUATING STEAM PRESSURE—PIPE LINE EFFECT

In a boiler house the fireman takes his cue from the pressure gauge as the only instrument normally available to tell him how the steam demand is varying. In a factory with intermittent processes, the first effect of a sudden demand for steam by any one consumer is to cause a local drop in pressure. This is transmitted back to the boiler house and is there translated into a demand for more steam. On its way, however, it causes pressure disturbances throughout the entire pipe system and, as the flow of steam to a consumer depends upon the pressure difference between the consumer and the supply main, all the consumers then in operation suffer steam shortage. To counteract this, the operatives in the various departments open steam valves under their control, causing a still greater demand and a still greater pressure drop, thus aggravating the difficulties.

The steam mains play an important part in peak loads. A simple example will illustrate the point. The steam main from the boiler house to the process plant in a factory is 500 feet long, 10 inches diameter and is designed to carry 24,000 lb. of saturated steam per hour at a pressure of 25 lb. per square inch gauge. Under these conditions the pressure drop along the main would be rather less than 2 lb. per square inch. If the rate of flow were increased to 48,000 lb. per hour, not an unusual condition, and if the initial pressure dropped to 20 lb. per square inch under the impact of this peak, the pressure loss along the main would be increased to approximately 8 lb. per square inch. Under these conditions steam would be supplied to process at a pressure of 12 lb. per square inch, instead of, say, 23 lb. per square inch. Loss of pressure to this extent would obviously slow down the whole department, apart from the fact that it would still further stimulate the scramble for steam.

" Steam theft " of one consumer from another, as described, is of common occurrence in industry and the consequent loss of output is important. Delay in the completion of a process may affect the quality of the finished article and wastage and spoilage of material on this account may be considerable. In addition, the quality of some products is affected by variations in temperature caused by change in steam pressure, e.g. vulcanised rubber, milk products, etc., while in the dyeing of fabrics, correct colour development may be dependent upon accurate temperature control which can only be obtained if the steam pressure is reasonably constant.

Peak demands are complicated when the process is such as to call for a fluctuating steam quantity.

In brewing and dyeing, for example, the process is, in effect, a double process, i.e. boiling up, which requires steam in quantity for a short period of time, followed by the more leisurely brewing or dyeing operation itself, during which only sufficient steam is necessary to replace heat lost by radiation. This

condition obviously introduces further complication, particularly as the first part of the process, i.e. boiling up, is non-productive and should be carried out as quickly as possible.

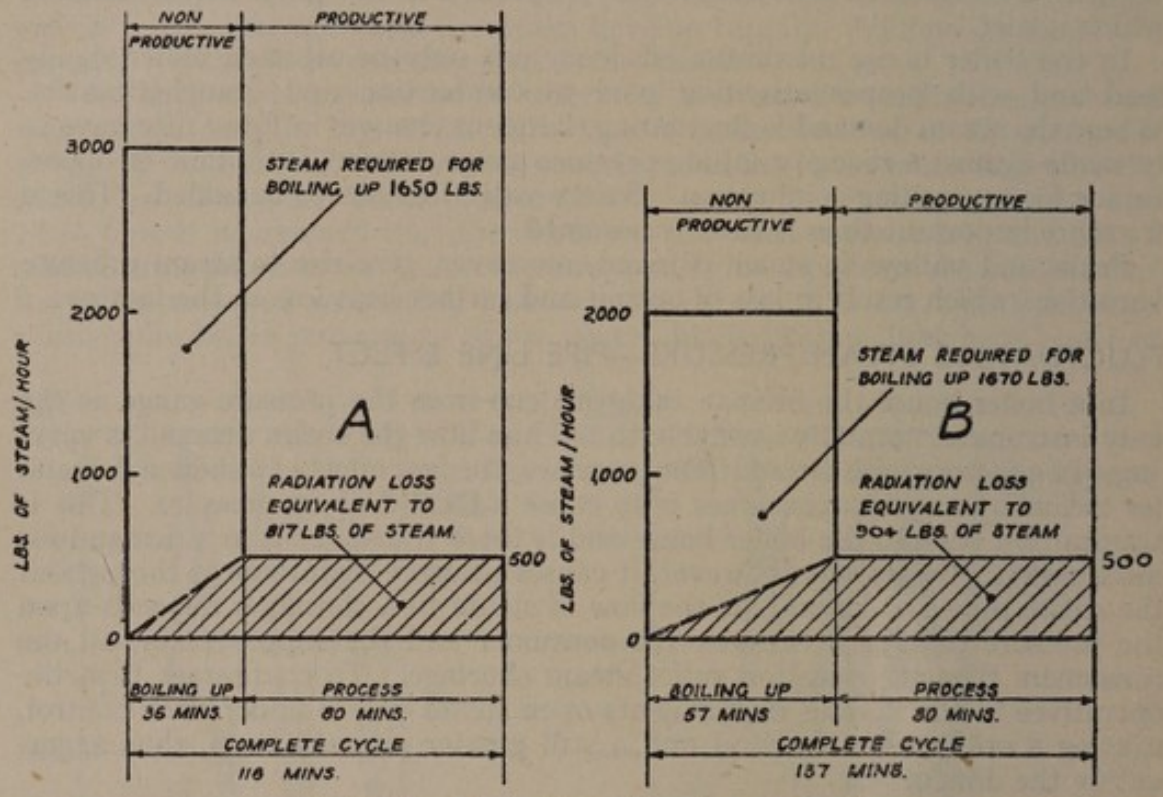

FIG. 213A. Cycle of operations in a dye vat.

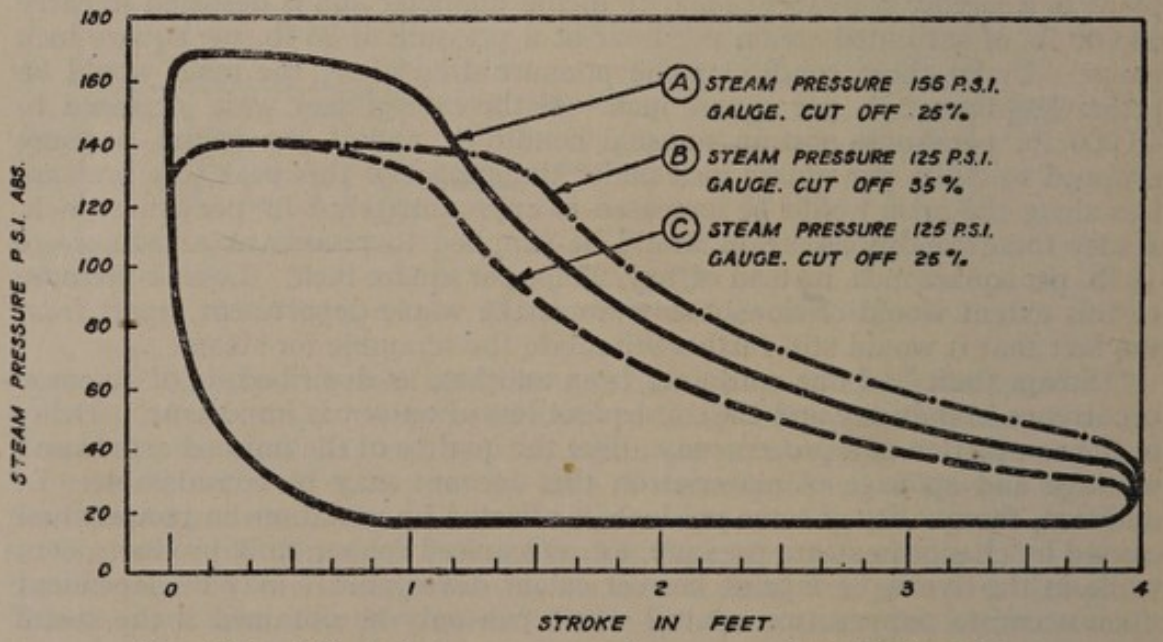

FIG. 213B. Effect of steam pressure drop on a reciprocating engine.

The attempt to do so, with several units competing for steam, may, however, actually delay the process instead of hastening it. The importance of this is illustrated in Fig. 213A, which shows on the left hand side the total normal steam demand throughout the cycle of operations in a typical case of this kind, that of a dye vat. It assumes that steam is required at the rate of 3,000 lb.

per hour for boiling up and that the vat contains 1,000 gallons of water at 50° F. to be raised to 212° F. Allowing for radiation, the time required to boil up would be thirty-six minutes. Assuming that the dyeing process itself occupies eighty minutes, the complete cycle would take 116 minutes.

If, due to competition for steam, the rate of steam supply to the vat during boiling up were reduced from 3,000 to 2,000 lb. per hour, as shown on the right hand side, the time taken to boil up would be increased from thirty-six to fifty-seven minutes and the complete cycle extended from 116 to 137 minutes. This corresponds to a loss of output of 15 per cent.

STEAM EMPLOYED TO DO MECHANICAL WORK

Steam may be employed for the generation of power and exhaust may take place, either to the atmosphere or to a condenser. A variant is the back pressure power plant exhausting at a pressure suitable to the use of the exhaust steam for heating or for process.

Where steam is used in heavy engines exhausting at atmospheric pressure as, for example, in many steel works and collieries, boiler pressure variation affects their performance to a much greater extent than it does in a condensing engine, as the pressure range available for generating power is only about 50 per cent. of what it is in a condensing unit. In many plants of this type, a cut-off as late as 75–90 per cent. is, in fact, adopted, to avoid undue loss of crank effort during periods when the boiler pressure is low. This is wasteful, as the steam is not expanded properly in the engine cylinder, and the energy loss at exhaust is excessive. It is by no means uncommon to find the boiler house in difficulty, the engines flat out on reduced steam pressure, while in spite of it all the engine is not taking its load properly, because it has pulled the pressure down. The use of late cut-off for the purpose of maintaining crank effort when the boiler pressure is low may, therefore, defeat its own object by increasing the peaks and still further reducing the steam pressure.

The general effect of drop in steam pressure in a reciprocating engine exhausting at atmospheric pressure is illustrated diagrammatically in Fig. 213B, which shows at A the ideal indicator diagram for a steam cylinder assumed to have a diameter of 30 inches and stroke of 48 inches with an inlet pressure of 155 lb. per square inch and a cut-off of 25 per cent. In such an engine the steam consumption would be approximately 1·92 lb. per stroke assuming a clearance volume of 5 per cent. and neglecting the effect of cylinder condensation. If, due to the occurrence of a peak load on the boilers, the admission pressure were to drop to 125 lb. per square inch, the cut-off would have to be increased to about 35 per cent. for the same amount of work done, as shown at B. This would raise the steam consumption to approximately 2·24 lb. per stroke ; an increase of 16 per cent. The effect of steam pressure variation upon the performance of an engine of the non-condensing type is, therefore, most important and, as each drop in pressure is accompanied by an increase in steam demand, the boiler pressure is liable to fall off at an increasingly rapid rate if the operation of the engine under these conditions is prolonged. Large artificial peaks are often built up in this way to the great embarrassment of the boiler house.

Where the cut-off is fixed, as it is in many engines, reduction in admission pressure to the above extent would reduce the work done per stroke by about 25 per cent., as shown at C, slowing down the engine and upsetting production.

The generation of power by process steam, using back pressure engines or turbines, offers important economic possibilities in industry, but the extent to which this method of generating power can be employed in a works depends upon the existence of a reasonably constant basic demand for steam at low

pressure. Heavy peaks and valleys are, therefore, detrimental to the use of the system. For example, due to wide variations in steam demand, it might not be possible to count upon more than a steady 10,000 lb. of steam per hour for back pressure power generation. With a pressure range of 200–25 lb. per square inch this would generate about 256 kW. With lesser peaks and valleys and a basic steam quantity of 15,000 lb. per hour the power output could be stepped up to 385 kW.

Again, with this type of power generating plant, the maintenance of power output depends upon the maintenance of a reasonably constant pressure difference across the engine or turbine. If, in the above example, the boiler pressure dropped from 200–150 lb. per square inch, due to the occurrence of a heavy peak, the process pressure remaining constant, the power output obtainable from 10,000 lb. of process steam per hour would fall from 256 kW. to about 210 kW.—a drop of 18 per cent.

Alternatively, if the power generating plant were capable of sustaining the electrical demand with the drop in boiler pressure indicated, the steam consumption would rise from 10,000 to about 12,000 lb. per hour, which would still further increase the peak and still further drop the boiler pressure. Under these conditions, more steam might be passed out to the process main from the engine or turbine than could be used and the surplus would be blown to the atmosphere through the relief valves.

In condensing engines and turbines the effect of steam pressure variation is less than with back pressure power plant, but even so, the effect is important. Where the steam demand fluctuates widely, this affects the temperature of the steam leaving the superheater and, usually, steam pressure and temperature keep step, rising and falling together. The stop valve pressure and temperature may well drop to such an extent as will, by increase in steam consumption, overload the condenser and impair the vacuum again aggravating the condition by calling for still more steam.

IRONING OUT THE PEAKS

By careful examination of the behaviour of the steam-using plant, simply by means of a watch and a pressure gauge, it will almost always be possible to find the offending engines or vessels. A recording steam meter, of course, shows up the offender at once.

One of the simplest ways of evening out the demand is to stagger the operation of vessels that cause peaks.

Where a number of consumers, such as a group of dye-vats, are normally brought into operation simultaneously, their operation can be staggered to reduce the peaks. Fig. 214 shows roughly what may be done in this way. Three dye-vats each have a steam cycle as shown at (A), boiling up occupying one hour, followed by the dyeing process itself, which takes two hours. If these are started up together the total steam demand would be as shown by the full line at (B), with an average demand of 3,000 lb. per hour and a peak of 6,000 lb. per hour, lasting sixty minutes. If the boiling up periods are completely staggered as shown at (E), the average demand would be reduced to 1,500 lb. per hour and the peak to 2,500 lb. per hour lasting approximately 180 minutes. Alternative schedules of operation are shown at (C) and (D). Fig. 214 is by no means complete, as it assumes that the peaks are adequately met at the rate indicated and that, therefore, the time taken to complete the process in each vat is constant and unaffected by the peaks. In fact, the building up of a heavy peak such as at (B) might drop the boiler pressure from 150–75 lb. per square inch, which would considerably increase the time taken to complete the cycle in each vat as already discussed with reference to Fig. 213A

In carrying out an investigation on the possibilities of staggering, it is

especially important to observe what occurs at the commencement of a shift. The whole day's work in a factory is often hampered by what has been allowed to happen in the first hour.

In many industries the heating of process water is responsible for the development of major steam demands. Where hot process water is required, involving the use of live steam, the water should be heated in bulk during meal times and off-peak periods and accumulated in storage tanks from which it can be drawn as required. This arrangement enables the boilers to be relieved of load throughout the working day and is especially useful because, prior to the commencement of a shift, the boilers can be brought up to a considerable head of steam which, until it is required elsewhere, can be used for the preparation of hot water. In this way the boilers can be given a flying start.

The water volume in a boiler is very important in relation to the effect of the peaks. The greater the total water content of a battery of boilers, the less the steam pressure will vary with a fluctuating load. If there is spare boiler equip-

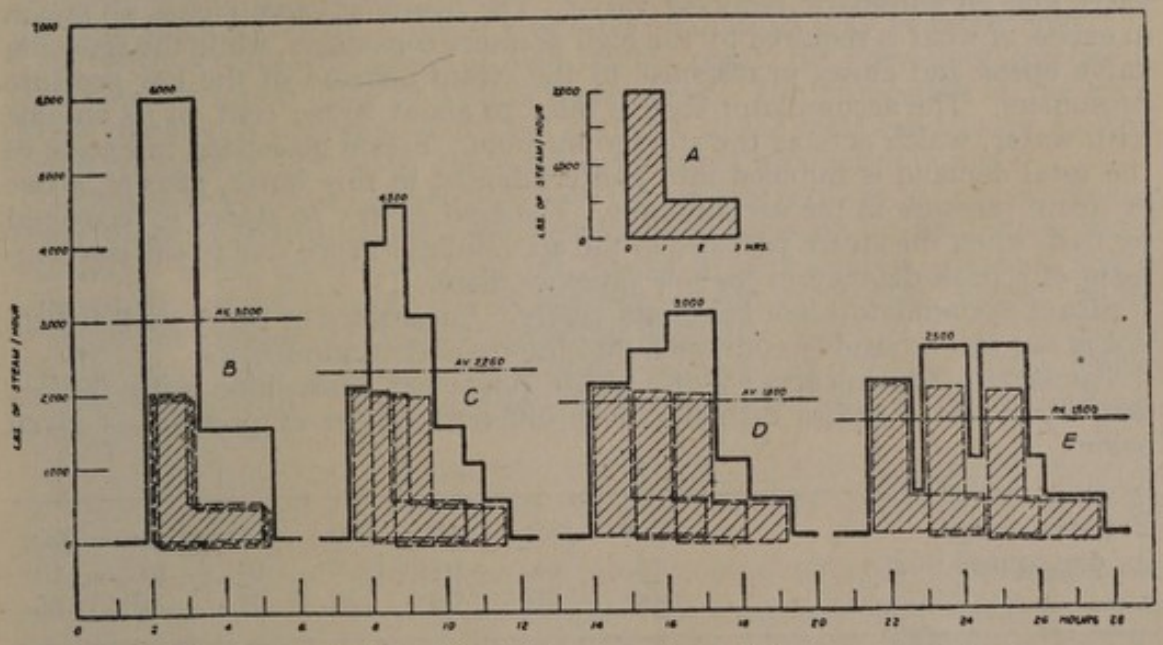

FIG. 214. Effect of staggering operations of steam consumers.

ment, especially of the Lancashire type, it may pay to bring into operation an additional boiler and to run each boiler in the battery at a lower rating. This may increase boiler operating efficiency ; it will certainly improve the efficiency of the process plant.

Safety valves should be properly adjusted to blow off at the full pressure for which the boilers are insured. In many boiler plants the safety valve on one of the boilers lifts ahead of the others as a result of which a considerable amount of peak load carrying capacity is lost to the whole battery. A gain in boiler pressure of 10 lb. per square inch is worth having. In a Lancashire boiler it would carry a 30 per cent. overload for about five minutes, which might well prevent a major pressure drop.

It is often the low pressure consumers which cause the peaks. If this is so it may be possible, by a simple rearrangement of the steam mains, to isolate a boiler or group of boilers to look after the low pressure consumers and to act as a steam accumulator working over a considerable range of pressure. A single Lancashire boiler 8 feet 6 inches diameter by 30 feet long, used in this way, would store and regenerate 1,700 lb. of steam over a pressure range from 120–80 lb. per square inch, corresponding to a 50 per cent. overload for about

twenty minutes. With a pressure range from 120–60 lb. per square inch an overload of 50 per cent. could be carried for about thirty-two minutes. Such an arrangement would help to absorb the peaks and would avoid interference between the high pressure and low pressure consumers.

IX. ACCUMULATORS

(1) STEAM ACCUMULATOR

The adoption of a steam accumulator is only applicable where a relatively large percentage of the total steam demand is met at a pressure substantially below the pressure in the main, from which steam is at times surplus and, therefore, available for storage.

The difference in pressure represents the range of pressure available for charging and discharging the accumulator. The high pressure and low pressure steam ranges are interconnected by a by-pass fitted with an automatic overflow valve and an automatic reducing valve. The overflow valve passes all steam in excess of what is required by the high pressure consumers, while the reducing valve opens and closes in response to the steam demand of the low pressure consumers. The accumulator shell is filled to about 90 per cent. of its volume with water, which acts as the storage medium. Steam generated in excess of the total demand is bubbled into and condensed in this water, causing a rise of steam pressure in the accumulator. The heat energy so stored is recovered by flash when the steam pressure in the accumulator drops due to the development of a peak demand in the low pressure main.

Steam accumulators are large and costly. Lancashire or other shell boilers not in use can be readily converted into improvised accumulators.

The steam storage capacity of a single converted Lancashire boiler 30 feet long by 8 feet 6 inches diameter over different ranges of pressure is given below.

Pressure range, lb. per square inch.	150 to 100	150 to 75	150 to 50	150 to 25	100 to 75	100 to 50	100 to 25
Steam storage capacity, lb.	2,700	4,350	6,300	9,000	1,725	3,830	6,600

The maximum permissible discharge of such an accumulator, to operate without priming, would be at the rate of approximately 25,000 lb. of steam per hour. Even one Lancashire boiler, so converted, would, therefore, be extremely useful as a means of meeting small peaks which cannot be levelled out in any other way.

(2) FEED WATER ACCUMULATOR

With this system thermal storage is obtained by condensing all surplus steam generated for the purpose of building up a reserve of boiler feed water at or near saturation temperature. Surplus steam and cold feed water are automatically and simultaneously introduced into a heat exchanger by means of suitable regulators, the steam being condensed in the heat exchanger and the resultant condensate passed into storage. The boilers are continuously fed with water drawn from the accumulator. During periods of low steam demand the accumulator is charged by rise in water level, while during peak load periods the quantity of steam and condensing water entering the heat exchanger is automatically reduced and the water level in the accumulator falls. This

process of heat storage is conducted at constant pressure, the accumulator charging and discharging by volumetric change.

With this particular system, heat can only be stored under the most favourable conditions over the range from normal feed temperature to saturation temperature and, therefore, the capacity of the system is limited.

(3) HOT WATER ACCUMULATOR

This simply consists, as already described, of using surplus steam to heat hot water, contained in a large well-lagged storage tank for use either as process or space heating water, during periods when the demand for steam is heavy.

X. DISTILLATION

This has been discussed in Chapter VII, but the details of the use of direct steam were only briefly treated.

The following examples indicate the conditions and the lines upon which the economics of the use of direct steam for distillation may be considered.

(*a*) With certain substances of high boiling point, the prevailing practice is sometimes to steam distil at ordinary pressures. The proportion of the compound distilled is sometimes only 1–2 per cent. of the total condensate, i.e. the steam required is very many times the quantity of product obtained.

Although the cost of equipment is low, the process is likely to be uneconomic, being wasteful of steam, and could profitably be replaced by distillation under vacuum. Thus, using a coal tar oil residue at atmospheric pressure and a temperature of 287° F. (142° C.) at the base of the distillation column, 30 lb. of steam are needed per gallon of naphtha distilled ; whereas if only 12 inches vacuum be employed, the steam required is cut down to 8 lb. per gallon. This is an example of the type of process in which direct steam is advantageously replaced by vacuum.

(*b*) The boiling point of glycerol is 554° F. (290° C.) at atmospheric pressure and 355° F. (179·5° C.) at 29½ inches vacuum. Glycerol is miscible with water and condensation must be avoided.

Owing to the great difficulty in continuously maintaining so high a degree of vacuum in large scale apparatus, superheated steam may legitimately be used. An economic balance is struck at about 28 inches vacuum, which is a vacuum feasible to maintain in such units.

(*c*) Experiments in deodorising oils have indicated that the steam requirements are directly proportional to the absolute pressure, i.e. the steam required at 29½ inches vacuum (½ inch pressure) is about double that required at 29¾ inches vacuum (¼ inch pressure). Also experiments conducted in the range of 300°–500° F. (150°–260° C.) indicate that the amount of steam required for deodorisation is in inverse proportion to the vapour pressure of the volatile components of the oil to be deodorised. Clearly, conditions should be used under which the vapour pressure of these bodies will be as high as possible. Since the vapour pressure increases with temperature, this operation should be carried out at the highest possible temperature.

Thus with a batch of 20,000 lb. of oil indirectly heated to maintain the required distillation temperature and with a direct steam input of 30 lb. per hour, if distillation be carried out under high vacuum the time required for deodorisation is sixteen hours at 350° F. (177° C.) at which the vapour pressure of oleic acid is 1·4 millimetres, whereas the time—and the proportional quantity of steam required—is reduced to 2¾ hours if the temperature be 427° F. (219° C.) when the oleic acid vapour pressure is 8·3 millimetres.

Such a batch of oil would require about 1,000,000 B.Th.U. to heat it from 350° F. (177° C.) to 427° F. (232° C.). The open steam used would be 300 lb.

per hour. If this steam were superheated from saturation at 350° F. (177° C.) to a temperature of 427° F. (232° C.) the thermal gain by superheating would be only about 15,000 B.Th.U. If the vacuum used be high and the steam requirements be low, there is, therefore, very little benefit to be obtained by superheating open steam.

XI. REDUCING THE WORK TO BE DONE BY STEAM

In the economical utilisation of steam the first essential is to see that the steam is given the least possible amount of work to do.

(*a*) EXCESS TEMPERATURE IN SPACE HEATING

This is possibly one of the most fruitful sources from which economy can be obtained. With an ordinary heating system a 3° F. excess temperature will result, in a British winter, in an extra fuel consumption of 10–15 per cent.

Unit heaters are often switched off at the fan with the steam still on the unit. This may result in an unnecessary waste of about 7 per cent.

Open windows and open doors used to cool a room or shop that has been overheated are direct heat wastes. This subject is discussed in Chapter XXIV.

(*b*) INADEQUATE MECHANICAL DRYING

In many processes excess moisture is first extracted mechanically before the drying process which uses a hot surface or heated air. Mechanical drying is as a rule much cheaper than thermal drying. The mechanical process can be either a squeezing or mangling process or centrifugal separation in a hydro-extractor. If the water is inadequately removed mechanically, not only is more steam heat needed in the subsequent process but the plant may be unable to remove the excess moisture and the material may have to be passed through the hot drier twice.

(*c*) ADDED WATER

In all wet processes producing a dry product the whole of the added water must be removed. Every effort should, therefore, be used to limit the addition of water at all points in the process.

(*d*) REPROCESSED MATERIAL

Some products are completely recoverable if spoiled in the process, e.g. sugar or paper. The knowledge that this is so, often makes for carelessness. Sugar spilt on the floor, for example, must be redissolved for reprocessing in 30 per cent. of water, all of which must be evaporated by steam.

XII. REDUCING OBVIOUS LOSSES

(*a*) LEAKS

The wastefulness of leaks from flanges, glands, valves, etc. would seem obvious, but the great amount of loss that can occur from an accumulation of small leaks is not always appreciated. As an indication of the heavy losses that can occur from small leaks, a hole $\frac{1}{32}$ inch in diameter in a steam pipe at 100 lb. per square inch, will waste steam equivalent to over 2 tons of coal a year.

(*b*) INADEQUATE LAGGING

Two feet of unlagged steam pipe in a 6-inch main at 100 lb. per square inch will waste about 1 ton of coal per year. An unlagged flange is approximately equivalent to 2 feet of plain pipe. There is a very natural objection to lagging

flanges due to possible corrosion of bolts if an unseen leak occurs. Box lagged flanges with a small copper tube pushed through the bottom will give warning of a leak and prevent damage. It has often been found that flanges, which give considerable trouble with leaks when bare, have remained tight after lagging due to the elimination of uneven temperature stresses.

(c) CRACKED VALVES FOR CONDENSATE DRAINING

This has already been discussed.

(d) BLOW-THROUGH SYSTEM

Alternatives to these have been suggested in the shape of steam circulation or high pressure hot water.

(e) BY-PASSED TRAPS

This common habit can be avoided by putting the traps and the trapping system in order.

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CHAPTER XXIV

CENTRAL HEATING

Space heating—Methods employed for central heating—Types of boilers used—Boiler firing—The correct management of central heating and hot water supply.

SPACE HEATING

THE problem of space heating is closely bound up with the laws of heat transmission explained in Chapter VIII. Low temperature sources of heat, such as hot water or steam radiators, generally emit a considerable amount of heat by convection. They also emit heat by radiation, the proportion depending on the emissivity, form and orientation of the surface.

Heating by convection warms the air of the room which then transmits heat to solid objects. Radiation, on the other hand, passes through the air almost unabsorbed and warms the solid objects, the air being then heated by contact with these.

High temperature sources of heat, such as an open coal, coke or gas fire have primarily a radiation effect, and the air is warmed mainly by contact with solid objects. Some appliances are designed to utilise the waste heat passing up the chimney to heat air which is then circulated into the room ; these are known as " convector " fires.

Heat is conducted through the structure and escapes from the outer surfaces by radiation and convection as has been explained elsewhere. Insulation of the structure can materially reduce heat losses ; in particular, the insulation of the roofs of buildings reduces the heat loss in winter and serves also to keep the building cooler in summer.

The amount of heat required to warm a room depends also on the rate of ventilation, which is usually expressed in terms of air-changes per hour. If, for example, a room having a total volume of 10,000 cubic feet has three air-changes per hour, this means that 30,000 cubic feet of air passes through the room in the course of an hour. A certain minimum number of air-changes, depending on the size of the room, the number of persons in it, and the type of work being performed must be allowed for health reasons, but should not be materially exceeded, since anything more than the minimum for health removes heat unnecessarily from the room or building, when the air which circulates through the room escapes into the atmosphere outside.

When designing heating installations the heat losses, air-changes and allowances for exposure, height and intermittency of heating should be based in the first place on the recommended standards set forth by the Institution of Heating and Ventilating Engineers in " Computation of Heat Requirements for Buildings." As a war-time measure for application to temporary structures, 85 per cent. of the total heat requirement so calculated should be provided, except in the following cases :—

(1) Factories where a high degree of manual dexterity is required (more heat is needed to avoid accidents and reduction of output) (see Industrial Health in War Emergency Report No. 1, 1940, H.M.S.O.).
(2) For sedentary work and in factories where female labour is employed, where 90 per cent. of normal requirements should be provided.
(3) Buildings placed at altitudes where the outdoor temperature may be considerably below 32° F. for several days at a time must receive special consideration.
(4) Buildings for living accommodation (i.e. hostels, welfare centres, etc.) where 100 per cent. of normal requirements should be provided.

These figures may require modification in some circumstances observing that :—

(*a*) Each extra degree Fahrenheit of average temperature maintained during the winter represents about 5 per cent. of the fuel consumption ; temperatures should therefore be kept as low as possible. On the other hand, complaints of coldness may lead to " forcing " of the boilers, with a consequent waste of fuel.

(*b*) Conditions in some factories are governed by the Factories Act, 1937, which provides *inter alia* that in workrooms in which a substantial portion of the work is done sitting and does not involve serious physical effort a temperature of at least 60° F. is required after the first hour.

In the following pages a general description of the types of system, boilers and the methods of firing is given, followed by a discussion on fuel economy with regard to central heating, central hot water services and combined heating and ventilating plants.

For space heating in general, heat may be produced in several ways :—

(1) By combustion within the room :
(*a*) in a grate or stove attached to a chimney ;
(*b*) in a flueless heater, such as an oil stove or a gas heater.

(2) By electric radiator.
This functions both by radiation and convection and does not require a chimney, though some form of ventilation is desirable.

(3) The fuel may be burnt outside the room, the heat being transferred to some suitable fluid, usually water, steam or air, which transmits it to the room. This is known as central heating, since the heat is generated at a central point which serves the whole system. It is this type of heating which will here be discussed. Central heating systems should embody thermostatic control (Chapter XXVI).

TYPES OF CENTRAL HEATING SYSTEMS

(*a*) LOW PRESSURE HOT WATER HEATING WITH RADIATORS AND PIPES

The most widely used method of central heating in this country is by means of low pressure hot water with cast iron sectional boiler and cast iron radiators. Water is heated in the boiler up to a maximum temperature of about 180° F. or less, according to the external weather conditions. The water from the boiler circulates through a system of pipes to the various radiators and returns from the radiators through the return pipes back to the boiler, the temperature of the water returning to the boiler being 20°–40° F. lower than the temperature leaving the boiler. This water again passes through the boiler, is heated to the flow temperature and the circulation continues in this way. The same water is circulated continually and after the initial filling only a very little make-up water is occasionally required.

In small installations, where the layout is favourable, the system may be designed for a natural gravity circulation, utilising the difference in weight between the hot flow water and the cooler return water to promote circulation round the system. This is only possible where these circuits are not very long and the heating surface is at a reasonable height above the boiler.

In larger installations or where the layout does not allow circulation by gravity, the water is circulated mechanically by an electrically driven centrifugal pump (or by a turbine driven pump if steam is available) as in Fig. 215. It is usual to install two pumps, one acting as a standby in case of breakdown. The pumps are fitted on a by-pass and provided with isolating valves ; a non-

return valve is fitted in the main pipe between the suction and delivery connections to the pumps. The pumps are usually on the return main, delivering into the boiler. While the pump is running the non-return valve is held in the closed position by the pressure differential created by the pump, but if the pump should stop the valve opens and a gravity circulation will continue as far as the design of the system makes this possible. This arrangement guards against the possibility of a complete stoppage of circulation in the event of pump failure with consequent overheating and possible damage to boilers.

The size of the pumps installed varies according to the size and layout of the system and they may work against heads varying between 3 and 20 or 25 feet. The various pipe circuits are arranged and the sizes calculated so that the total resistance set up by the flow of water round each complete circuit is approximately the same and equal to the pump head.

The quantity of water circulated by the pump is constant, and variations in heat requirements in the building are met by varying the temperature at

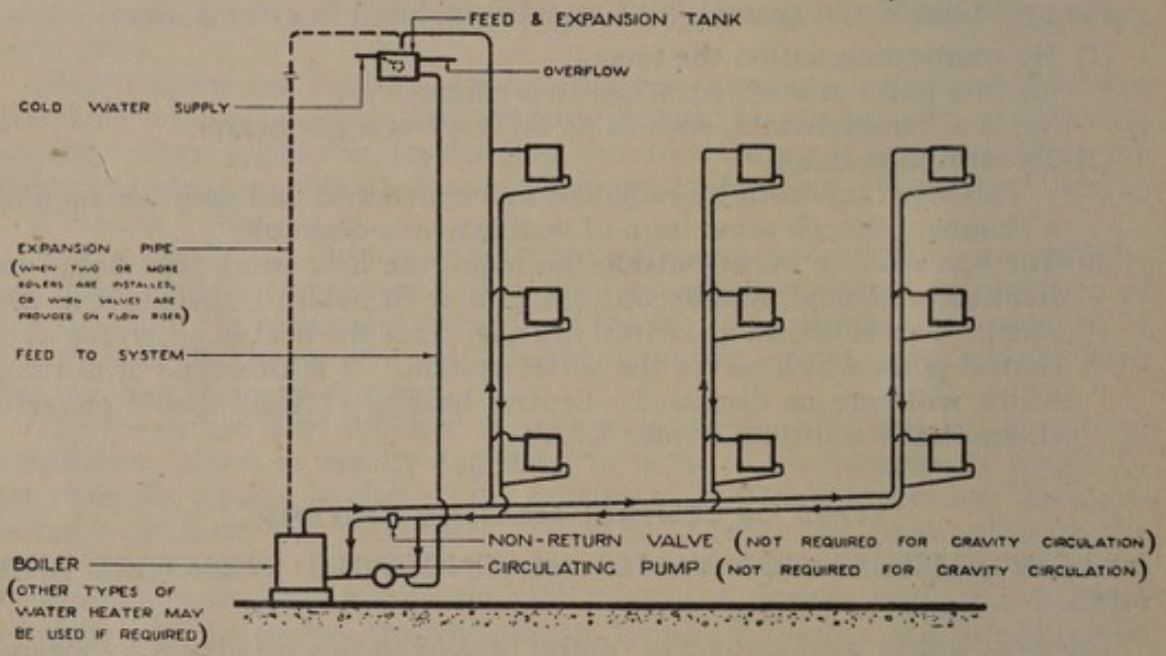


FIG. 215. Low pressure hot water heating with radiators and pipes.

which the water leaves the boilers between a maximum of about 180° F. in coldest weather down to about 100° F. in mild weather.

This type of system is open to the atmosphere so that the pressure on the boiler depends solely upon the static head of water in the system. A feed and expansion tank is installed above the highest point in the circulation and connected to a point near the boiler by means of a feed pipe. The water level in this tank is maintained by means of a ball valve. All pipes are arranged with a slight rise so that any air in the system will collect at definite high points. At these points open air pipes are provided, these being carried to a height well above the water level in the tank or if this cannot be done automatic air release valves are used instead.

In most installations radiators are of the cast iron sectional type. The heat from the water in the radiators is conducted through the metal and raises the temperature of the air immediately in contact with the surface. This warmed air rises, cooler surrounding air takes its place and a continuous circulation of air is thus induced over the radiator. The hot surface of the radiator also emits some heat to the room directly by radiation. Small rooms may have only one radiator, but larger rooms require several according to their size.

As heat is lost from the rooms mainly through the external walls, radiators are placed against these as far as possible. Windows cause down-draughts of cold air, and it is usual to place radiators beneath them to counteract this effect. In a similar way down-draughts from skylights or windows at high level may be guarded against by running pipes beneath them. There is a slight temperature gradient between floor and ceiling in a room heated in this manner, therefore where the height exceeds about 12 feet additional heating surface is installed to ensure that the required temperature is obtained in the occupied stratum.

The amount of heating surface in the different rooms of a building must be carefully proportioned according to the size of rooms and the area of walls, windows and other surfaces exposed to the outside air. Different temperatures may also be required in different rooms. The rate of ventilation must also be taken into account and allowance made for warming all incoming air to the temperature of the room. In offices and similar rooms the allowance for air change is one-and-a-half or two per hour, but in buildings such as schools or hospitals higher rates of air-change are necessary. In industrial buildings the rate of ventilation will depend upon the number of occupants and the kind of processes being carried out.

Boilers, pipes and any other surfaces which are not used for direct heating purposes should be insulated with non-conductive material to prevent unnecessary waste of heat.

(*b*) LOW PRESSURE HOT WATER HEATING WITH PANELS

Heating by means of radiant panels is often adopted in certain types of buildings such as offices, schools, hospitals and public buildings. The heat is imparted to the rooms by large plane surfaces warmed to a temperature above that of the air in the room. These panels may be in the ceiling, walls or floor, but ceiling panels are generally used (Fig. 216).

One type of panel consists of a flat steel or cast iron plate backed with small waterways in the form of ribs through which the hot water circulates in exactly the same way as for an ordinary radiator. These panels are superimposed or

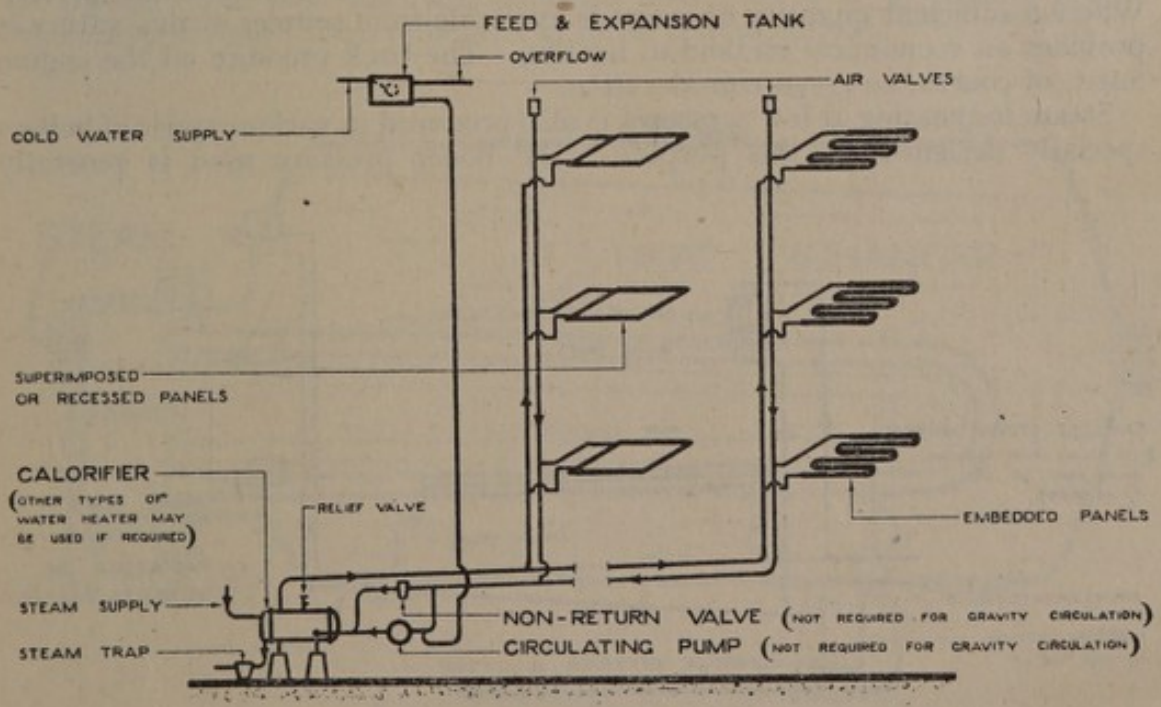


FIG. 216. Low pressure hot water heating with panels.

recessed into the structure of the building in various ways and the backs are insulated with non-conducting material. The pipes connecting to them are usually concealed so that the heating device is hardly perceptible. In this type the water may be circulated at the same temperature as it would be in an ordinary radiator heating system.

Another method is to embed small pipes, $\frac{1}{2}$–$\frac{3}{4}$-inch bore, in the structure itself. Pipes made up into grids or coils are embedded in the concrete floor or ceiling and the warmed area of the floor or ceiling radiates heat to the room. The flow and return connections to the panel coils are also embedded in the structure so that the heating panel is unobtrusive. Pipes solidly embedded in concrete in this way are heated only to a moderate temperature and the temperature of the circulating water usually does not exceed 120° F., and may be varied as follows, according to weather conditions :—

Outside shade temperature	55	50	45	40	35	30° F.
Flow temperature	85	90	97	105	112	120° F.

A suitable method of plastering is adopted to guard against the plaster cracking.

There are several methods of floor heating by which the pipes are laid in a hollow space beneath the normal floor finish so that the whole or part of the area of the floor is warmed, but the permissible surface temperature must be kept low otherwise uncomfortable effects may be produced.

As heating by panels, particularly ceiling panels, depends upon radiation rather than convection, that is to say it heats the objects in a room directly without necessarily also warming the air, the rate of air interchange by ventilation does not have such an effect on comfortable conditions in the room as it would have by ordinary convection heating ; panel heating in a building is generally found to result in a lower fuel consumption than convection heating.

(c) LOW PRESSURE STEAM HEATING

Steam at low pressure is often used for heating, particularly in industrial premises where a plentiful supply of steam is available (Fig. 217). Exhaust steam from turbines or engines may be utilised where circumstances permit. Where a sufficient quantity of steam is available from sources of this nature it provides an economical method of heating. The back pressure on the engine must, of course, be given consideration.

Steam for heating at low pressures is also produced in various types of boilers specially designed for this purpose. The boiler pressure used is generally

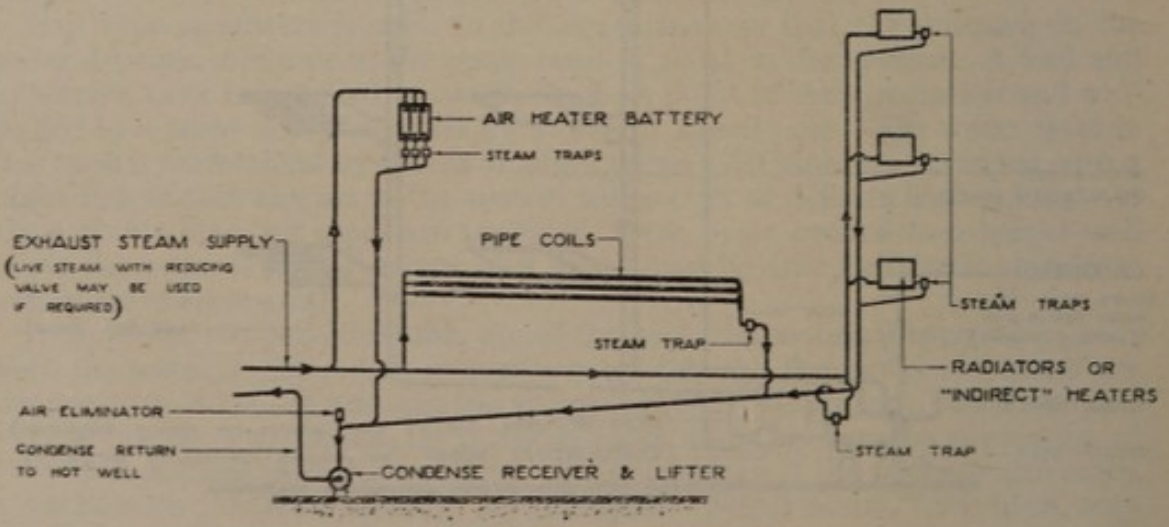


FIG. 217. Low pressure steam heating.

between 3 and 10 lb. per square inch, and cast iron radiators of the type used for hot water heating may be used with steam at this pressure. The steam is conveyed to the radiators by one pipe, and is condensed in the radiators.

It is then discharged by a steam trap into a condensate pipe through which it returns to the boiler. Steam traps are fitted to the steam main at intervals to drain away any condensate which may collect, thus keeping the steam as dry as possible. The return lines are arranged to fall back to the boiler house so that the condensate will return by gravity flow. If exhaust steam is being used the condensate will be returned to the hot-well (providing it is in a suitable condition for re-use).

When steam is generated in low pressure boilers specially designed for heating, the condensate is very often returned to the boilers by gravity without the use of a feed pump. The same water is repeatedly evaporated and condensed so that there is very little make-up required. It is usual to fit a boiler feeder consisting of a float valve in a suitable chamber attached to the side of the boiler, arranged so that when the water level falls below the required point, make-up is automatically admitted. The water supply to this feeder is taken from a small tank at a height such that the static head is sufficient to overcome the boiler pressure.

This system is often used in workshops, etc., for heating by means of pipe coils. Radiators and pipes utilising low pressure steam have surface temperatures in the region of 230° F.

A recent development has been the introduction of "indirect" heaters suitable for this type of system. These comprise a special form of cast iron radiator or a battery of gilled tubes encased in a metal cabinet having openings at the bottom and top, so that when steam passes through the heater a continuous convection current of air is induced through the bottom opening, over the heater, and out at the top.

The output from radiators, etc., will, of course, depend upon the surface temperature which will in turn depend upon the steam pressure. The pressure in the radiator will be equal to the boiler pressure less the drop in pressure in the piping. The pipe sizes are therefore proportioned so as to give a predetermined pressure at each radiator or other heating device.

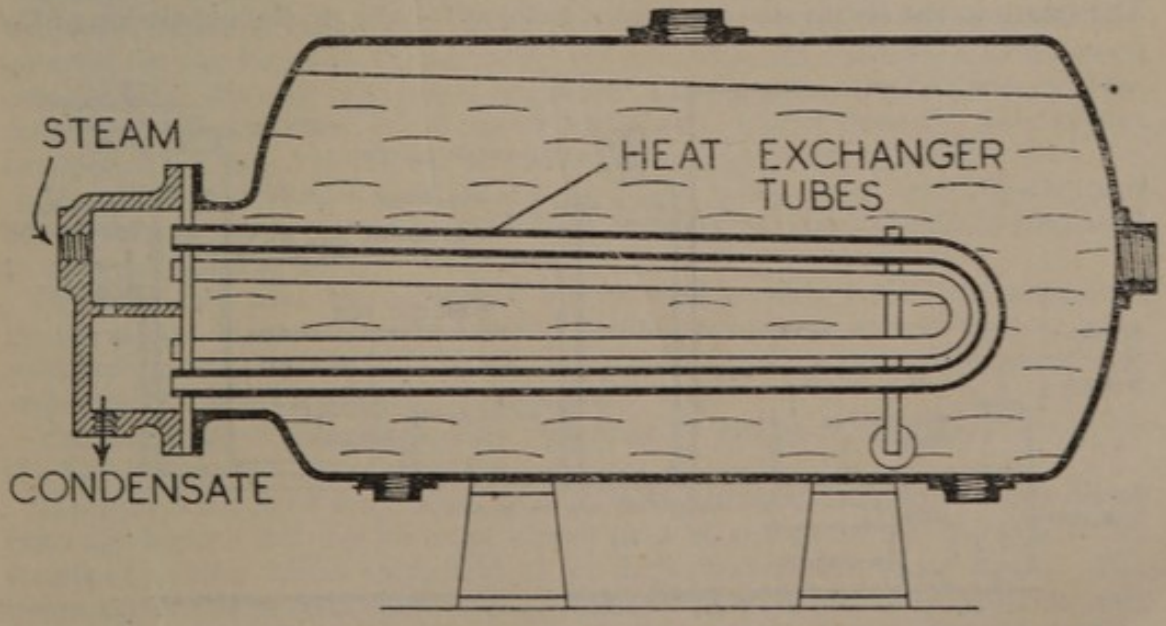


FIG. 218. Hot water supply storage calorifier.

Where steam is available for heating and it is desired to heat by low pressure hot water, this is done by means of calorifiers which consist of a steel cylinder containing a battery of steam pipes for heating the water which is circulated through a heating system in the usual way (Fig. 218).

(d) VACUUM STEAM AND VAPOUR HEATING SYSTEMS

The vacuum system consists in essence of a boiler, pipework and radiators arranged in a similar manner to an ordinary low pressure steam system, the main difference being the addition of a vacuum pump and receiver on the main return line close to the boilers.

The duty of the pump is to maintain a partial vacuum in the return line and deliver the condensate back to boiler. This assists in preventing air accumulating in the radiators and ensures more even heating. The rate of flow of steam through the radiators may also be varied by changing the amount of vacuum. The return of condensate from radiators is more positive and under control, and if necessary it may be lifted so that difficulties due to unfavourable levels may be overcome.

A further modification is the " sub-atmospheric " system in which the boiler pressure is allowed to vary from a few pounds per square inch above atmosphere down to a vacuum of about 25 inches of mercury, so that steam is generated at temperatures varying from about 220° F. down to about 130° F. A vacuum pump is provided to deal with the condensate. The steam pressure at radiator inlet and vacuum at outlet, produce a differential pressure causing the steam to flow through the radiator. The rate of flow may be controlled by varying the differential pressure according to weather conditions. Air is eliminated from the system by the vacuum pump and a positive return of the condensate is provided. It is also possible to control individual radiators by means of suitable regulating valves.

(e) HIGH PRESSURE HOT WATER HEATING

A recent development in hot water heating, chiefly for industrial applications, is the use of a completely closed system of pipework connected to a steam boiler below the water line, so that the water in the boiler may be circulated through the pipe system (Fig. 219). Steam is raised in the usual way, the only difference being that the water line is maintained at a rather higher level than normally. The steam in the steam space acts as a cushion to take up the expansion of the

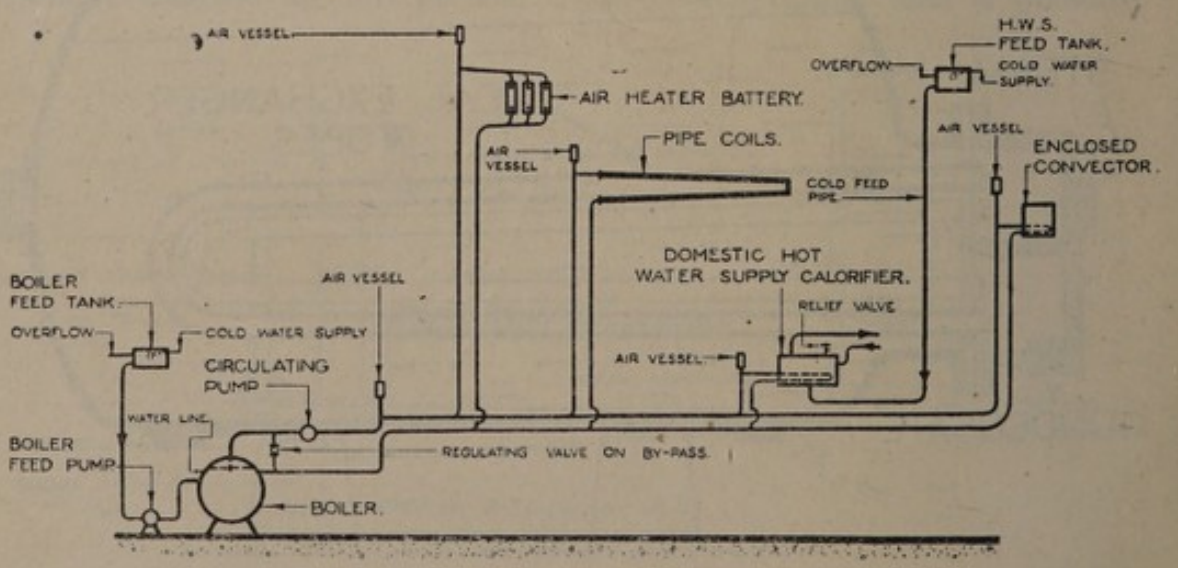


FIG. 219. High pressure hot water heating.

water in the system as its temperature rises, and the boiler pressure is imposed upon the whole of the system. Where more than one boiler is installed these are connected together above and below the water line by means of balance pipes to ensure a constant water level in all boilers. Working pressures up to 250 lb. per square inch are used, but an average figure is 150 lb. per square inch.

The water from the boilers is circulated through the pipe system by means of centrifugal pumps. The flow temperature of the water leaving the boilers will correspond to the steam pressure in the boilers, i.e. if the boiler gauge pressure is 150 lb. per square inch the water temperature will be 366° F. Any reduction in pressure will therefore cause some of the water to flash into steam ; to guard against this a by-pass is installed between the return main and the flow main.

The pumps are placed on the flow main immediately after the by-pass connection, so that cooler return water can be mixed with the boiler water when passing through the pump. The temperature of the water flowing to the system may therefore be varied by controlling the proportion of water at the two different temperatures.

Circulating pumps are installed on the flow so that the additional pressure imposed on the system helps to reduce the possibility of flashing. The connection between boilers and pumps must be as short as possible.

An installation of this type can provide heat for process work also. Whilst the temperature required for this purpose will be constant throughout the year, that required for heating will depend upon the outside weather conditions, and will not be required at all during the summer months. Separate heating and process mains are therefore necessary, a common practice being to install separate flow mains with a common return.

The boilers are usually worked at full pressure at all times, so as to maintain a constant pressure on the system ; variations in heat requirements are met by the above mixing arrangements and variations in load by varying the number of boilers in operation.

Air vessels are provided at suitable points for collecting the air in the system from which it is released by means of air pipes and valves. It is most important that all air be excluded from the system before it is put into operation otherwise serious waterhammer may be produced.

The pipe sizes are calculated in a similar manner to those for low pressure hot water heating, allowance being made for the effect of temperature and pressure on the viscosity of the water which affects the resistance to flow and consequently the size and power of the circulating pumps. The temperature drop round the system varies according to the initial pressure, but figures between 50° F. and 100° F. are common, an average being about 70° F.

From the point of view of efficient operation it is important that all pipes and other surfaces which are not used for heat emitting purposes should be insulated to prevent waste of heat.

Ordinary cast iron radiators must not be used as they would not withstand the pressure. Heat is imparted to the building by means of unit heaters, pipe coils and in small spaces such as offices, by enclosed convectors containing pipe coils capable of withstanding the pressure.

Hot water for domestic purposes is heated by means of calorifiers served by the high pressure hot water mains.

If steam is required for process work at high pressures it may be taken direct from the boilers, but low pressure steam may be generated where required by means of a steam raiser connected to the high pressure hot water mains. The steam raiser comprises a mild steel cylinder fitted with the usual boiler mountings and provided with a small electrically driven centrifugal pump for feeding

purposes. This is generally under the control of a float-operated switch to maintain a constant water line. The high pressure water is circulated through a pipe coil immersed in the water, causing evaporation and the production of steam. The circulating water should be under the control of a valve operated by a pressure switch attached to the steam space of the cylinder so that constant steam pressure is maintained. The pressure which can be raised depends upon the temperature of the high pressure water ; with water at 350° F. it is possible to generate steam up to pressures of about 30 lb. per square inch.

(*f*) UNIT HEATERS

Unit heaters are extensively used for heating large areas in industrial premises and are suitable for low and high pressure hot water or steam. A unit heater consists of a battery of steel or copper tubes, usually gilled, in a metal case with an opening in the back and adjustable louvres in front. A propeller fan driven by an electric motor is mounted behind so that it blows air over the heater battery, this warmed air being deflected downward by the louvres.

The heaters are usually fixed at a height of 12–15 feet above the floor. The horizontal spacing depends upon the height and velocity of air discharged, and heat output of the units used. Their effectiveness in heating depends upon the spacing and orientation and the adjustment of the louvres. Higher units require a greater discharge velocity, but if placed too high the tendency of the warm air to rise will overcome the downward velocity before it reaches the working zone and they will not be so effective in heating. On the other hand a unit which is placed too low may produce draughts and patchiness in temperature.

(*g*) COMBINED HEATING AND VENTILATING SYSTEMS

Heating by any of the methods described may be combined in various ways with ventilation so as to ensure a positive supply of warmed fresh air.

For theatres and places where a large number of people are gathered together, a positive method of ventilation is necessary to ensure an adequate quantity of fresh air. To meet these conditions a combined system of heating and ventilation is installed. Air from the outside is drawn over a heater by means of a fan which discharges the warmed air into a duct system connecting to grilles of suitable size and location in the various spaces to be dealt with (Fig. 220).

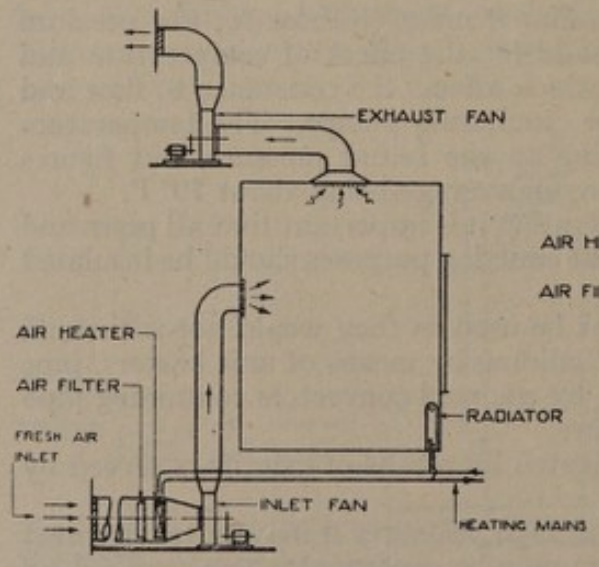


FIG. 220. Combined heating and ventilating plants—group (*a*).

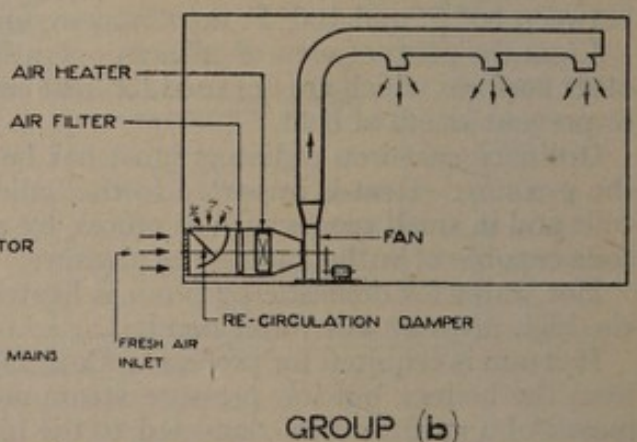


FIG. 221. Combined heating and ventilating plants—group (*b*).

The air is heated so that it enters the rooms at approximately the same temperature as the air in the rooms. Direct hot water or steam heating surface is also installed to make good the heat loss through walls, windows, etc.

An extraction fan connected to a separate duct system exhausts the air from the rooms through extract grilles so placed in relation to the fresh air grilles that an equable distribution of the air is obtained throughout the space concerned.

Arrangements may be made for part of this extracted air to be re-circulated through the fresh air ducts where it is mixed with a proportion of fresh air and returned to the rooms. This arrangement economises in fuel consumption, but depends upon the condition of the exhausted air and to some extent upon whether the air is passed over air filters or washers before being re-delivered into the rooms.

The air heaters may be served by hot water or steam produced in heating boilers of the usual type, and the boiler may also serve for direct heating by radiators or other means.

Alternatively, there may be no direct heating surface, the air then being delivered into the rooms at a higher temperature, so that after giving up heat to the cooler room surfaces its final temperature is equal to the room temperature required (Fig. 221). This method is largely used in workshops and factories.

In industrial buildings heated by unit heaters fresh air may be necessary to replace the large quantities of air exhausted by dust-collecting plant, fume exhausting systems, etc. This is usually accomplished by arranging the unit heaters to draw air from outside the building through metal ducts, re-circulating dampers being provided on these ducts so that all or part of the air passing through the heater may be taken from the building when additional fresh air is not required.

(*h*) HEATING BY ELECTRICITY

Electricity is usually applied to the heating of large buildings by the thermal storage system. Current is used during off-peak periods for warming up water in large storage cylinders which are very efficiently insulated to reduce heat loss, the heat stored in the water being sufficient to supply all heating requirements during the period when current is switched off. This water is circulated by a pump through an ordinary hot water heating system, the storage cylinders merely replacing the more usual boilers.

The temperatures to which the water may be warmed will govern the amount of useful heat which may be stored in each cubic foot of water, and the maximum temperature depends upon the static head on the storage cylinders, i.e. the higher the building being heated, the greater the amount of heat storage possible in a given volume. The maximum storage temperature at the end of the off-peak period should be about 20° F. below the boiling-point of water at the static head concerned. This temperature will usually be above the boiling-point of water at normal atmospheric pressure and much too high for circulation round the heating system.

The water in the heating system is therefore circulated by means of centrifugal pumps and high temperature storage water is added to the circulating water through a thermostatically controlled mixing valve in such quantities as are required to maintain the necessary temperature of the circulating water. This temperature will vary according to the outside weather conditions, etc., and will also depend upon the type of system, e.g. it will be lower for an embedded panel system than for a " radiator " system.

During the daytime there will be a gradual fall in the temperature of the storage water and by the time the current is again switched on the storage

temperature may have fallen very nearly to the flow temperature to the heating system.

The water may be heated by electrical immersion heaters in the cylinders, or more frequently by an electrode water heater which consists of a mild steel shell containing water in which are immersed three electrodes connected to the three phases of the supply. A small quantity of a suitable chemical is added to the water to increase its conductivity to the necessary value. The current then passes through the water between the electrodes, so heating it. The shell of the heater is connected to the neutral of the supply and is earthed. (It should be noted that a supply cannot be earthed in this way without the permission of the Electricity Commissioners.) The electrode heater is usually connected to the storage cylinders by means of flow and return pipes and the water is circulated between them by " primary " circulating pumps.

TYPES OF BOILERS

(*a*) *Cast Iron Sectional Boilers.* The most popular type of boiler for low pressure hot water heating and low pressure steam heating is the sectional boiler (Fig. 222). This is usually of cast iron, although mild steel boilers of this type are also frequently used for particular reasons.

The sectional cast iron boiler is inexpensive, independent of brick settings, and easy to install. Being in comparatively small sections, it is easy to transport, and to get into places having very limited access. A damaged section

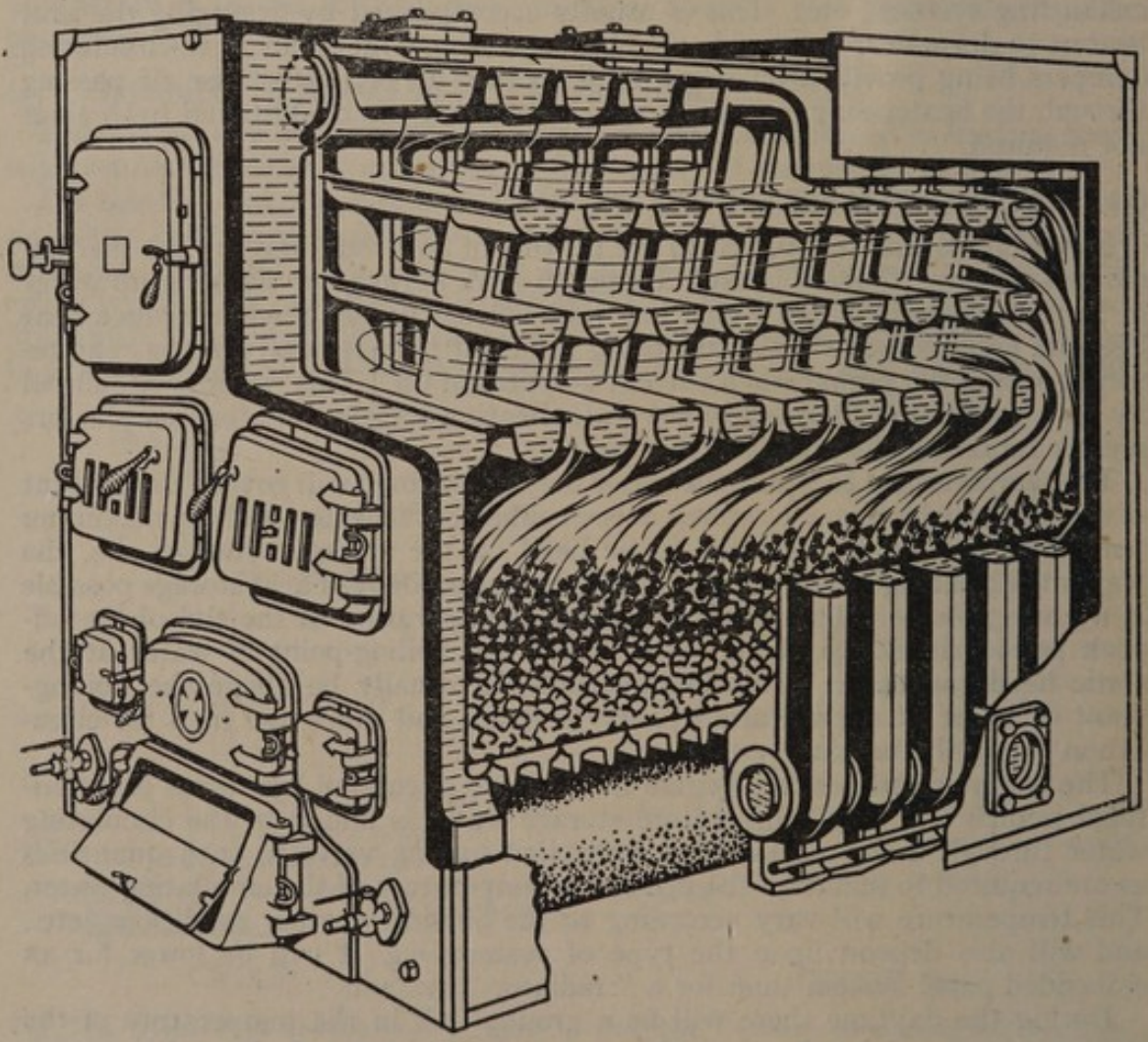


FIG. 222. Cast iron sectional hot water boiler.

may be replaced without having to renew the whole boiler. The boiler is built up of vertical sections held together by bolts, the waterways being connected by nipples. In some types the sections are mounted on a separate base, in others the sections are designed to stand direct on the floor.

If an installation is increased in size the extra load may be carried by adding more sections to the boiler, up to its maximum limit in length. The permissible length depends upon the method of firing adopted, being greater for mechanical firing than for hand firing. When fired by hand, coke or anthracite is used.

For hot water heating, the flow water leaves the top of the boiler, and the cooler return water enters at low level. Return connections at both sides are preferable to ensure good circulation of the water within the boiler. For this reason, also, it is preferable that flow and return connections be made at opposite ends of the boiler.

Smaller sizes have ordinary grille firebars which tend to become distorted and burn away in time and require periodical replacement. The larger sizes, and in some makes also the small sizes, are now provided with waterway firebars which form part of the sections. These have two advantages ; the temperature of the firebars is kept low so that they cannot burn out, and the disposition of water-cooled firebars increases the overall efficiency of the boiler.

The front section of the boiler is usually provided with three central doors ; the lower is the ashpit door, the next the clinker door for cleaning out the fire, and the top one is the fuel door through which fuel is thrown on to the fire. There are also two or more doors to give access to the boiler flues for cleaning purposes.

This type of boiler is suitable for low pressure hot water heating to heads up to 100 feet. For low pressure steam heating the maximum permissible working pressure is 15 lb. per square inch.

When used for low pressure steam heating, steam drums may be mounted above the boiler and connected to the steam and water spaces. The boiler is then completely filled with water and the working water line maintained at about the centre line of the drum, instead of a few inches above the crown of the furnace. Water and steam gauges are mounted on the drum. The steam supply is taken from the top of the drum. Make-up water and condensate are returned to the boiler in various ways according to the type of system.

These boilers are generally designed so that the gases of combustion leave the furnace at the back, then pass forward through flues to the front, returning to the back and passing out through a smoke nozzle at the back. The heating surface exposed directly to radiation from the fire is known as primary heating surface, and that in the flues, secondary heating surface. For efficient working the chimney should be of ample size and provided with access for cleaning.

The draught is controlled by dampers which are normally manually operated, although thermostatically controlled dampers are being provided to an increasing extent. The ashpit damper forms part of the ash door and controls the passage of air through the grate. A secondary air damper is provided in the firing door, usually arranged so that air passing through it is preheated by some simple device before entering the combustion chamber. The flue damper is usually of the butterfly type fitted in the smoke nozzle at the back of boiler and operated by a rod extended to the front.

(*b*) *Mild Steel Sectional Boilers.* Sectional boilers are also manufactured in mild steel, similar in design to the cast iron types, but made of mild steel plates welded together. They are more expensive than cast iron boilers, but have certain advantages. They are capable of withstanding somewhat higher pressures and may be installed for hot water heating in higher buildings, or may be used for steam heating at pressures up to 25 lb. per square inch. Also, steel is less easily damaged. The sections of steel boilers are not connected by

nipples, but usually by external flow and return headers separately connected to each section.

(*c*) *Gravity Feed Boilers.* A variation of the foregoing types is the cast iron or mild steel gravity feed boiler (Fig. 223). This type is constructed on a similar principle to the ordinary sectional patterns, except that a fuel bunker or hopper is incorporated above the grate, so arranged that as the fuel on the grate is burned fresh fuel falls on to the grate to take its place. The grate is often sloping, the fuel being fed on to the higher end, gradually falling to the lower end so that by the time the lowest point is reached it is completely burned, the ashes falling through the grate bars into the ashpit below. The aperture through which the fuel falls on to the grate is adjustable so that the thickness of the fire may be controlled according to requirements. The rate of combustion is under control, either by the use of thermostatically operated dampers or a thermostatically controlled forced draught fan which delivers air into the furnace for combustion.

Coal or coke may be used, a fairly small size being required for most types. The storage capacity of the fuel hoppers may be up to twenty-four hours supply, and if the main fuel bunkers are above the boilers so that they may be

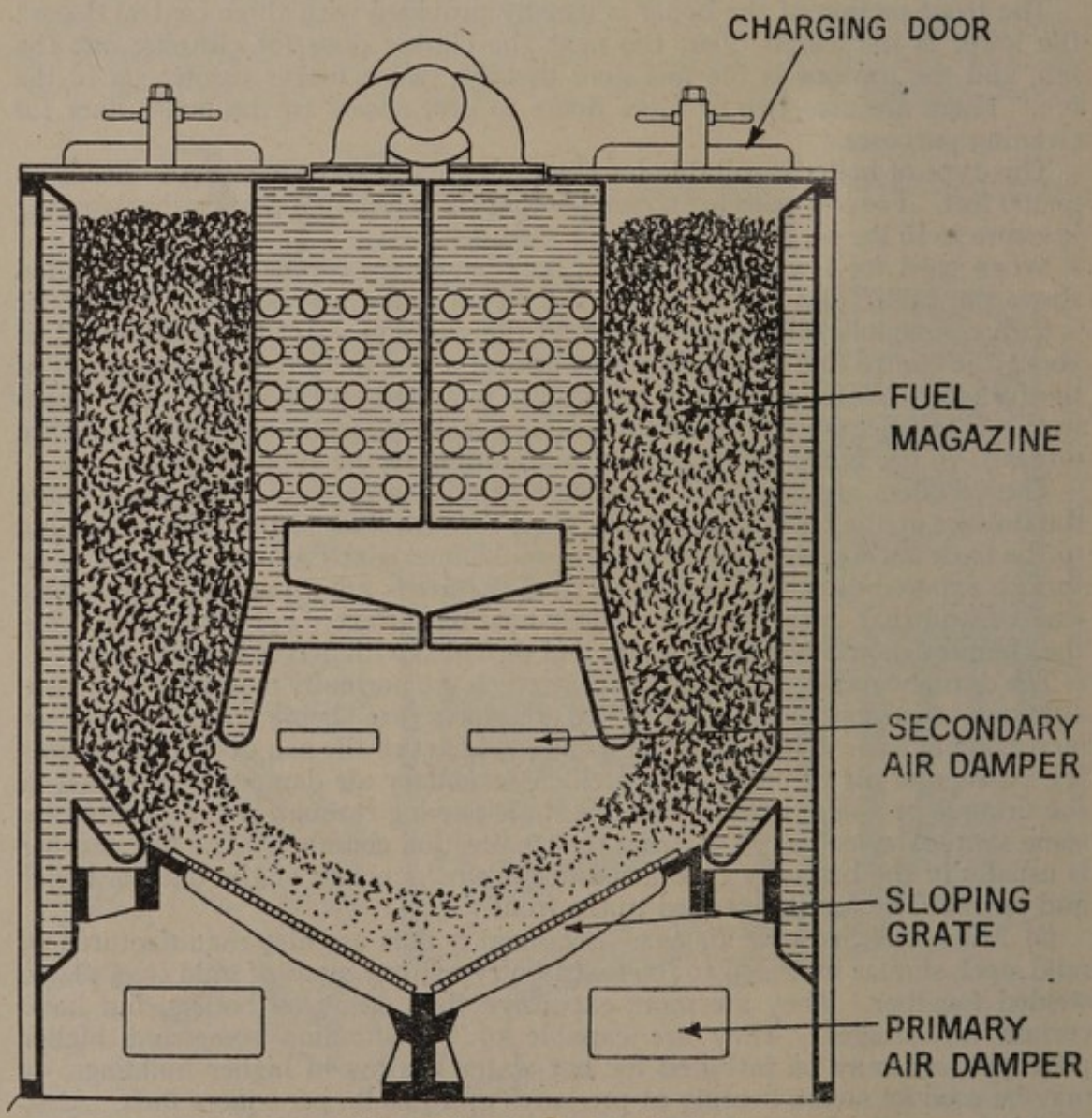


Fig. 223. Gravity feed boiler.

refilled by means of simple chutes, the handling of the fuel is practically eliminated. With automatic draught control, boilers of this type may therefore be left for fairly long periods without attention, apart from occasional clinkering and removal of ash. Air-tight lids are provided to the hoppers to prevent the possibility of the fire creeping back from the grate into the hopper. It is essential that the seatings have proper attention so that the lids may be closed tightly.

(*d*) *Independent Wrought Iron and Mild Steel Boilers.* This category includes a large variety of types all of which may be used for heating work and range from a boiler made out of mild steel plates and comprising little more than a box with double walls, to the Lancashire and Economic types. All may be used for heating by hot water or steam at high or low pressures.

FITTINGS AND MOUNTINGS FOR HOT WATER BOILERS

Boilers for low pressure hot water heating are provided with some or all of the following fittings :—

Safety valve : This is essential and must be fitted to all boilers. It may be of the deadweight or spring pattern, and should be of the totally enclosed type. Safety valves having loose weights or exposed springs should be avoided as they are liable to be tampered with. The valve consists of an accurately ground disc held on to its seat by a spiral spring or a series of weights. Adjustment is made by varying the pressure exerted by the spring or changing the number and size of the weights. The valve is enclosed in a metal body having an outlet to discharge water if the valve should lift. A pipe should be screwed into this outlet and carried clear of the boiler so that water will be discharged where it will not do any damage. The cover should be held securely in position and a padlock fitted to prevent interference by an unauthorised person. Testing levers are sometimes fitted, but are not essential. The area of the valve should be of ample size, and it should be adjusted to blow off at a pressure of 10 lb. per square inch above the normal static head on the boiler. It should be noted that if a safety valve is knocked it may discharge water for some considerable time afterwards. This should not be confused with leakage which may occur in time due to wear of the valve, when the scaling must be removed and the seating reground.

Thermometer : This may be of mercury-in-glass type, protected by a steel or brass case, or a dial pattern with pointer. It should be fitted on all boilers to indicate the temperature of the water as a guide to rate of firing and adjustment of dampers. It is fitted on the top of the boiler and it is essential that the bulb be properly immersed in the waterway. It is also an advantage to have a thermometer on each main return circuit in the boiler house as a guide to the temperature drop through the system and to facilitate the accurate regulation of the various circuits to ensure proper distribution of the water.

Emptying Cocks : These are fitted at the lowest point of the boiler and are operated by loose keys. They are used for emptying the system. Sectional boilers may need one on both sides, otherwise it may be only possible to empty one half of the boiler. It is sometimes found that persons in charge of boilers draw water off through the emptying cocks for cleaning purposes. This must never be allowed, as the resulting continual influx of fresh water to the boiler will eventually result in damage due to corrosion or by overheating caused by deposits of scale.

Altitude Gauge : A Bourdon type pressure gauge calibrated in feet head is used to indicate the head of water on the system. While not absolutely essential it is useful to indicate shortage of water in the system.

Damper Regulator : This is a useful piece of apparatus which is very often fitted. It consists of a bulb immersed in the top waterway of the boiler, the

bulb being connected to metallic bellows filled with a liquid which changes in volume with change of temperature so producing a corresponding change in the length of the bellows. The resulting movement is transmitted to a lever which is connected to the ashpit door by means of a rod or chain in such a way that as the boiler temperature falls the ashpit door is opened and as it rises the door is closed, by this means automatically maintaining a constant water temperature, the temperature being regulated by altering the length of the connecting chain or rod. The use of a damper regulator prevents overheating and assists in the economical working of the boiler.

Boilers for low pressure steam heating are provided with the following :—

Safety Valve : Enclosed spring or deadweight types are used.
Emptying or blow-down cocks.
Pressure gauge, syphon and cock.
One or two water gauges.
Set of stoking and cleaning tools.

Damper Regulators : Similar to that described for hot water boilers, but operated by steam pressure.

Automatic Boiler Feeder : A device consisting of a float-controlled valve arranged to admit water to the boiler when the water level falls below a certain point. This may only be used for boilers working at fairly low pressures, and is fed from an overhead tank, the height of which must be sufficient to produce a static head greater than the pressure in the boiler.

METHODS OF FIRING

(*a*) *Hand Firing with Coke or Coal.* Heating boilers may be hand fired with coal or coke. In the usual types of sectional boiler it is not desirable to burn soft bituminous coals, as these result in large deposits of soot in the flues. Moreover, these fuels are not so suitable for banking for long periods. Sectional boilers are designed to burn coke or anthracite, the sizes used varying between $\frac{3}{4}$ inch for the smaller types, and 2–4 inches for the larger types.

Sectional heating boilers are designed and rated on the basis of one charge of fuel lasting for six hours when being burned at such a rate that the specified output of the boiler is maintained, leaving sufficient burning fuel at the end of this period to rekindle the next charge. A boiler can, of course, be forced to some extent beyond its normal rating and this usually happens when it is not large enough for its duty, but forcing should be avoided as the more extreme working conditions will shorten the life of the boiler and reduce the combustion efficiency.

When the fuel is getting low in the combustion chamber the dampers are all opened and the fire is thoroughly cleaned out, clinker is removed and the ashes raked out of the ashpit. The boiler is then recharged with fresh fuel to about half-way up the firing door. All the doors are now closed and the ashpit and flue dampers kept open for a short time until the new fuel is alight. The dampers are then set at an intermediate position according to the conditions required.

If the temperature in the building tends to rise above the required figure the dampers should be closed. Overheating should be guarded against to economise in fuel consumption. During cold weather the boiler will be refuelled about once every six hours while the building is occupied, but during mild weather less frequently.

It may not be economical in a building occupied every day to let the fire go out each night and to relight again early the following morning, as it may use more fuel than if a slow fire is kept going during the night. In mild weather,

if the building is unoccupied during week-ends the boiler may be put out of commission between Saturday morning and Sunday evening, but in cold weather it is advisable to keep a slow fire going the whole of the time. If there is any likelihood of frost the fire must not be allowed to go out or damage by freezing may result.

An average day's firing will comprise cleaning the fire and filling up with fuel about two or more hours before the building is to be occupied, and repeating this procedure again about mid-day. During the afternoon a slower fire will generally be sufficient to maintain the building temperature and this charge of fuel should last until the evening when the fire is again cleaned out, and refilled, the ashpit door and flue dampers now being almost completely closed and the boiler left banked for the night. The secondary air damper should be left open during banked periods to admit additional air over the fire and ensure proper combustion of the gases distilled from the fuel.

The boiler flues should be cleaned out once every day.

(*b*) *Mechanical Stokers.* Mechanical strokers of the underfeed type are now

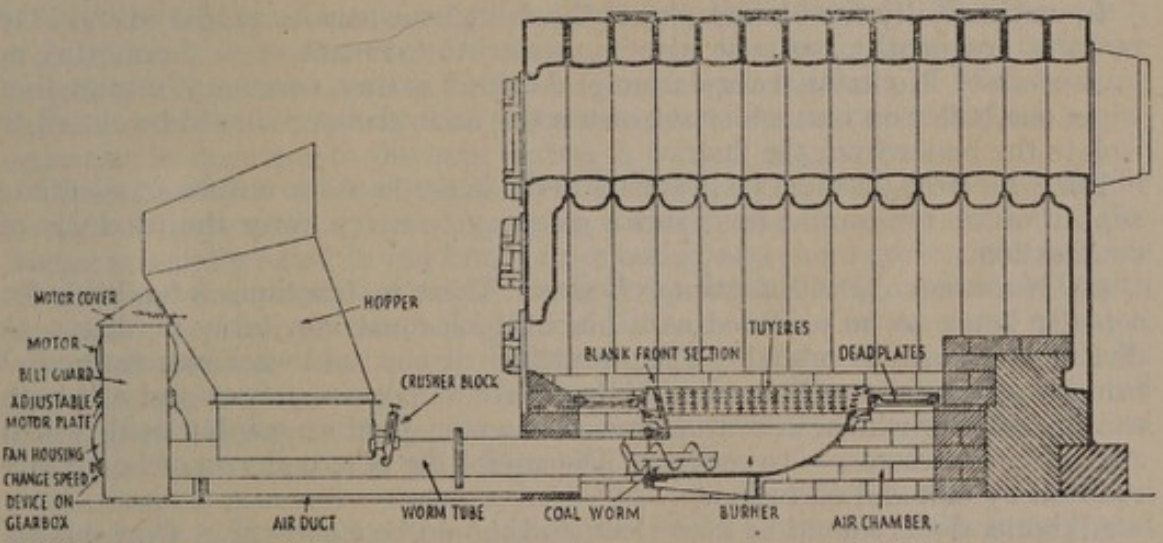


FIG. 224. Boiler with mechanical stoker.

much used for sectional heating boilers (Figs. 86 and 224). They consist of a steel fuel hopper in front of the boiler and a retort inside the boiler, the two being connected by coal feed and air supply tubes. Coal is fed into the retort from the hopper by means of a revolving worm in the coal tube driven by an electric motor. The motor also drives a small fan which blows air into the retort through a number of openings, this air assisting in the cooling of the retort as well as supporting combustion. The motor is under the control of a pressure switch or thermostat according to whether the boiler is for steam or hot water. The hopper usually holds six or eight hours' supply of fuel, and may be filled by hand or by chutes from overhead bunkers or by an elevator and conveyor.

This type of stoker will burn soft low grade coals of small size. A washed fuel is preferable, but not essential. The fuel enters the combustion zone from underneath, the volatiles are distilled off and pass through the burning fuel above, where they are consumed on or before leaving the fuel bed, so that fuels with high volatile content are burned in a smokeless manner. Much closer control is possible than by hand firing with consequent reduction in fuel consumption and a cheaper fuel can generally be burned. Against this, the capital cost of the stoker, repairs and replacements, and the cost of electric current must be considered.

Mechanical stokers of various types are used on large plants utilising Lancashire, Economic and similar boilers (Chapter XI).

(*c*) *Control of draught.* The correct control of draught is important if efficient combustion of fuel is to be obtained.

Hand fired boilers have three dampers, the ashpit damper, flue damper and secondary air damper. When the first two are fairly fully open the third is usually nearly closed. These two main dampers control the passage of air through the firebed and consequently the rate of combustion of the fuel, and their setting depends upon the amount of heat required. The secondary air damper controls the admission of the additional air to the combustion space above the fuel bed. Its function is to assist in the complete combustion of the gases rising from the fuel bed. After the fuel is first added to the fire the amount of secondary air required is at its maximum, but as the fuel burns through the secondary air can be reduced.

The use of damper regulators has already been described. Another method of damper control is to have the dampers coupled to small pulling motors which are operated electrically by a thermostat or pressurestat.

In mechanically fired boiler plants the draught is usually produced by a fan or fans, frequently, but not always, under the control of a thermostat or pressurestat. No manual regulation of dampers is then necessary, except that when one boiler on a range is out of use the main damper should be closed to isolate the boiler from the flue.

Inlets for fresh air must be provided to the boiler house to ensure an adequate supply of air for combustion, and a chimney to carry away the products of combustion.

(*d*) *Maintenance and Attention to Boilers.* There is sometimes a tendency for a boiler house to be regarded as a place which must inevitably be dirty. A dirty boiler house is probably also an inefficient one, and whatever method of firing is adopted, cleanliness and tidiness are important. Soot and ash both contain sulphur which, in conjunction with water, produce sulphur acids which are extremely corrosive to metals. The useful life of a boiler may be greatly reduced by dirt and corrosion.

All boiler doors should be kept clean and should be a good fit. Door hinges should be oiled occasionally and dampers and operating gear maintained so that they work freely without sticking.

The floor of the boiler house should not be allowed to harbour ashes and cinders, particularly in contact with the boiler base and the floor should be kept dry. All mountings should be kept clean and in proper working order. The pressure in steam boilers may occasionally be raised until the safety valve blows off as a test to ensure that it is working correctly.

Automatic stokers, pumps and similar apparatus should be given regular attention in accordance with the makers' instructions.

Boiler or pipe lagging should be kept in good repair.

Firebars of the grille pattern require renewal periodically, and a spare set should be in hand for use when required.

The firing and clinker doors usually have removable lining. This becomes burned and distorted in the course of time and must be renewed.

Water should not be emptied out of a hot water system unless absolutely necessary for repairs, etc., as each refilling with fresh water provides additional deposition of scale in hard water districts, or corrosion in soft water districts. No harm is done by leaving the system full of water during the summer months when it is not being used.

When the boiler is shut down at the end of the heating season it should be thoroughly cleaned out, the firebars and all unburnt fuel, clinker and ash removed, and the flues and metal smoke connection to chimney stack cleaned

and scraped. Where practicable, the surface of the fire pot and internal flues should then be painted with red lead paint to give protection against condensation. If necessary the chimney should also be cleaned. The inside of the furnace and flues should be examined for traces of corrosion.

While not in use the boiler should be left with all its doors and dampers fully open so that a circulation of air can take place through it and the chimney. This will assist in preventing condensation and so reduce the possibility of corrosion.

Before starting up at the beginning of the heating season it should be ascertained that the system is full of water, and accumulations of air should be released through the aircocks and air pipes.

(*e*) *Gas and Oil Firing.* Gas burners are available for attachment to ordinary sectional boilers. There are in addition several boilers on the market specially designed for burning gas, and also special types, incorporating forced draught fans, which burn a gas-air mixture in specially designed chambers or tubes, which may contain refractory material arranged to extract the maximum amount of heat from the gas before it passes into the chimney. Some of these boilers are highly efficient under close thermostatic control. Various safety precautions are essential to guard against escape of unburned gas due to failure to ignite. Flues also require baffles to guard against effects of wind blowing back into the boiler, and special precautions must be taken to deal with condensation in flues due to the high hydrogen content of coal gas.

When a boiler is to be oil-fired, firebrick baffles are usually necessary in the furnace to prevent the flame impinging directly on the boiler surfaces. The burner is usually fitted to the front of the boiler, and there are various systems of burning, the general principle being to atomise the oil and mix it with preheated air, the mixture being blown into the boiler in the form of a jet, where combustion takes place. Certain precautions are necessary, such as guarding against oil being delivered to the burners without being ignited, and shutting off oil supply in the event of flame failure. Automatic controls and safety devices form part of all oil burning equipment.

The fuel oil is usually stored in tanks outside the boiler house ; a separate service tank is provided in the boiler house holding say, one day's supply, and this is filled each day from the main tanks. This eliminates the storage of large quantities of oil close to the boilers. Precautions and regulations to be observed in the storage of oil should be carefully studied before using oil fuel.

THERMOSTATIC CONTROL

The application of thermostatic control to heating systems can result in considerable fuel economy. The methods of operation and utilisation are many and varied, controls being available for operation by electricity, compressed air, water pressure and other means. Some of the ways in which control may be applied to space heating are as follows :—

(1) Control of boiler draught (damper or fan control).
(2) Control of boiler fuel feed (mechanical stokers or gravity feed boilers).
(3) Control of flow temperature by thermostats placed at key points in the building.
(4) As in (1), (2) and (3), but with the addition of an external thermostat to anticipate changes in heat demand produced by changing external conditions.
(5) Individual thermostats in each room controlling the heating surfaces for each room separately.
(6) Control of heating by warm air by means of thermostats controlling heater batteries.

Thermostatic control is discussed in Chapter XXVI.

THE ECONOMICAL USE OF FUEL IN CENTRAL HEATING AND HOT WATER SUPPLY

(I) CENTRAL HEATING

In considering how material reductions in the consumption of fuel can be achieved the first question to be asked is whether more warmth is being provided than is necessary. A reduction of 2° F. may represent a saving of 10 per cent. in the consumption of fuel.

The temperature requirements, the efficiency of the installation and the method of operating the plant are all of importance. It will be convenient to consider these under eight headings and in the light of the measures to be taken to promote economy in the use of fuel :—

(i) *Temperature of Occupied Rooms*

Considerable economies can be made in times of fuel shortage by keeping the temperatures of the rooms from 3°–5° F. lower than is usual in normal times. Living rooms in flats and residences should be kept to 60° F. With low temperature hot water heating the temperature in the rooms should be controlled as far as possible by varying the boiler flow temperature according to outside weather conditions. Where this is impracticable, as for example with steam systems or intermittent heating, part of the heating in individual rooms should be turned off according to weather conditions. In rooms having more than one radiator, one or more should be turned off if the room becomes too warm. Where there is only one radiator in the room, the emission of heat from it can be reduced by a cloth jacket or other form of blanketing.

A guide to suitable flow temperatures of the boiler to vary with outside temperatures is given later (Section viii (8)).

(ii) *Reduction of Heat in Unoccupied Rooms*

Bedrooms, staircases, corridors, cloak-rooms and lavatories should not be heated except in severe weather to prevent damage by frost, to prevent deterioration of furnishings due to dampness and in so far as may be necessary to preserve the balance of the system. It will be realised that while normally some rooms are to some extent kept warm by an adjacent warmed corridor, others are not. If such a corridor is not warmed the heating may be unbalanced, some rooms being underheated, and in attempting to remedy this other rooms may be overheated. The waste due to this form of overheating may be serious.

(iii) *Cooling of Overheated Rooms*

In certain circumstances it may be possible, if a room should become too warm, to divert the unwanted heat to an adjoining room or corridor by opening a door. This is less wasteful than allowing the heat to escape out of a window.

(iv) *Draughts and Ventilation*

Draughts should be avoided as they reduce the warmth of the body, and it is important to ensure that the rate of air-change is not greater than necessary. Care should be taken to reduce the air-change when rooms cease to be occupied.

In schools, windows that are kept open while the classrooms are occupied should be closed during the mid-day interval, and the rate of combustion should be reduced during this period.

(v) *Infiltration of Air through the Structure and through Doorways*

To reduce the air-change to a minimum it is profitable to seal up cracks and crevices, to examine the fitting of windows and doors, and to fit draught excluders. It will sometimes be found beneficial to fit simple air-locks to external doors.

(vi) *Insulation of Hot Surfaces not used as Heating Surfaces*

A special survey should be made to ensure that heat is not wasted owing to the omission of insulation from hot pipes or surfaces which are not used as heating surfaces. All insulation should be adequate and kept in good repair.

(vii) *Thermal Insulation of Buildings*

Attention should be paid to the possibility of reducing the heat requirements of the building by the judicious use of insulation, particularly where roof losses are high.

The loss from single-sheeted walls is also high and should be similarly treated. It is advantageous to provide insulating boards on external walls behind radiators, particularly steam-heated radiators.

Ventilated wood floors are also a source of heat loss, which may be reduced by suitable insulating material, e.g. a carpeted floor may be underlaid with newspapers.

Detailed information on this subject will be found in Fuel Efficiency Bulletin No. 12, obtainable from the Ministry of Fuel and Power.

(viii) *Operation of Boiler and Plant*

Much thought may profitably be given to the consideration of what regime should be adopted in the operation of the plant. The relative merits of continuous and intermittent heating depend upon a variety of factors, including the nature of the building and the duration of occupancy. A considerable amount of heat has to be provided to heat up a building of substantial construction which has been allowed to become cold, and in such buildings it is advantageous to bank the boiler at night. In buildings of sheeted construction the fire may be let out at night, except in severe weather when there is risk of frost.

The principles of operation of hand fired central heating boiler plant differ from those discussed for steam boilers in other chapters, in so far as they have accommodation in the fire-box for a six to eight hours' fuel charge, while certain types, known as "magazine" boilers, have storage capacity apart from the fire-pot for a charge up to twenty-four hours. For this reason, management of this type of boiler is described in some detail below.

The main supply of air for combustion is from underneath the fuel bed, and for efficient combustion too much air should not be admitted above the fire.

The proper balance is provided when, with a well distributed fuel charge, the air inlets of the ashpit and the apertures in the firing door are both open, but the firing door itself tightly closed.

The output of the boiler should be regulated by means of the damper in the flue pipe. Excessive chimney pull can be reduced by admitting air direct into the chimney.

To secure the fullest economy, the depth of the fuel bed should be maintained within the following limits :—

Size of Fuel	Depth of Bed	
	Coke	Anthracite
1″ to 2″	9″ to 15″	6″ to 10″
2″ to 3″	12″ to 18″	8″ to 15″

The boiler should never be filled to more than three-quarters of the capacity of the fire-box if coke is used, and with anthracite to not more than half capacity. These limits should be observed when banking for the night.

Certain common faults in hand firing central heating hot water boilers can be avoided if the following recommendations are observed; most of these recommendations apply also to low pressure steam and domestic hot water boilers :—

(1) The fire-door should fit properly and not be opened unnecessarily.
(2) The boiler output should be controlled as far as possible by using the dampers at the back of the boiler ; too much air should not be allowed to enter the boiler as secondary air above the fuel bed.
(3) Air inleakage should be prevented. All likely points, joints, dampers, doors, etc., should be tested periodically with a lighted taper.
(4) Clinker and ashes should not be allowed to accumulate on the firebars, as they increase the resistance of the fuel bed and cause too much secondary air to be drawn in.
(5) The fire should not become too hot, since an intense fire destroys the bars, promotes clinker formation and may increase the amount of gas that escapes unburned.
(6) The fuel must be evenly distributed over the grate bars. It should not be disturbed unnecessarily as this promotes clinker formation (cf. Chapter VI).
(7) When the fire has been refuelled or restarted in the morning it should not be left too long unattended. The dampers should be adjusted to suit the demand for heat as soon as the fire has become established.
(8) The boiler flow temperature should be varied according to the weather, the following figures being a convenient guide.

Outside shade temperature—° F.	55	50	45	40	35	30
Boiler flow temperature—° F.	..105	120	135	150	165	180

(9) It is generally better to bank the fire for the night, or to keep a slow fire, than to force the boiler in the morning. When a fire is to be banked it should not be thin, and the bulk of the fuel should be heated to dull redness before the dampers are adjusted for the night.
(10) Boiler flues should be cleaned at least once each week, and more frequently if bituminous coal is used ; soot and ash prevent the transmission of heat from the flue gas to the water in the boiler.
(11) Fuel should be kept dry. Unburnt cinders should be recovered from the clinker and ashes and put back on to the fire. Bituminous coal should not be used in central heating boilers, except in emergency, as it coats the heating surfaces with soot.

The general technique of banking and stoking is discussed in Chapter XI.

(2) CENTRAL HOT WATER SERVICE

(i) *Temperature*

Water for baths and hand washing is required at a temperature between 100° and 110° F. The temperature of the secondary flow leaving the storage cylinder should be so adjusted that the temperature of the water at the taps is very little in excess of the above figures, and certainly not more than 120° F. This can usually be attained with a flow temperature of not more than 135° to 140° F. The lower this temperature can be kept the less heat is lost by radiation and convection from the circulation.

Hotter water required in small quantities is best obtained by providing additional heat at the point of use as, for example, for washing greasy dishes where a temperature of 140° to 160° F. is required. The loss from pipes containing water at 160° F. is at least 25 per cent. more than from similar pipes containing water at 140° F. It is therefore more economical to make provision locally for boosting the temperature of water for dish washing. For hotels and hospital kitchens this is usually done by thermostatically controlled calorifiers fed from the hot water service and heated up to the required temperature by steam, gas, or electricity.

(ii) *Capacity of Boiler and Storage Vessels*

Boilers should not be forced to meet peak demands and when these are likely to occur storage capacity should be increased, or if this is impossible, the water in the cylinders should be stored at a higher temperature, an automatic mixing valve being provided to limit the temperature of the water in the secondary circulations. The quantity of hot water stored should be sufficient to meet the peak demand, but should not be greatly in excess of this.

If, however, it is found that with sufficient storage capacity it is still necessary to force the boiler, then the boiler power must also be increased.

(iii) *Loss from Circulating Systems*

In many domestic hot water systems the amount of fuel required to make good the loss of heat from the circulating system accounts for half the total fuel required. Both circulating pipes and storage vessels should therefore be effectively lagged.

Loss may also occur through leaking taps, but the principal loss of heat from the system is generally through water withdrawn for use. Any saving that can be effected in this respect is a positive economy.

Loss of heat from the circulating system may also be reduced by interrupting the circulation during periods of no demand, e.g. during the night in blocks of flats. To do this, the secondary return valve should be closed during the no-demand period. Letting the boiler out at night does not lead to the same result, because a gravity circulation will take place through the system and will cool down the water in the storage vessels. Until this water has been reheated, the water at the taps will remain cold. The proper practice is thus to close the valve on the secondary circulation and just maintain temperatures by banked fires.

(3) COMBINED HEATING AND VENTILATION PLANTS

The combined heating and ventilation plants previously described can be arranged in two groups :—

Group (*a*) Those in which fabric and infiltration losses in the building are offset by heating surfaces situated directly in warmed spaces, and the air for ventilating is introduced at or near room temperature (Fig. 220, page 614).

The local heating surfaces are the pipes, radiators, or panels of a hot water or steam heating system, or they may be electric or gas-heated surfaces. This system is used for offices, public buildings, hotels, restaurants, cinemas, theatres and halls, swimming baths, gymnasiums, operating theatres, departmental stores and all buildings where there is likely to be overcrowding.

Group (*b*) Those which heat a building solely by the introduction and distribution of hot air (Fig. 221, page 614).

A proportion of the air at room temperature is usually re-circulated by a fan. In large buildings there may be one fan and heater with distributing ducts or a number of smaller fan and heater units each with a connection to the outside air and provided with a re-circulation damper.

Many factories and workshops are heated and ventilated in this manner and the system is also used occasionally for offices and public buildings.

The ventilating plant for both groups consists of an air filter or washer for cleaning the incoming air, a heater battery for warming the air and a fan for

passing the air through the filter and heater battery using hot water, steam, electricity, gas, oil or solid fuel, and delivering it through ducts to the distribution points in the heated spaces.

The chief factors affecting the economical operation of combined heating and ventilating plants are :—

(1) The method of working the boiler or heat generating unit and the control of the distribution system serving direct heating surfaces and air heaters.
(2) Thermal insulation of hot surfaces not being used for useful heating and insulation of parts of the structures where the heat losses are unduly high.
(3) Volume of fresh air introduced by ventilating plant.
(4) Temperature of heated spaces and of the air introduced by the ventilation plant.
(5) Leakage of warm air and infiltration of cold air.
(6) Air movement, diffusion and distribution.
(7) Re-circulation of warmed air.

These factors will now be considered in turn.

(1) *and* (2) *Boiler Operation and Insulation*

These items are common to all types of heating plants and are dealt with elsewhere. Insulation should be applied to the casing of air heaters and to the ducts in systems of Group (*b*) conveying heated air through unheated spaces.

(3) *Volume of Air*

Two-speed or variable-speed fans are used to enable more fresh air to be delivered into a building in summer than in winter. The volume may also be varied by regulating dampers. Since all the incoming air must be heated in winter, the quantity taken in should be reduced to a minimum, depending on the number of occupants and the working conditions.

(4) *Air Temperature*

The difference between the internal temperature of a heated space and the external air affects in a direct ratio the quantity of fuel required for heating. To conserve fuel the internal temperature should not be higher than is necessary.

With Group (*a*) plants, the temperature of the warmed air should not exceed the required temperature of the room and the air may be introduced a few degrees below that temperature, except in rooms which are sparsely occupied.

With Group (*b*) plants, the temperature of the hot air introduced into the heated space is determined by the weight of air circulated through the building and the amount of heat which the air has to impart to the fabric of the building. For a given heat quantity, the requirements can be met by a small air volume and a high air temperature or a large air volume and a low air temperature. Where the ratio of occupants to cubical content is small, such as in large factory buildings, it is more economical to circulate the large air volume with the low air temperature as the increased cooling effect of the air movement corresponding to the larger volume is relatively small, whereas the higher air temperature tends to increase the heat losses through that part of the fabric with which the hot air comes in contact during diffusion.

With Group (*b*) plants, since the hot air provides the whole of the heat for the building, a higher air temperature is required in cold weather than in mild weather. Therefore, the temperature of the air leaving the heater should be regulated from time to time according to the external temperature to maintain the required temperature in the heated spaces at the working levels.

Most modern combined heating and ventilating plants are equipped with

automatic controls operating on the heater batteries. In Group (*a*) plants, automatic control ensures that the air is introduced at a steady temperature irrespective of changes in the external temperature. Group (*b*) plants should be arranged so that the air temperature is not higher at any time than is necessary to maintain the required internal temperature. By such control no unnecessary heat is added to the air and waste is avoided.

(5) *Effect of Natural Circulation*

When the ventilating fans are stopped cold air may flow by gravity into the building through the intake or discharge openings and the warm air initially in the building will escape. Thereafter the continued interchange of air absorbs heat from the fabric of the building. All this heat must be restored when the plant is re-started. The loss of heat involved can be materially reduced by closing the main dampers when the fans are idle. If this is impossible or inconvenient, supplementary dampers should be added or detachable covers provided for all discharge openings.

Since the plant provides the necessary fresh air, all windows and skylights should be kept closed during the heating season to avoid loss of hot air. In buildings of sheeted construction all constructional joints should as far as is practicable be rendered airtight. Spaces formed by corrugated sheets on the valleys and ridges of roofs and air spaces at the lower edges of patent glazing should be sealed to prevent air leakage. This is particularly important in Group (*b*) plants, owing to the higher temperature at which the heated air is delivered.

Any natural ventilators in use should be provided with dampers so that they can be closed when not actively required for the working of the plant or for clearing process fumes.

(6) *Air Movement*

Air movement has a marked influence on sensations of warmth. Increasing the air velocity from 25 to 120 feet per minute may require 20 per cent. more heat to maintain the same degree of comfort. For winter working, therefore, the volume of fresh air introduced to the plant should be kept to a minimum and also the total volume of air delivered to the rooms should not be so large as to cause excessive air movement.

Variation in air velocity that generally occurs in heated spaces in the region of some inlets may be corrected by altering the volume of air by adjustment of the local regulating device, and variations near to an inlet by fixing deflector plates or grilles to break up the air stream and distribute it over a wider area.

The distribution of the hot air in Group (*b*) plants has an important bearing on the amount of heat necessary to produce the required temperature at the working level. Except in special circumstances the deflecting louvres or nozzles on the inlet openings should be adjusted so that the air stream is directed towards the working level and does not impinge on any part of the structure, particularly the roof or outer walls. If the air stream impinges on the roof or outer walls the heat loss from the structure is increased.

Regulating dampers at the different outlets should also be adjusted so that the temperature of the diffused air in the horizontal plane is as uniform as possible.

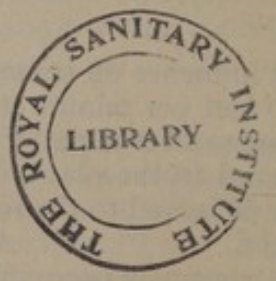
When adjustment of the regulating dampers does not produce uniformity in the temperature of the diffused air, instead of heating the whole of the air to an unduly high temperature for the sake of one section of the system, a booster heater may be installed in the branch duct serving the particular section, or alternatively local heating surfaces may be provided.

(7) *Air Re-circulation*

During the heating season, when the intake of fresh air is at a minimum, a proportion of air from the heated spaces should be mixed with the fresh air to reduce the load on the air-heating battery.

In Group (*a*) plants serving spaces where people are assembled in numbers varying with the occasion, the quantity of re-circulated air will require adjustment, but in plants serving spaces occupied by a regular number of people and in most Group (*b*) plants, the quantity of re-circulated air need not be varied.

In plants where no provision for re-circulating the air has been made, consideration should be given to the addition of re-circulation equipment where this is practicable.



CHAPTER XXV

DRYING

Vapour pressure and the principles of drying—The practice of drying—Types of drying plant—Dust collection—Control of conditions—Thermal efficiency—Technical data—Heat losses—Methods of heating, furnaces and fuels.

DRYING, or dehydration as it is sometimes termed, is widely necessary in the processing of material for industrial purposes and for preserving foodstuffs. Many materials which in the past were dried naturally in ordinary atmospheric conditions are now dried artificially, as natural conditions of temperature, humidity and wind are too variable and too slow to suit modern requirements. In artificial drying, conditions of temperature, air flow and humidity can be controlled to ensure rapid drying without detriment to the material being dried.

New methods of using materials other than metals, generally involve drying and increasingly large quantities of fuel are required for this purpose. Some materials require to be dried because it is necessary or desirable to remove moisture but generally, and particularly with foodstuffs, drying also serves to sterilise the material so that it can be kept for a long period without deteriorating.

Drying or dehydration of materials particularly foodstuffs, results in an enormous saving in transport. Many foodstuffs contain 70–85 per cent. of water, and the following advantages accrue from drying :—

(1) Foodstuffs, when dried under suitable conditions, contain practically the full food value of the original material.
(2) In drying, the foodstuff is sterilised and in the dried state will keep for a very long time without deteriorating.
(3) Dried foodstuffs occupy less than half the space and some are only one-fourth of the weight of the material before drying.
(4) As a result of research in the drying of foodstuffs, dried food is now being produced which, when soaked and cooked, is almost indistinguishable from the original fresh food.

VAPOUR PRESSURE AND THE PRINCIPLES OF DRYING

The molecules of gases are in constant motion at very high velocity. The pressure exerted by a gas or vapour arises from the bombardment of the containing walls by the gaseous molecules. In a mixture of gases each gas contributes its share of this pressure and the pressure exerted by any individual gas in the mixture is its concentration in percentage by volume multiplied by the total pressure and divided by 100.

Thus if a gas consisting of nitrogen, 70 per cent. ; oxygen, 14·7 per cent. ; and water vapour, 15·3 per cent., is under a total pressure of 31 inches of mercury, the " partial pressure " of each of the constituents is as follows :—

			In., mercury	Mm., mercury
Nitrogen :	70	per cent. of 31 =	21·70	551
Oxygen :	14·7	,, ,, of 31 =	4·56	116
Water vapour :	15·3	,, ,, of 31 =	4·74	120
			31·00	787

When water is in contact with a gas, the molecules of water leave the liquid and enter the gas as vapour ; similarly some water vapour molecules in the

gas return to the liquid. When the rates of these two reactions are equal, the gas is said to be saturated with water vapour ; until equality is reached, water will evaporate from the liquid, or water will condense from the gas, according to which reaction predominates. The partial pressure of the water in the form of vapour in the gas space at saturation is known as the " saturation pressure," or the " vapour pressure " of the water.

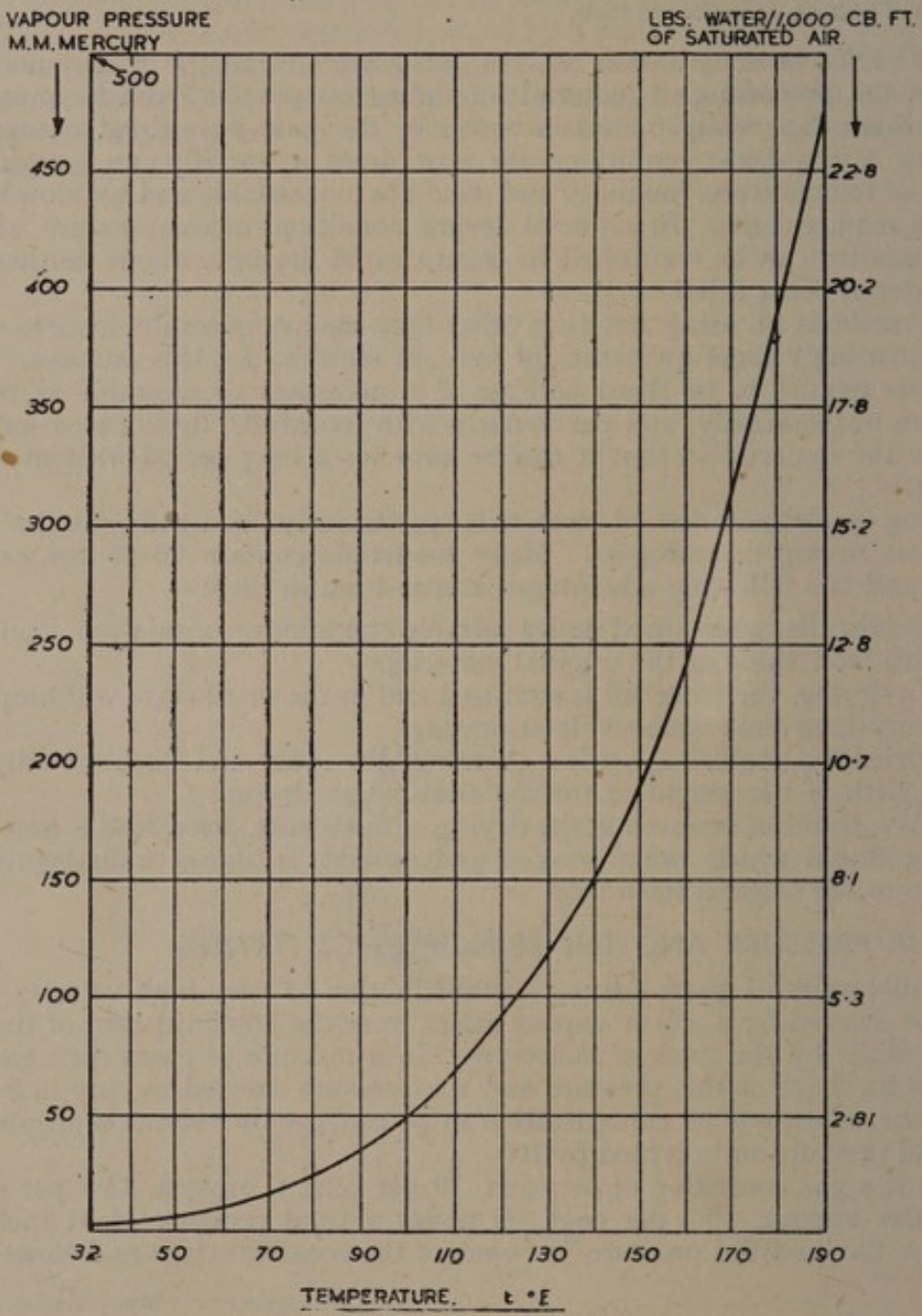


FIG. 225. Vapour pressure of water.

The vapour pressure of water depends on the temperature as shown in Fig. 225 ; the higher the temperature, the greater is the tendency of water to vaporise, or " evaporate," and the more readily wet substances become dry by evaporation of the water they contain. When the vapour pressure of water becomes equal to the pressure of the atmosphere above it, water boils. The general dependence of the boiling temperature of water on the pressure above it can be seen from Chapter VII and from the Steam Tables in Appendix II.

The vapour pressure is reduced by the presence of dissolved substances, so that it is more difficult to dry a material wetted by a solution than one wetted with pure water, other conditions being similar.

These statements are true of liquids generally, though, of course, each liquid has its own peculiar vapour pressure curve. That in Fig. 225 is given for water only, since water is the liquid with which it is necessary to deal in drying problems.

When a wetted substance is brought into contact with dry air, water vapour molecules leave the substance and enter the air. The air against the wetted surface thus becomes saturated with water vapour, and no more water can leave the substance until some of that which has done so diffuses into the main body of the air. Diffusion is a slow process, and it is clearly desirable that the air shall be in motion. The rate of drying will thus be increased by the velocity of the air. The physical mechanism of the flow of fluids over surfaces (Chapter IX) will be of interest in this connection.

The rate of drying will also depend upon the difference between the concentration of water vapour in the air and the maximum concentration at saturation point. The rate is a maximum when the air is dry, and falls to zero when the air is saturated. Thus the air used for drying will also leave the drier in an unsaturated condition, i.e. containing less than the maximum amount of water vapour than it could theoretically absorb.

The maximum amount of water vapour that can be vaporised into air (or any other gas, or a vacuum) increases greatly at higher temperatures, as is shown in Fig. 225. Consequently, when practical conditions connected with the nature of the product permit, higher temperatures are used to increase the rate of drying.

When the water content of the gases is less than the full saturation amount, the degree of saturation is expressed as a percentage in a manner analogous to the dryness fraction of steam. Thus if at a particular temperature saturation pressure is 387 millimetres and the water content of the air is 20 lb. per 1,000 cubic feet, if the gases are leaving the drier with 80 per cent. saturation, the vapour pressure of the water they contain is 80 per cent. of 387, or 309·6 millimetres, and their content of water is 16 lb. per 1,000 cubic feet.

THE PRACTICE OF DRYING

The broad principle of drying is thus that material is heated in an atmosphere which is kept below its moisture saturation point, the atmosphere then absorbing moisture from the material ; the rate of evaporation is proportional to the difference between the vapour pressure of the evaporating water and the vapour pressure of water in the surrounding atmosphere.

The process of drying may be divided into two stages. Firstly, the evaporation of surface moisture which is affected by conditions external to the material, such as humidity, temperature and velocity of air passing over its surface. Secondly, penetration of heat into the material and the rate of flow of moisture or diffusion from the interior to the surface. These are factors which differ with the nature and structure of various materials, and it is for this reason that the problem of drying cannot be dealt with purely by thermodynamic or empirical formulæ.

The capacity of air or gas to absorb moisture increases rapidly with increase in temperature. The absorption capacity as shown in Fig. 225 is increased about fifteen times for 100° F. rise in temperature.

Table 115 (given later under " Technical Data ") shows the percentage saturation of air at different temperatures and the corresponding pounds of moisture per 1,000 cubic feet for various degrees of humidity as measured by a wet and dry bulb hygrometer.

The material to be dried may be heated by convection, conduction or by radiant heating or by a combination of these. In addition to heating it is necessary to carry away the moisture liberated from the material by a current of air or gas flowing over it.

In most driers the material to be dried is heated by convection from a current of hot air or gas flowing through or over the material ; the current of air also carries away the moisture liberated.

In some driers, principally of the steam-jacketed type, the material is tumbled over and over and is heated by conduction in its contact with the steam-heated surfaces of the drier. Another example of this type is the cylindrical film drier, in which plastic or semi-liquid material is spread over a steam-heated cylinder and the material is heated by conduction from the hot surface of the cylinder.

Radiant heating is seldom employed as the principal method, although in all driers the material is heated to some extent by radiation from the hot surfaces of the drier. The material is heated by radiant heat from suitable heater elements, adjacent to, but not in contact with, the material.

The essence of satisfactory drying is to dry uniformly without losing any desirable qualities or properties in the process. There are obvious difficulties in drying uniformly material which is in pieces of different size, as heat will penetrate and quickly dry the smaller pieces, which may then become overheated and scorched before the larger pieces are dry. Material which is fibrous may be matted or denser in places and the looser fibres may be dried and overheated before the denser patches are dry.

The temperature at which drying can be effected is dependent on the nature and structure of the material, its moisture content and time of drying. For these and other reasons there are many types of driers to suit different materials, and the whole technique is a highly specialised branch of engineering.

MOISTURE CONTENT OF DRIED MATERIAL

When a substance is termed " dry," this does not usually mean that it contains no moisture. Most organic materials, if dried below a certain moisture content will re-absorb moisture from the atmosphere, the amount varying from approximately 12 to 16 per cent. for different materials. It is generally undesirable to dry much beyond the point at which moisture would naturally be re-absorbed from the atmosphere, and it will generally be harmful to the structure and nature of the material to dry it completely. Overdrying may, therefore, be as harmful as underdrying.

Inorganic materials, such as iron ore, salt, sand, etc., are dried to lower moisture contents of 1 per cent. or less.

AIR VELOCITY

The velocity of the hot gases over the surface affects the rate of drying ; the greater the velocity the greater the rate of evaporation. There are limitations in velocities, varying with the nature of the material. In rotary driers, when drying light or dusty material—or material which becomes dusty when being dried—the hot gas velocity through the drier may have to be kept as low as 1–3 feet per second, but when drying heavier materials which do not tend to become powdery during drying, speeds of hot gases may safely be as high as 12–15 feet per second.

DRYING TEMPERATURES

It is essential that no substance should be subjected to temperatures which will adversely affect its structure or desirable qualities during drying.

The drying temperature and rate of drying are governed by the nature and structure or texture of the material, as well as by its moisture content, heat conductivity and what might be termed its freedom to liberate moisture.

When material containing moisture is surrounded by hot gases, its temperature will not rise to that of the hot gases until it begins to become dry, because heat is being absorbed as latent heat in evaporating moisture and not purely as sensible heat in raising the temperature. It is no more harmful to subject many materials to temperatures of 1,200° or 1,400° F. for a short time in the initial stages of drying than to dry them at a temperature of 240° F. for a long time. Material in high temperature driers is rarely in contact with drying gases exceeding 800° F. for more than 7 or 8 seconds, whereas in low temperature driers it may be in contact with drying gases at 250° F. for 30 or 40 minutes.

Most substances containing and giving off moisture can, therefore, be subjected to a high temperature for a short time without harmful effects. There are, however, some exceptions to this ; for instance, some materials, if initially subjected to too high a temperature, tend to bake and form an outside skin which seals and prevents liberation of moisture, with the result that the material will bake and burn outside before the centre is dry.

When drying grain which is to be used for seed, if it is subjected to too high a temperature in the initial stages of drying, its germinating properties are destroyed, but as it becomes dry the temperature can be increased without harmful effect.

TYPES OF DRYING PLANT

There are many types of plant made for drying various classes of materials, but, broadly speaking, they may be classified as follows :—

Drying rooms and chambers.
Tray driers.
Conveyors.
Tunnel driers with wheeled trucks.
Simple rotary.
Double shell rotary.
Special type of rotary (proprietary designs).
Vertical cylindrical.
Film driers.
Spray driers.
Pneumatic driers.
Air-swept rotary mills.
Vacuum driers.
Sundry driers, not included in above.

Drying Rooms and Chambers. A simple method of drying is to place the material to be dried in a room or chamber through which hot air is circulated. This method is particularly applicable to the drying of large bodies such as building slabs, bricks, wallboard, fibre board, clothes, foundry cores and so forth.

The material to be dried is symmetrically stacked or placed in the drying room so that the maximum surface of each piece is exposed for drying. It may also be stacked on wheeled trucks.

Hot air is circulated through ducting suitably arranged in the building with outlets to discharge the hot drying air or gases uniformly over the surfaces of the material being dried. To obtain maximum thermal efficiency the hot air, after circulating through the room, is drawn off, re-heated and re-circulated by

the furnace and fan. The humidity is controlled by allowing a proportion of hot air to escape and admitting fresh air in its place.

For efficient drying it is essential that all the material be subjected to equal hot gas flow and temperature. Open fans are sometimes fitted in the chamber to equalise the temperature.

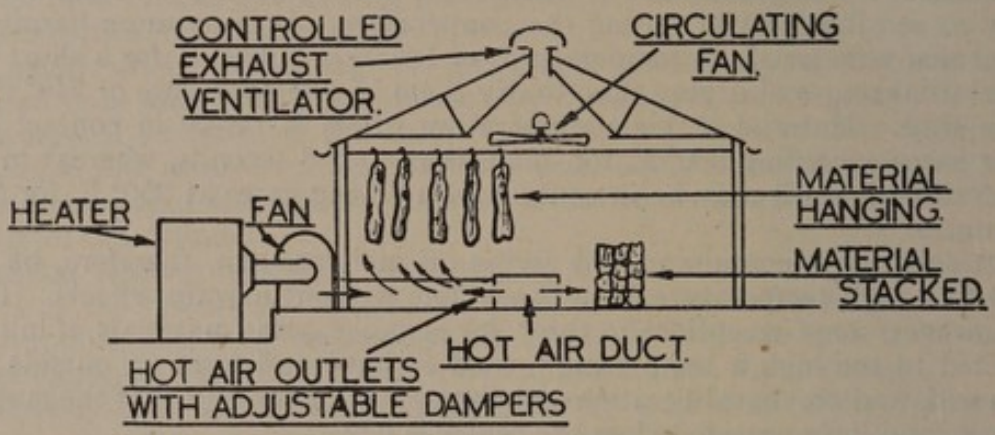


FIG. 226. Chamber drier.

Fig. 226 shows an arrangement of a drying room. Hot air or gases from a heater discharged by a fan to a duct running through the centre of the building, are uniformly discharged over the whole floor of the building. In addition an overhead circulating fan or fans, as shown, are sometimes fitted to provide more uniform distribution of the hot air. It is usual to maintain a fixed drying temperature, and to control humidity in the drying room by exhaust ventilator or by re-heating and re-circulating a part of the hot air in the chamber.

The material to be dried may either be suspended from hooks or be suitably stacked to ensure uniform flow of hot air around all its surfaces.

Tray Driers. This is a method for drying small goods, such as small foundry cores, colours, chemicals, food products. The trays are perforated and the hot gases pass upwards through the perforated bottoms and through the material, or the hot gases pass to and fro over the trays.

The trays sometimes slide into racks in a drying chamber or, for convenience in handling, they may fit into racks in a wheeled trolley which is pushed into

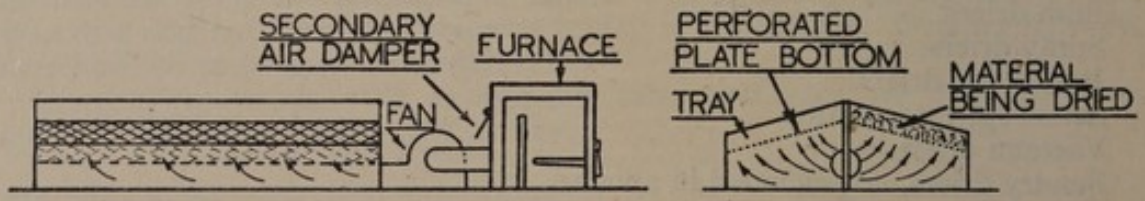


FIG. 227. Simple tray drier.

the drying chamber. The system of hot gas ducting and re-circulation and control of humidity is similar to that for chamber drying already described.

Fig. 227 shows the simplest type of tray drier. Hot gases from a furnace or heater are discharged by fan into a space below the trays. The trays have perforated bottoms which allow the hot gases to pass up through the material on the trays. With this simplest form of tray drier the efficiency is low as towards the end of the drying period the hot gases take up very little moisture and, therefore, leave the material at a very low degree of humidity and consequent low efficiency.

A more efficient form of simple tray drier comprises two or more superimposed trays (see Fig. 228). The material can be half-dried on the top trays

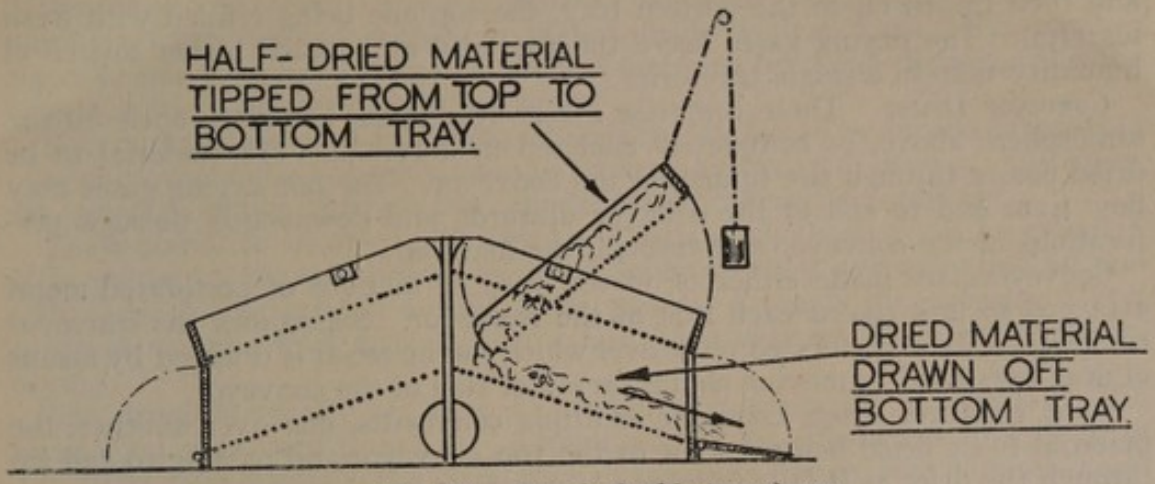


FIG. 228. Simple double-tray dryer.

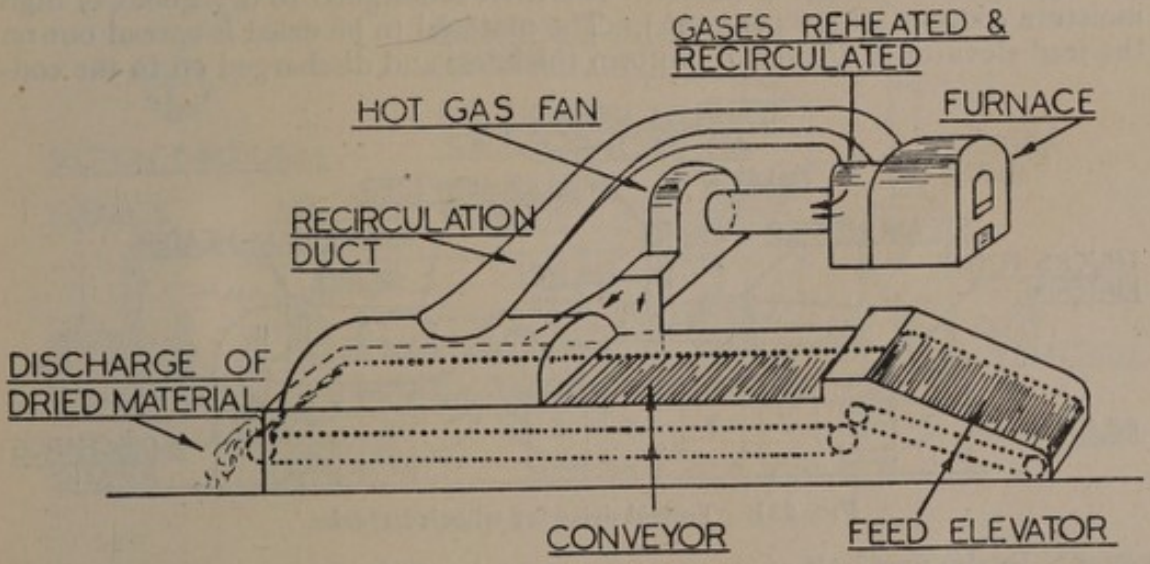


FIG. 229. Conveyor dryer.

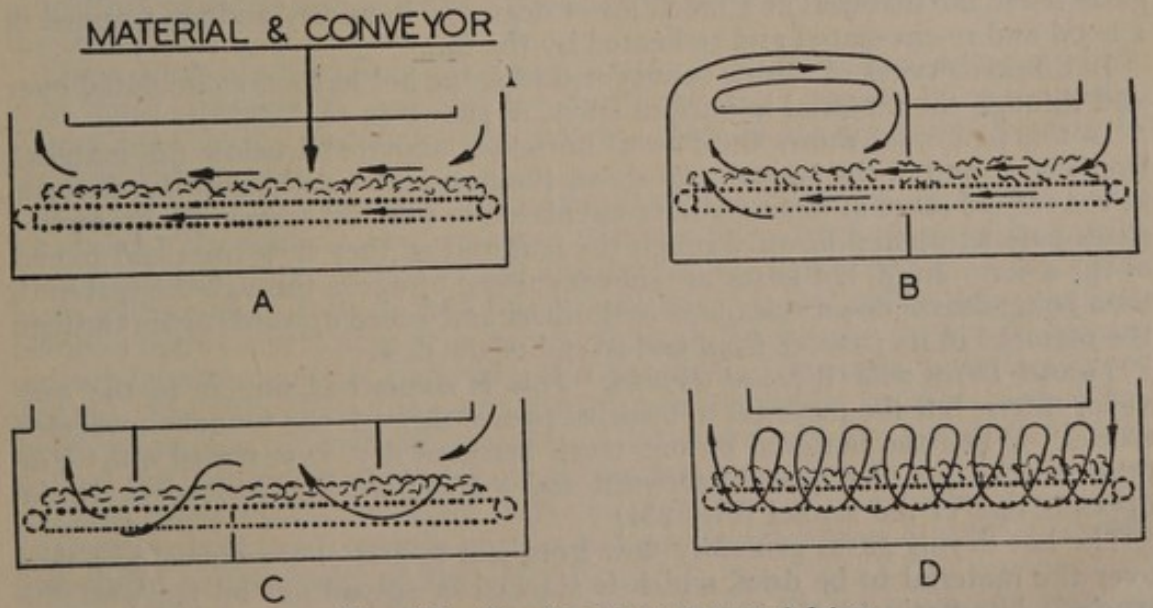


FIG. 230. Gas flow through tunnel driers.

and then tipped on to the bottom tray, the top one being refilled with fresh material. The drying gases leave the top tray at a much higher degree of humidity than in a single tray drier.

Conveyor Driers. These comprise a conveyor open, or partly open, to the atmosphere above, or completely enclosed in a tunnel. The material to be dried passes through the tunnel on the conveyor. The hot drying gases may flow from end to end of the drier or upwards and downwards through perforations in the conveyor and through the material.

Conveyors are made either of woven wire, or sections of perforated metal attached to link chains each side of the conveyor. Sometimes the conveyor comprises a fixed perforated plate over which the material is dragged by means of drag-bars fixed to moving chains on either side of the conveyor.

Some conveyor driers comprise multiple conveyors, one over another, the material to be dried being fed on to the top conveyor and passing to and fro through the drier as it falls from one conveyor to another.

Fig. 229 shows a semi-open conveyor, as distinct from those in which the conveyor passes through a tunnel. This drier is designed to dry goods of high moisture content (76–80 per cent.). The material to be dried is spread out on the feed elevator in a layer of uniform thickness and discharged on to the con-

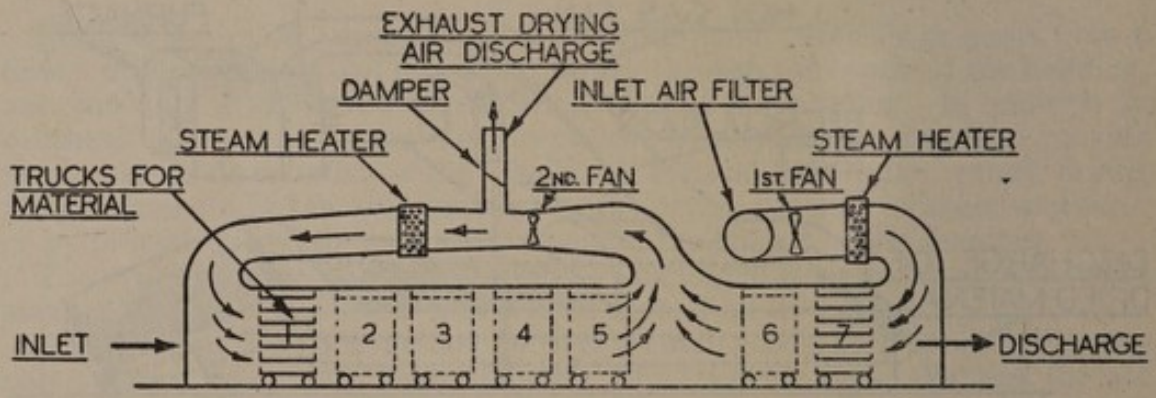


FIG. 231. Tunnel drier for wheeled trucks.

veyor. In the first half of its passage along the drier, hot gases pass through the material and are discharged at a high degree of humidity, as the material contains a considerable amount of moisture. In the second half, or stage, the gases leave the material at a much lower degree of humidity and are trapped in a hood and re-circulated and re-heated by the fan.

In different types of tunnel conveyor driers, the hot gases are circulated over and through the material in various ways, as shown in Fig. 230.

In this figure, (*a*) shows the flow of hot gases above and below the material from end to end of the tunnel ; (*b*) shows the flow above and below the material in the second stage of drying and re-circulated in the first stage ; (*c*) shows hot gases passing up and down through the material as they flow from end to end of the drier. In (*d*) hot gases are shown passing upwards through the material, then being drawn down side ducts in the drier and forced upwards again through the material in its passage from end to end of the drier.

Tunnel Drier with Wheeled Trucks. This is somewhat similar to the conveyor driers, but the material to be dried passes through the tunnel on wheeled trucks. When the material on one truck becomes dry, it is pulled out. The other trucks are then pushed forward and a fresh truck is pushed in at the opposite end of the tunnel (Fig. 231).

The hot drying gases generally flow from end to end of the tunnel and flow over the material to be dried which is stacked or spread out on trays on the trucks. Alternatively, the hot gases may be circulated as shown in Fig. 231.

Simple Rotary Driers. These comprise a horizontal rotating cylinder with a number of longitudinal shelves or flights inside it (Fig. 232). The smallest driers of this type have cylinders about 6 feet long by 2 feet diameter and the largest, 8 feet diameter and 70 feet long.

Rotary driers are used for drying material which has to be turned or tumbled over and over in the hot gas stream to ensure uniformity of drying. Hot gases are generally drawn through the cylinder by an induced draught fan.

The material to be dried is fed into and falls to the bottom of the cylinder. It is picked up by the shelves as the cylinder revolves and is spilled off and falls through the stream of hot gases passing through the cylinder. This constant movement of material in the hot gas stream results in uniformity of drying.

The drum is usually set at a slight inclination towards the outlet. The inclination may vary from 1 in 16 for quick drying substances to 1 in 30 or 1 in 40 for slow drying. The inclination of the drum, shape, width, number and form of the shelves or lifting flights, are determined by experience to produce

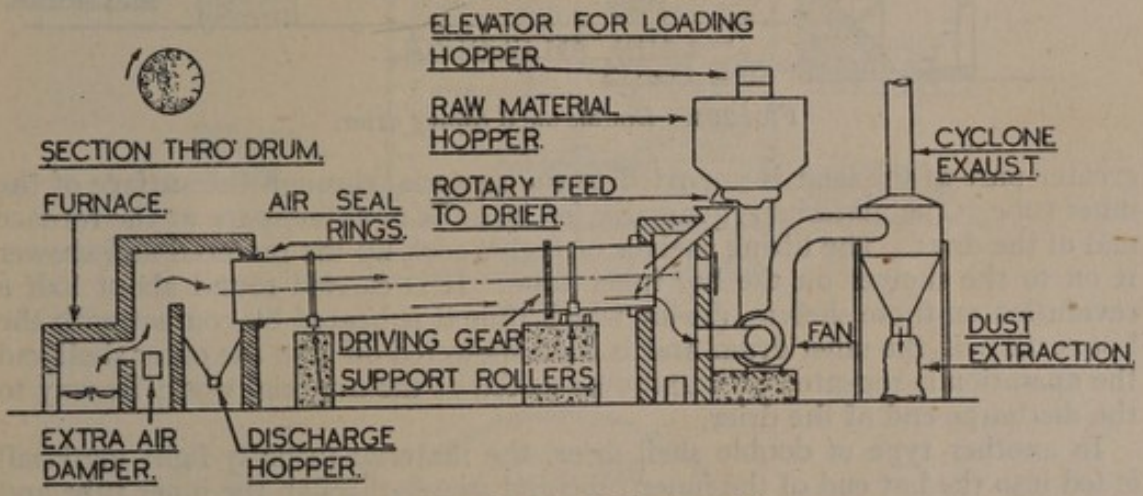


FIG. 232. Single shell rotary drier.

the best showering effect and rate of feed of the material through the stream of hot gases. At the feed end of the drier, spiral lifters are sometimes fitted to propel the material well into the cylinder and prevent spillage over the end of the drum.

These driers are used for drying sticky material such as clay, material which is in fairly large lumps, and also for chemicals and a variety of other goods. Where possible the material is fed into the drier in the opposite direction to the flow of hot gases.

It is not possible to employ the contra-flow principle when drying materials which are very light or powdery when they are dry. Such material is usually passed through the drier in the same direction as the flow of hot gases and there is an advantage in this, in that the part of the material which dries quickest becomes lighter and is, therefore, carried through the drier by the hot gas stream quicker than the moister material ; this automatically assists in uniformity of drying. The rate of feed through the drum is controlled by the speed of hot gases, by sloping the cylinder down to the outlet end and by sloping the shelves to feed the material more rapidly to the outlet as necessary.

These driers evaporate about 2–2¼ lb. of water per cubic foot of cylinder volume per hour and operate at inlet hot gas temperatures of 800°–1,500° F. and outlet temperatures of 250°–450° F.

Rotary Driers—Double Shell. In this type of drier the hot gases pass through

a central tube and return through the annular space between the inner and outer drums and are exhausted by an induced draught fan (see Fig. 233).

Lifting shelves or flights are fitted inside the outer drum and outside the central tube.

Hot gases enter the inner tube at a temperature of 1,000°–1,500° F., and a

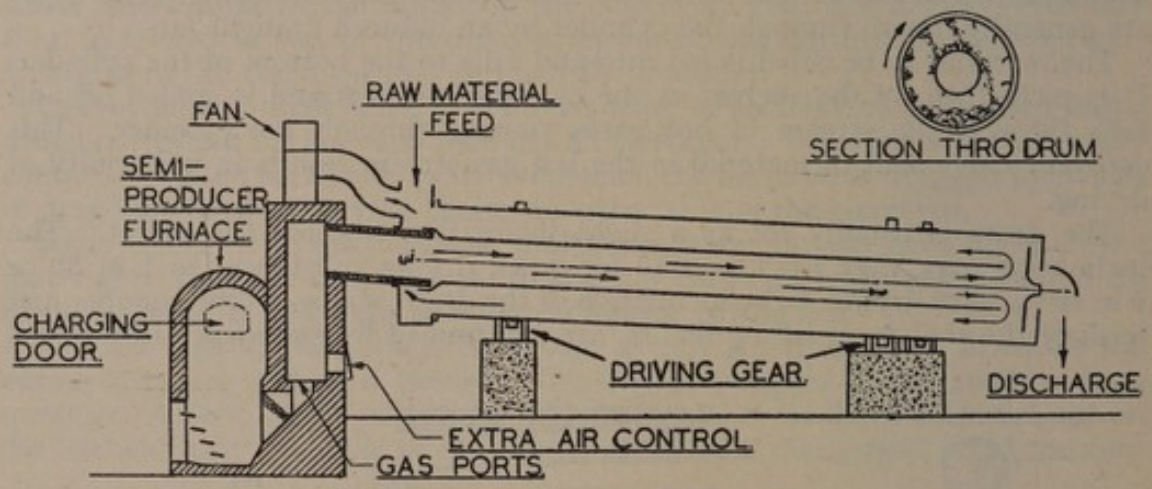


FIG. 233. Double shell rotary drier.

greater part of the heat is imparted to the material through the surface of the inner tube. The material is generally fed into the annular space at the furnace end of the drier. The lifting shelves or flights pick up the material and shower it on to the shelves on the hot inner tube. It is carried round about half a revolution on these shelves, during which time it is heated by contact with the hot surface of the inner drum and is then showered off into the outer shell and the operation is repeated over and over again, as the material works its way to the discharge end of the drier.

In another type of double shell drier, the material (usually light material) is fed into the hot end of the inner tube and passes through the inner tube and back through the annular space, between the outer shell and inner tube.

Double shell driers generally have a much higher thermal efficiency than simple shell rotary driers.

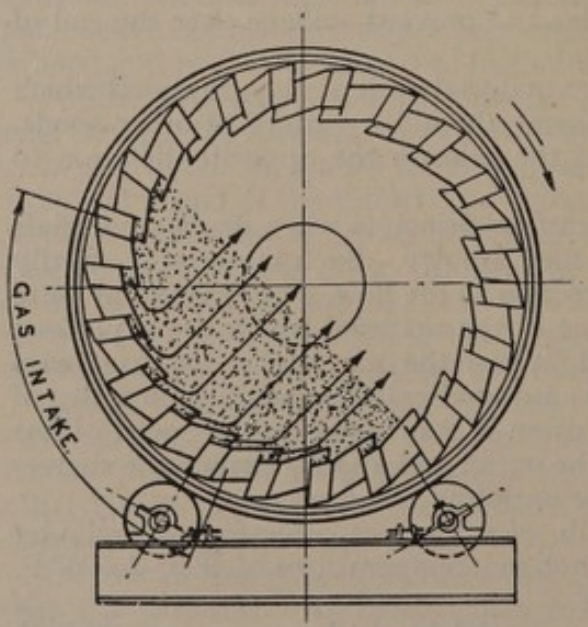


FIG. 234. Louvre rotary drier.

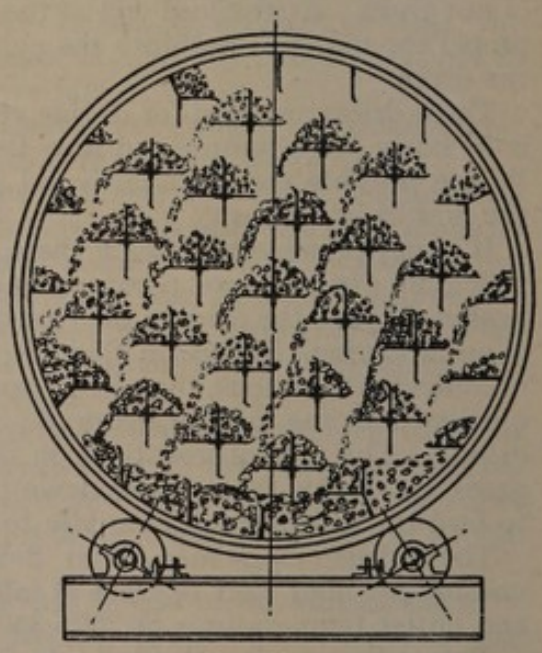


FIG. 235. Cruciform rotary drier.

These driers evaporate 4½ lb. of water per cubic foot of cylinder volume per hour at inlet gas temperatures of 1,000°–1,500° F. The gases leave the inner tube at approximately 400°–500° F., and are discharged from the drier at about 150° F.

Special Types of Rotary Driers. There are various proprietary types of high efficiency rotary driers. One type consists of a horizontal drum with a series of internal channels near the circumference into which hot gases are admitted from a fan. The hot air can pass from these channels through louvres to the

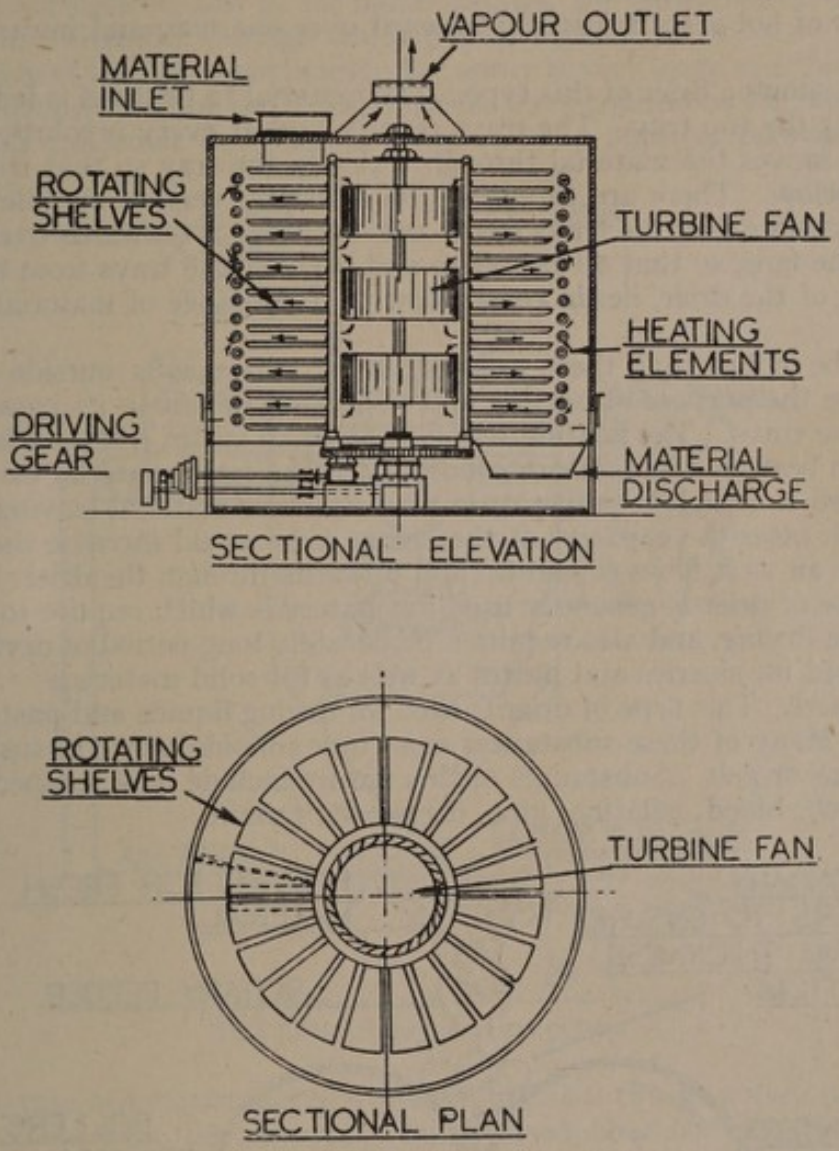


Fig. 236. Vertical cylindrical drier.

inside of the drum, the louvres being so shaped that the material inside the drum cannot spill back through them (Fig. 234).

This drier generally operates at inlet hot gas temperatures of 750°–1,000° F. The material to be dried is fed into the drum, mounts one side of the drum and rolls and tumbles over as it reaches its angle of repose. The hot drying gases are forced through the material as it tumbles over and over.

Another type consists of a horizontal drum with a large number of cross-shaped shelves (Fig. 235). The material to be dried is continually spilling from one shelf to another and is turned over four times in each revolution of the drum, thus ensuring uniformity of drying. In these types of proprietary drier the continuous and uniform tumbling over of the material ensures uniformity

of drying and these driers have an evaporative capacity of 6–12 lb. of water per cubic foot of cylinder volume per hour.

Vertical Cylindrical Driers. Inside a vertical cylindrical casing are a number of concentric rings or shelves. The material to be dried is fed on to the top shelf and is pushed round and turned over at the same time by a revolving rake or scraper. When the material has moved round once, it falls through a gap in the concentric shelf on to the shelf below. Alternatively, the concentric shelves are fixed to a central spindle and revolve and the material is turned over by a fixed rake.

The flow of hot gases is usually outward over one tray and inward over the next.

Fig. 236 shows a drier of this type. The material to be dried is fed on to and spread over the top tray. The trays revolve and at every revolution the rake or scraper moves the material through a slot in the tray so that it falls on to the tray below. There are three fans on a central vertical spindle and these draw air inwards over the trays below and discharge it outwards over the trays opposite the fans, so that air passes to and fro over the trays from the bottom to the top of the drier, i.e. in contra-flow to the passage of material down the drier.

It will be noted that there are concentric steam coils outside the trays. These serve the purpose of heating and re-heating the air in its passage to and fro over the trays. The heating capacity of these steam coils can be adjusted to give the best air temperature conditions. The fresh material can generally be subjected to a higher temperature than the drier material leaving the drier, and in such cases the capacity of the steam coils would increase the temperature of the air as it flows to and fro and upwards through the drier.

This type of drier is generally used for materials which require to be turned over during drying, and also require a moderately long period of drying. They are also used for slurries and pastes as well as for solid materials.

Film Driers. This type of drier is used for drying liquids and pasty or pulpy material. Many of these substances exist in a colloidal state in suspension or as emulsions or gels. Substances of this nature include milk, pulped potatoes, yeast, starch, blood, gelatine, glue, chemicals, tannin.

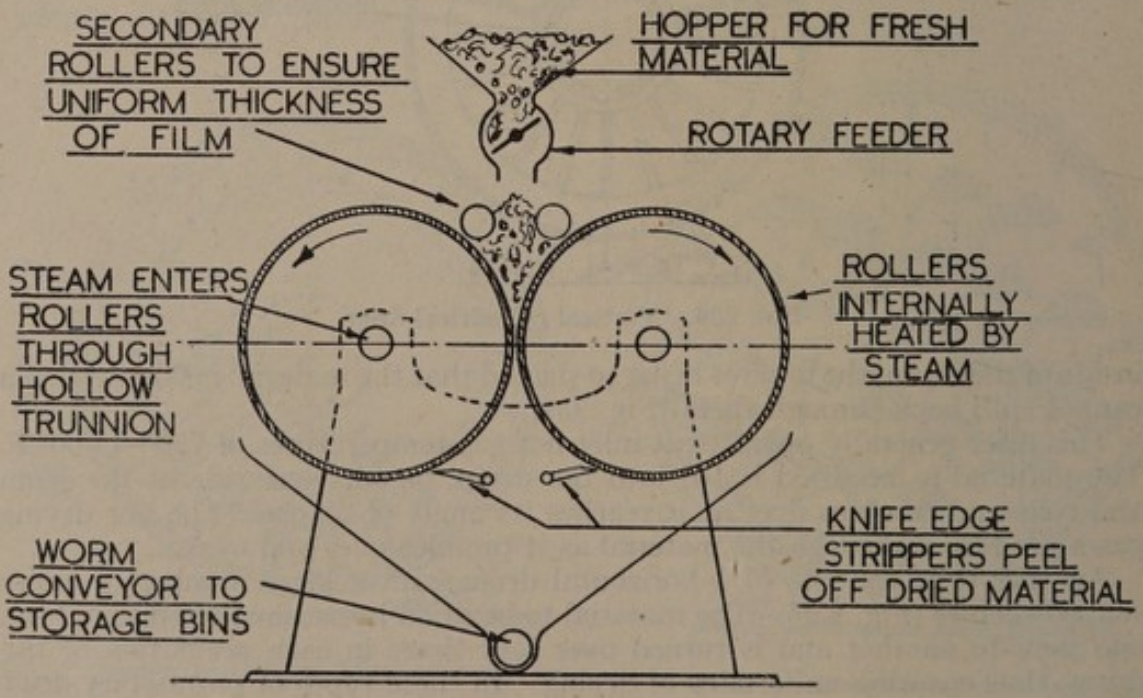


FIG. 237. Film drier.

The material is fed or extruded on to a steam-heated revolving cylinder. Drying is completed in one-half or three-fourths of a revolution of the steam cylinder, and the material is then stripped off in the form of a thin sheet or breaks up into flakes or powder. For satisfactory operation it is essential that a film of uniform thickness be spread on to the steam-heated drum. It is, therefore, accurately machined and ground and a roller feeding arrangement may be fitted. Film driers are made either with single or twin rollers (Fig. 237).

Spray Drier. This is an alternative to the film drier for drying liquid and semi-liquid substances, and as its name implies, the substance to be dried is sprayed into a chamber through which pass hot gases.

The surface of the many particles in the spray is very large and this, together with the movement of the particles, provides ideal conditions for rapid drying. Drying gases can enter the drier at comparatively high temperatures, as the

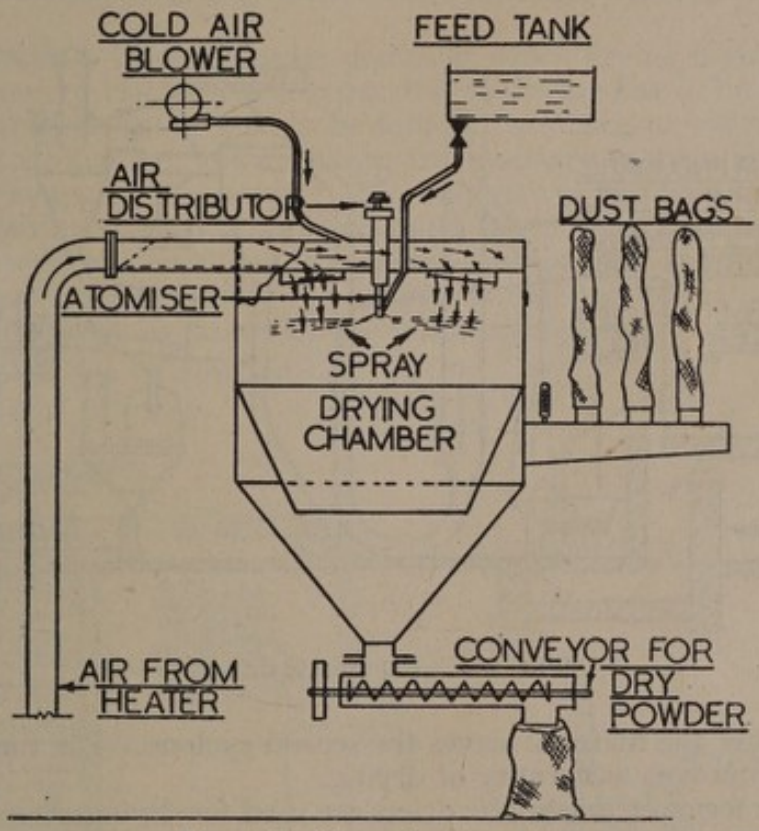


FIG. 238. Spray drying system.

very quick rate of evaporation and heat absorption causes a very rapid fall in temperature, so that the substance being dried does not rise to a harmful temperature.

The heavier dried particles fall to the bottom of the chamber, while the lighter particles are carried over in the exhaust gases and are collected in a dust collector, generally of the filter type.

To obtain satisfactory results with spray driers, it is essential that the substance be sprayed into globules of more or less uniform size, otherwise drying will not be uniform. To prevent drying before the atomised particles are sufficiently dispersed, a cold air duct surrounds the spray or atomiser itself. This type of drier is shown in Fig. 238.

Spray driers are used for drying milk, eggs, meat and vegetable extracts, and other foodstuffs and a great variety of chemicals.

Pneumatic Driers. This is a type of drier used for drying chemicals and other materials in small pieces of uniform size. Basically, this is the simplest of any

type of drier. The main element is a vertical tube 30 feet or more in height. Hot gases flow upwards through the tube and the material to be dried is fed in at the bottom. The upward velocity of the hot gases is such that the fresh, moist, heavy material remains almost suspended, whereas the dry material, being lighter, is carried up more rapidly to the discharge outlet.

A drier of this type is shown in Fig. 239. The hot gases from the furnace are induced upwards through the first vertical pneumatic tube, flow down through a second tube and then up through a third to two cyclones with a fan between them. Any undried and therefore heavy material entering the first cyclone falls to the bottom through a discharge pipe and is picked up by a fan and re-circulated through the third pneumatic uptake tube and so re-dried. From the second cyclone the hot drying exhaust gases are discharged upwards and the dried material is separated and discharged for bagging.

Some pneumatic driers for certain materials incorporate a rotary drier for

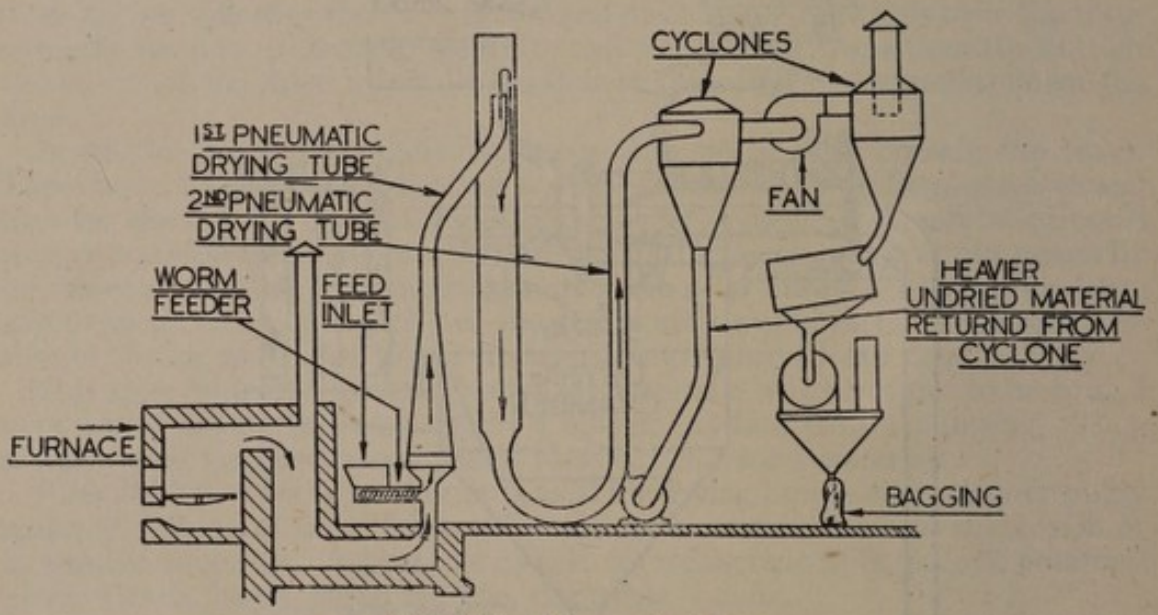


FIG. 239. Pneumatic drier.

final drying after the material leaves the second cyclone. The tumbling in the rotary drier improves uniformity of drying.

The simpler forms of pneumatic driers are used for drying chemicals and the type of drier described is used on the Continent for drying chaffed dried grass for cattle feeding and for drying coal slurry.

Air-swept Rotary Mills. Some materials require to be pulverised and dried, and these two processes can be carried out in one operation.

An example is the preparation of agricultural lime or whiting from carbonate of lime. Chalk or carbonate of lime as quarried is partly in lumps 3 inches or more in size and also contains fines and powder. Agricultural lime can be dried in various types of driers and the large lumps are broken down in the drying process, but air-swept mills are sometimes used.

Another example is the production of pulverised fuel, where it is necessary for the fuel to be pulverised and dried simultaneously.

Fig. 240 shows an arrangement of an air-swept rotary ball mill, which pulverises the material while a continuous stream of hot gases passes through the mill and dries and carries away the pulverised material. The hot gases are drawn through the mill by an induced draught fan and the powdered or pulverised material is separated from the hot gases in a cyclone or other type of separating device.

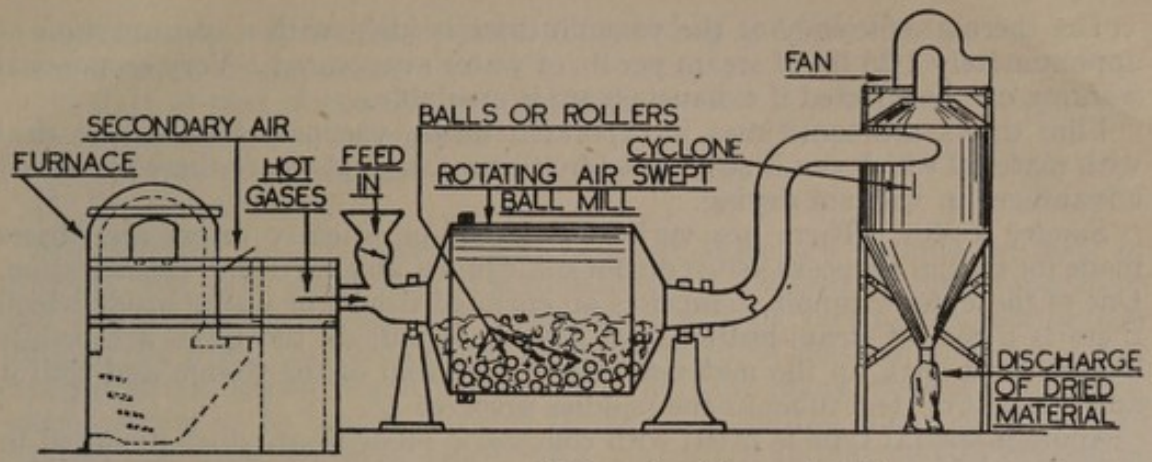


FIG. 240. Air-swept mill for pulverising and drying.

Vacuum Driers. These comprise plants in which drying is carried out in a vacuum. Driers of this type are expensive, as drying has to be carried out in vessels or chambers which have to be sufficiently strong to withstand external pressure, and a condenser and air pump are necessary to maintain the vacuum and draw off evaporated moisture. The majority of vacuum driers are batch driers, as a continuous feed drier necessitates the incorporation of a seal device to prevent loss of vacuum when the material enters and leaves the drier.

The great advantage of this type of drier is that the boiling point of water is very much reduced, as shown by the following figures :—

Vacuum—inches of mercury	26	27	28	29
Boiling point of water ..	125° F.	115° F.	101° F.	79° F.

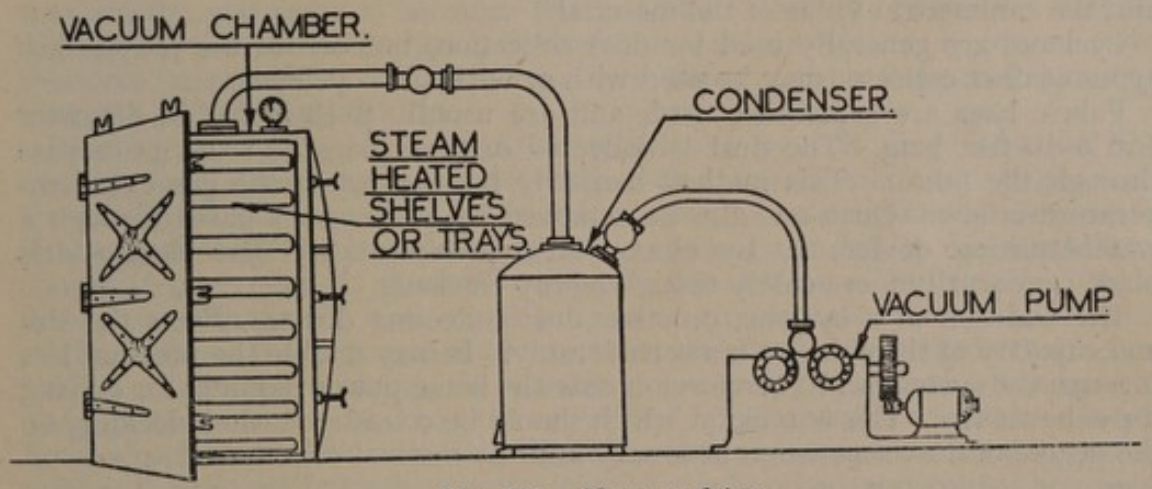


FIG. 241. Vacuum drier.

When drying in a vacuum, a high rate of evaporation can be maintained at a low temperature and, therefore, it is the best method of drying materials which would be harmed by drying at higher temperatures, and which are difficult to dry owing to the low rate of diffusion of moisture from the centre to the surface of the material. Sugar, chemicals, dyestuffs, rubber, white lead, foodstuffs, and explosives, are dried in vacuum driers.

The material may be heated before it is put into the drier. Inside the drier the heating is by conduction in contact with the hot metal steam-heated surfaces of the drier and to a lesser extent by radiation. It is sometimes necessary to provide revolving arms or agitators to turn over the material to equalise its temperature, or the material is spread in a thin layer on steam-heated trays in the drier (Fig. 241).

The thermal efficiency of the vacuum drier is high, with a consumption of approximately 1·25 lb. of steam per lb. of water evaporated. Very economical working can be effected if exhaust steam is available.

Film driers are sometimes incorporated inside vacuum chambers to deal with material which can best be dried by this means and where there are added advantages in vacuum drying.

Sundry Types. There are various types of proprietary driers and driers made for special purposes which do not come under any particular classification. One of these, for example, comprises an enclosed trough or casing inside which there is a reel of steam-heated steel tubes and outside the tubes are paddle blades which pick up the material from the bottom of the trough and spill it into the nest of steel tubes as the paddles revolve.

Another special type is fitted with concentric ploughshare discs mounted in a casing which is about half filled with the material to be dried. The drying gases pass through the casing and heat the concentric discs which revolve and plough through the material to be dried. The material is turned over by the ploughshares and heated by conduction from them.

DUST COLLECTION

A considerable amount of dust may be present in the material being dried. This may adhere to the damp surface of the material, but is carried away in the hot gas stream as it becomes dry. Other materials tend to become powdered in the course of drying. In either event the dust is carried through the drier in the hot gas stream and discharged by the fan. To blow dust into the atmosphere in this way is objectionable, and will result in loss. It is generally necessary to collect this dust and the method employed depends principally on the fineness and quantity of the dust, the degree of dust separation required and the commercial value of the material.

Cyclones are generally used for dust collection, but one of the proprietary types of dust collector may be used with or without a cyclone.

Fabric bags are sometimes used, and are usually 8–12 inches in diameter and 6–10 feet long. The dust is collected in these bags and the gases pass through the fabric. This method can only be used when the gas exit temperature is low. Dust can also be removed by passing the gases through a water-washing device, or by electrostatic precipitation; the electrostatic plant is generally more costly though highly efficient.

The addition of a cyclone, or other dust collecting device, affects the size and capacity of the exhaust fan considerably. It may double the pressure loss through the system and, therefore, double the horse-power required for driving the exhaust fan. This is a factor which should be considered when deciding on the degree of dust separation necessary and the method of removal.

POWER REQUIREMENTS

A fan is an integral part of all driers except film driers, and the power required for driving this is proportional to the air volume and pressure loss through the system. The volume is fixed by the temperature and quantity of hot gases required to evaporate the necessary quantity of moisture, but the pressure drop through the system varies with the type of plant. The pressure loss is considerably increased if dust separation is necessary.

Power is also required for driving rotary and conveyor driers.

CONTROL OF DRYING CONDITIONS

It is of primary importance that material be efficiently and uniformly dried and that it is not subject to such conditions as are harmful to its structure and nature.

Conditions for efficient drying are influenced by the following :—

(1) Type of drying plant.
(2) Rate of feed of material through drier.
(3) Temperature of drying gases.
(4) Velocity of drying gases through the material.
(5) Control of humidity.

For different materials and types of plant there are various methods of control. The most usual method is to maintain a constant drying temperature and velocity of drying gases and to control drying by varying the rate of feed through the drier, the rate of feed being adjusted so that the material is discharged at the correct degree of dryness. There are many materials, however, which require to be heated or dried slowly in the initial stages of drying, as rapid drying tends to seal or case-harden the surface and prevents liberation of moisture from the interior. This is avoided by one or other of the following methods of control :—

(1) By reducing hot gas velocity.
(2) By starting to dry at a low temperature and increasing it as drying proceeds.
(3) By re-circulating, or partly re-circulating, the hot gases through the drier in the initial stages of drying, so that the humidity rises and restricts the rate of liberation of moisture.

Examples of Special Conditions. An example of material requiring such methods of drying is timber. If, in the initial stages of drying, the surface moisture is allowed to evaporate too rapidly, the surface structure shrinks and this sets up stresses in the timber which cause it to split and crack.

A similar example is that of leather. Like timber, leather has to be dried very slowly, otherwise it becomes brittle and its desirable properties may be destroyed. On the other hand, if dried too slowly mildew may form. It is, therefore, necessary to control hot gas circulation, temperature and humidity to ensure satisfactory drying.

Another example is the drying of foundry cores. If they are dried too rapidly the surface may be correctly dried, but the interior may still contain too much moisture and may cause cracks. Also, when a casting is being made, if too much moisture is present, steam may be generated in the mould and distort it or render it unsound, and spoil the casting. The usual means of drying cores uniformly and correctly is to re-circulate the hot gases so that the humidity in the drying chamber increases and checks the rate of evaporation in the early stages of drying, the humidity being reduced as drying proceeds.

An instance in which drying is controlled by temperature is that of hops. From experience it has been found that the best results are obtained by maintaining a constant hot gas velocity through the hop bed and controlling temperature at 100°–110° F. at the commencement, increasing to 160°–165° F. at the end of the drying period.

From these examples it will be appreciated that it is hardly possible to generalise on conditions and methods of control covering the very wide range of materials which are dried for commercial purposes and the different types of drying plant employed.

Instruments. It is most desirable to have dial thermometers (and where necessary a humidity recorder), with dials of sufficient size and in such a position that they are in full view and easily seen by the plant operator.

Thermometers should be provided to show the hot gas inlet temperature to the drier and also the exhaust temperature.

Thermographs are frequently used with large plants and where it is considered necessary to keep a record of temperatures.

Automatic Controls. Automatic controls are a means of insurance against fluctuations in temperature or humidity which might result in a whole batch of material being spoiled. For instance, if the drier is fitted with an ordinary hand fired furnace, there may be fluctuation in temperature if the operator fails to fire at regular intervals and does not keep a constant watch on the temperature.

If a drier is fitted with burners or automatic fuel feed to the furnace, which will maintain a constant heat output without attention, automatic control may be a refinement only necessary if valuable material is being dried which is susceptible to, or easily spoilt by, small variations in drying conditions.

Thermostatic Control (see also Chapter XXVI). The simplest form of thermostat is the direct acting type, and consists of a sensitive portion, or bulb, a small bore flexible tube and a bellows, the whole being filled with a volatile fluid, and hermetically sealed. Changing temperatures round the bulb cause an increase or decrease of pressure in the system and the bellows either expand or contract. The movement of the bellows is used to open and close dampers or fuel valves as necessary to control the temperature.

These direct-acting thermostats are not generally used for control of drying temperatures, as the power they exert is limited and they are not so sensitive as electric thermostats.

Electric thermostats are the most commonly used, and provide efficient means of controlling temperatures to within limits of 1½–3 per cent. The thermostat comprises a special form of switch operated by expansion and contraction of a bi-metal element or fluid in the thermostat ; an indicator is generally provided to vary the temperature setting.

Changes in temperature open and close the switch which operates a motor to open and close valves or dampers of the furnace, or extra air dampers.

Humidity Control. This is very similar in operation to the thermostat except that the switch control is operated by a humidistat in place of the thermostat.

FACTORS AFFECTING THERMAL EFFICIENCY OF DRYING PLANT

Desirable features for maximum theoretical thermal efficiency are as follows :—

(1) Drying gases should be discharged nearly at saturation point.
(2) The temperature range of drying gases between inlet and outlet of drier should be as high as possible.
(3) The flow of material should be in the opposite direction to the flow of drying gases.

It is impracticable to apply all or any of the above principles to the drying of many materials and the limits of efficiency are governed to some extent by the necessity of ensuring that material is uniformly dried and does not lose any desirable properties in the process of drying.

Humidity of Hot Air Discharge. The hot gases leaving the drier should not be discharged to atmosphere much below their saturation point. The degree of saturation at which the gases can be discharged depends to some extent on the material being dried, and the particular design of drier. A drying process can be considered efficient if exhaust gases leave the drier at 80 per cent. saturation.

When commencing to dry material with a high moisture content, hot gases passed once through the substance may be almost saturated and can be discharged to atmosphere without loss of efficiency but when the material is almost dry very little moisture is given off to the drying gases. To obtain the required degree of saturation of the hot gases, it is common practice in some types of drying plant to re-circulate and re-heat the hot gases, sufficient hot gas being discharged and fresh air admitted through controlled dampers to ensure the correct degree of humidity.

In many types of drier the exhaust gas humidity cannot be controlled and is dependent on the design of drier and its suitability for the particular material which is being dried. In such driers, the correct inlet temperature should be maintained and the throughput of material controlled so that it is discharged at the correct degree of dryness.

Re-circulation to avoid exhausting hot gases at low humidity is common practice in chamber and tunnel driers, but is not applicable to rotary driers.

Re-heating. The use of intermediate re-heating of the gases decreases the volume of hot gases required to carry away a definite quantity of moisture from the material.

Re-heating is carried out by passing the hot gases through a steam heater or by mixing with it high temperature gases from a gas burner or from a furnace.

Temperature Range. For high thermal efficiency the fall in temperature of the drying gases through the drier should be as great as possible, but initial temperature is governed by the temperature to which the material can be subjected without damage.

Contra-Flow of Material and Drying Gases. In continuous-flow driers, the hot gases should preferably pass in a contra-flow direction to the flow of material, but this is only possible when dealing with heavy material, as light material would be picked up and carried along in the hot gas stream.

Direct and Indirect Heating. There are two methods of heating air for delivery to driers by fan. The air may either be heated in an air heater, which is sometimes referred to as indirect heating, or the air may be heated by mixing with it the products of combustion from a furnace or burner—a method termed direct heating, as the whole of the heat in the furnace flue gases is directly and effectively used in heating. The hot air is delivered to the drier by a pressure fan, or by an induced draught fan at the drier outlet.

Indirect heating by means of an air heater is much more costly than direct heating as the fitting of an air heater adds 15–20 per cent. to the cost of the drying plant. The general overall thermal efficiency of an air heater is about 55–65 per cent., while that of a furnace alone for direct heating is 86–90 per cent. The quantity and cost of fuel for indirect heating is, therefore, about 50 per cent. higher than that required for direct heating, except where waste steam is available.

It is generally only necessary to employ indirect heating when drying material which would undergo chemical change in the presence of a small quantity of CO_2 or traces of sulphur, and for drying enamel work.

The direct heating system of drying is more widely used, particularly for most of the larger drying plants, and would no doubt be more largely used in preference to indirect drying if it were realised that, provided a suitable furnace and fuel is used, there is no risk of harmful contamination from the small quantity of products of combustion which are present. The principal product of combustion is CO_2, which is beneficial rather than otherwise in the drying of foodstuffs. There are also slight traces of sulphur, dust and tarry matter, but provided suitable fuel and furnace are used, the quantity is so small as to be negligible.

Maintenance of Steady Temperature. The temperature at which materials are dried should be fixed with a margin of safety below the temperature which would be harmful in any way to the material. It is obvious that any upward fluctuation in temperature may be harmful to the material, and falls in temperature may result in material leaving the drier insufficiently dried and will considerably reduce output. For these reasons the maintenance of a steady temperature is of the utmost importance for efficient operation.

The design of burners or furnace should be such that a steady heat output

and temperature are maintained. If necessary thermostatic control should be fitted to ensure a constant temperature.

Efficient Heat Insulation. Some driers are insufficiently lagged. If lagging is removed or damaged loss of efficiency results. Efficient lagging is most desirable, particularly for high temperature driers. In a draughty building the heat losses will be considerably greater than in a building where there is little movement of air.

Air Leakages. Air leakages into driers with exhaust fans and leakage of hot gases from driers under pressure, should be avoided.

Rotary driers usually have a labyrinth gland between the rotating drum and the stationary ends. The drum may slightly distort with heat, and it is, therefore, desirable to examine this from time to time. If adjustable, the labyrinth plates should be set with the minimum clearance at which the rotating drum will run without rubbing.

Uniformity of Drying. If there are damp patches when the bulk of the material is correctly dried, it may be necessary to continue drying merely to dry a small damp part of the material. The additional time taken represents loss in efficiency and output.

THERMAL EFFICIENCY OF DRIERS

A drier is essentially a plant to evaporate moisture and its thermal efficiency is most simply expressed in terms of pounds of water evaporated per pound of fuel, but for any particular type of plant this varies considerably according to the material to be dried. Some materials might be termed easy to dry, i.e. materials having such a structure and form that under the influence of heat, moisture is freely transferred from the centre to the surface ; a higher rate of evaporation per pound of fuel is then obtained than when drying other materials.

The heat losses and effective heat utilised in various types of drying plant are of the following order :—

	Per cent.
Sensible heat used in warming material	3–25
Heat lost in exhaust gases	7–25
Heat lost by radiation and other causes	7–20
Effective heat used in evaporating moisture from the material being dried	40–72

The evaporation of water from the material being dried per pound of fuel for different types of drier is, approximately, as in Table 113.

TABLE 113

Type of drier.	Lb. water evaporated per lb. of fuel.	
	Direct heating.	Indirect heating.
Simple tray driers, without re-circulation ..	3 –5	2 –4
Drying rooms, conveyor and tray driers, with re-circulation	5 –8½	3 –5
Simple rotary driers	3½–6	2 –4
Double shell and certain proprietary rotary driers	5 –8½	2½–5

TECHNICAL DATA

Although some of this information has been given in earlier chapters, for convenience, the technical data specially applicable to drying problems is summarised here.

Weight of Air. The weight of a cubic foot of air at atmospheric pressure and any temperature may be calculated from the following formula (noting that the absolute temperature in °F. absolute is obtained by adding 460 to the scale reading in °F. ; and in °C. absolute by adding 273 to the scale reading in °C.) :—

At temperatures Fahrenheit :—

$$\text{Weight of a cubic foot of air in pounds} = \frac{39{\cdot}819}{\text{°F. absolute}} \quad . \quad . \quad (1)$$

At temperatures centigrade :—

$$\text{Weight of a cubic foot of air in pounds} = \frac{22{\cdot}121}{\text{°C. absolute}} \quad . \quad . \quad (2)$$

The weight of air in pounds per 1,000 cubic feet at normal atmospheric pressure and at temperatures between zero and 1,600° F. or 1,000° C., is shown in Fig. 242.

Volume of Air. The volume of any given weight of air at atmospheric pressure and any given temperature, may be calculated from the following :—

$$\text{Volume cubic feet at temperatures °F.} = \frac{\text{Weight} \times \text{°F. absolute}}{39{\cdot}819} \quad . \quad (3)$$

$$\text{Volume cubic feet at temperature °C.} = \frac{\text{Weight} \times \text{°C. absolute}}{22{\cdot}121} \quad . \quad (4)$$

Change in Volume with Temperature Change. The volume of air varies as the absolute temperature. If V_1 = volume at t_1, volume V_2 at t_2, T_1 and T_2 being the respective temperatures on the absolute scale,

$$V_2 = V_1 T_2/T_1 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (5)$$

Heat Required for Heating Air.

$$\text{Heat required} = W(t_2 - t_1) \times S. \quad . \quad . \quad . \quad . \quad . \quad (6)$$

When W is the weight of air in pounds and S is the specific heat (cf. Chapter V) in B.Th.U./°F./lb., this formula gives the heat required in B.Th.U. to heat a given weight of air from t_1°F. to t_2°F.

Alternatively, the heat required $= V(t_2 - t_1) \times S$ B.Th.U. . . . (7)

When V is the volume of air in cubic feet, S is the specific heat in B.Th.U./cu. ft./°F. and t_1 and t_2 are °F. as before.

Specific heat of air at constant pressure :—

°F.	°C.	B.Th.U./° F./lb.	B.Th.U./° F./cu. ft.
32–212	0–100	0·241	0·0194
32–393	0–200	0·242	0·0195
32–573	0–300	0·244	0·0196

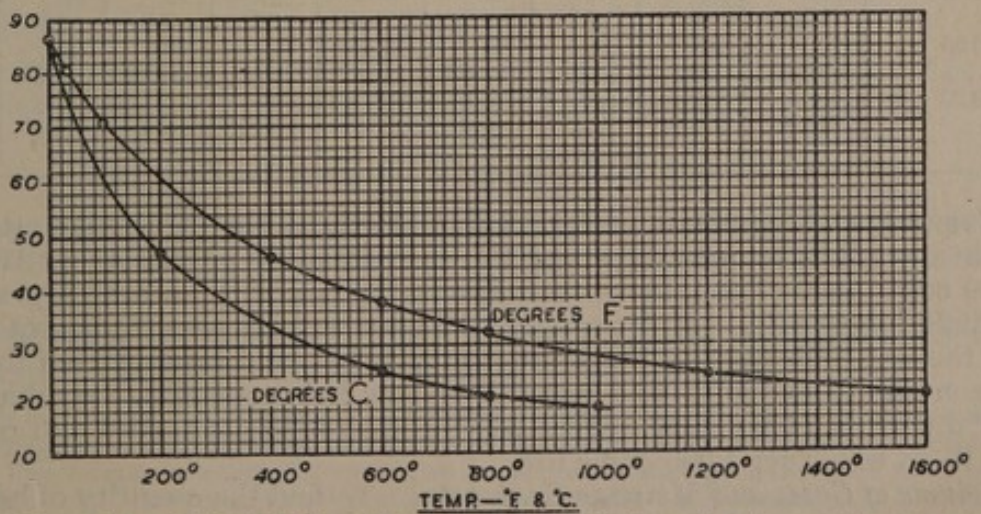


FIG. 242. Weight in lb. of 1,000 cu. ft. of air at various temperatures.

Example. 1,000 cubic feet of air is required at 300° F., the initial air temperature being 60° F.

To find the heat required to heat the air first find the weight of air from formula (1) above or Fig. 242. It will be found to be 52·4 lb. per 1,000 cubic feet, and therefore :—

Heat required = 52·4 (300–60) × 0·242 = 3,043 B.Th.U. per 1,000 cubic feet.

The heat required to raise the temperature of air at normal atmospheric pressure and normal temperature of 60° F. to any temperature up to 1,600° F. or 1,000° C., is shown on Fig. 243. The heat required in B.Th.U. is given per 1,000 cubic feet of air at the temperature to which it is heated.

TABLE 114. PERCENTAGE SATURATION OF AIR AT TEMPERATURE FROM 50° TO 170° F. AND CORRESPONDING POUNDS OF MOISTURE PER 1,000 CUBIC FEET

Air temperature dry bulb reading ° F.	Percentage saturation of air and lb. of moisture per 1,000 cu. ft. of the saturated air measured at dry bulb temperature								
	Depression of wet bulb, ° F.								
	0	2	4	6	8	10	12	14	16
50	100	86	72	59	46	33	21	9	
	0·59	0·5	0·42	0·35	0·27	0·19	0·12	0·05	
55	100	87	75	63	51	40	28	18	7·5
	0·7	0·61	0·52	0·44	0·36	0·28	0·20	0·1	0·05
60	100	88	77	66	55	45	35	25·5	16
	0·83	0·73	0·64	0·54	0·45	0·37	0·29	0·21	0·13
70	100	90	80	71	62	53	45	37	29
	1·14	1·03	0·92	0·81	0·71	0·6	0·51	0·42	0·33
80	100	91	83	75	67	59	52	45	38
	1·57	1·42	1·3	1·17	1·05	0·93	0·81	0·71	0·6
90	100	92	84	77	70	63·5	57	51	45
	2·15	1·98	1·8	1·66	1·5	1·36	1·22	1·10	0·97
100	100	93	86	79	73	67	61	55	50
	2·85	2·65	2·44	2·26	2·07	1·94	1·74	1·57	1·42
110	100	93·5	87	81	75	69·5	64	59	54
	3·79	3·54	3·3	3·08	2·85	2·62	2·43	2·22	2·04
120	100	93·5	88	82	76	71	66·5	62	57
	4·85	4·54	4·28	4·0	3·7	3·44	3·23	3·0	2·77
130	100	94	89	83	78	73	68·5	64	60
	6·28	5·91	5·61	5·22	4·9	4·59	4·3	4·02	3·7
140	100	94·5	89	84	79	75	70	66	62
	8·1	7·63	7·19	6·8	6·4	6·06	5·66	5·33	5·02
150	100	95	90	85	80·5	76	72	68	64
	10·1	9·6	9·1	8·6	8·2	7·7	7·3	6·9	6·5
160	100	95	90·5	86	81·5	77	73	69	66
	13·0	12·2	11·6	11·06	10·47	9·88	9·39	8·9	8·7
170	100	95	91	86·5	82	78	74	70	67
	15·8	15·14	14·7	13·8	13·07	12·44	11·8	11·2	10·7

Humidity and Saturation of Air. Table 114 shows the percentage saturation of air at various temperatures and the corresponding pounds of moisture per 1,000 cubic feet. The table is intended to be used with an ordinary wet and dry bulb hygrometer. If, for example, the dry bulb shows a reading of 120° F. and the wet bulb 110° F., the depression is 10° F., and reading from the table horizontally from 120° F. and vertically below 10° depression, the table shows that the air is 71 per cent. saturated, and that it has absorbed and contains 3·46 lb. of water per 1,000 cubic feet.

Volume of Gases and Moisture Evaporated. To find the quantity of hot gases required to be passed through a drier to evaporate a given quantity of water,

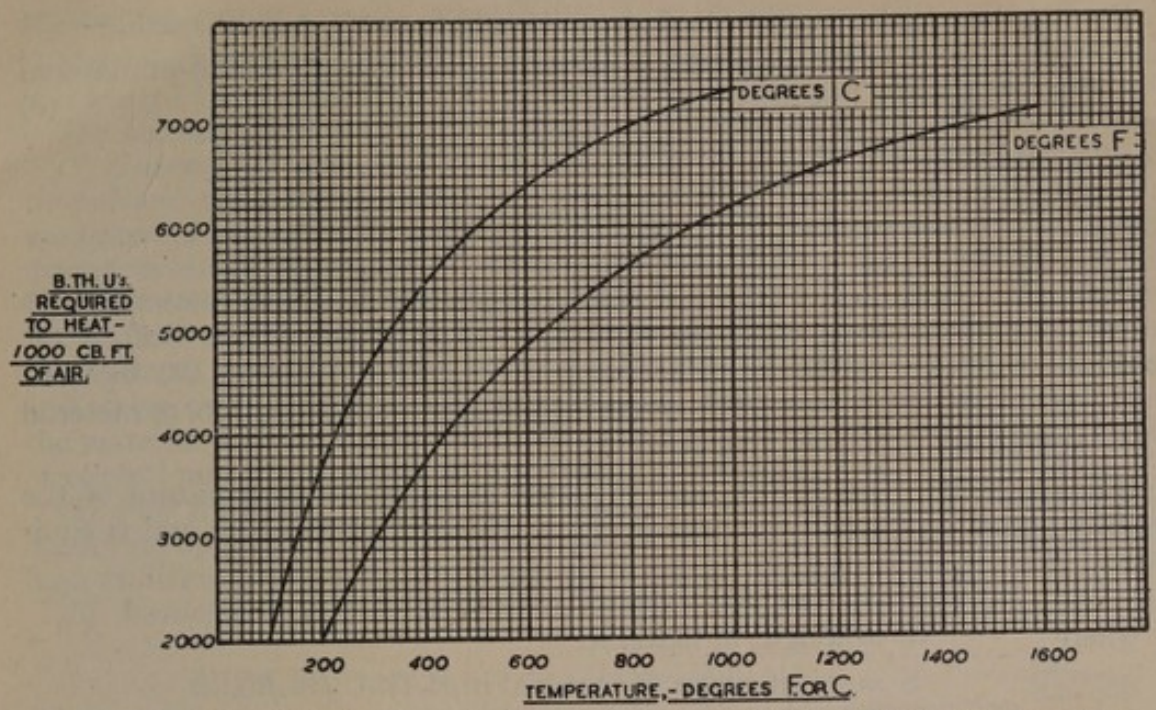


FIG. 243. Heat in B.Th.U. required to heat air from 60° F. or 16° C. the air volume being taken at the hot temperature given by the abscissæ.

it is necessary to know the initial and final temperature and humidity of the gases.

Under normal atmospheric conditions in this country it may be assumed that air before heating is at 50° F. or 60° F. and 40 per cent. saturation or relative humidity ; the quantity of moisture it carries in pounds per 1,000 cubic feet can be found from Table 114.

A drying process may be assumed to be efficient if the exhaust gases leave the drier at a saturation of 80 per cent.

Example. To find moisture evaporated per 1,000 cubic feet of exhaust gases leaving a drier at 130° F. and 80 per cent. saturation, if the atmospheric air is at 60° F. and 40 per cent. saturation.

From Table 114 it will be found that at 130° F. and 80 per cent. saturation the hot gases contain 5·028 lb. of moisture per 1,000 cubic feet.

The volume of these gases entering the drier furnace or heater at 60° F. will be :—

$$1{,}000 \times \frac{(460 + 60)}{(460 + 130)} = 881 \text{ cubic feet,}$$

and at 60° F. and 40 per cent. saturation it will be seen from Table 114 that the hot gases will contain 0·332 lb. of moisture per 1,000 cubic feet, or

$$\frac{881}{1{,}000} \times 0{\cdot}332 = 0{\cdot}29 \text{ lb. per 881 cubic feet.}$$

The quantity of water evaporated, therefore, will be 5·024 − 0·29 = 4·73 lb. per 1,000 cubic feet of exhaust gases leaving the drier at 130° F. and 80 per cent. saturation.*

Quantity of Water Evaporated. When drying it is not usual to evaporate all the moisture from the material for reasons previously given.

* This method of calculation is not strictly accurate since the gases which are 80 per cent. saturated at 130° F. would lose part of their moisture by condensation if cooled to 60° F. The volume entering, 881 cu. ft., as calculated by this method is thus too large by about 50 cu. ft. Since the water content of the entering gases is small, the error is not generally serious.

To find the moisture evaporated the following formula may be used :—

$$\frac{bc}{100a} - \frac{d}{100} = \begin{cases}\text{Ratio of weight of water evaporated to weight of material} \\ \text{as dried.} \quad . \quad . \quad .

METHODS OF HEATING—FURNACES AND FUELS

Nearly all methods of drying involve the supply of hot air or air mixed with and heated by the products of combustion from a burner or furnace.

The factors which decide the best method of heating are as follow :—

(1) *Convenience and Ease of Control.* This assumes that the required heat output and temperature can be correctly and continuously maintained with a minimum of labour and attention. The continuous maintenance of the correct drying temperature is of vital importance as it ensures that material is dried under the best conditions of temperature and is not damaged or spoiled by fluctuations in temperature. The cost of fuel per ton of material dried is roughly proportional to the quantity of moisture which has to be evaporated from it and where only a relatively small quantity of water has to be evaporated from the material, or where valuable products are being dried, convenience and ease of control may to some extent over-ride fuel efficiency.

(2) *Fuel Efficiency.* High thermal efficiency and low fuel cost is desirable in all drying operations, but particularly so where the material being dried has a high moisture content and the value of the dried product is not high.

(3) *Availability of Fuel.* The above factors (1) and (2) are obviously governed

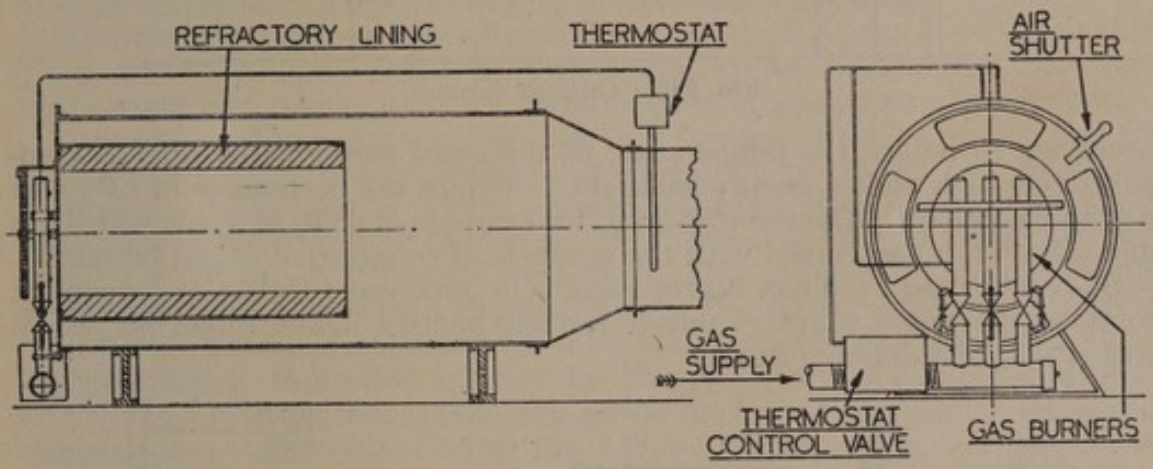


FIG. 244. Gas-fired direct air heater.

to a considerable extent by the availability and relative costs of various fuels at the site of the drying plant.

The uses of various types of heating media and their characteristics are as follow :—

DIRECT HEATING

This, as already described, is the most efficient method of providing hot air and gases for drying. Air is heated by mixing with it the products of combustion from gas or oil burners, or from a solid fuel furnace.

Gas, oil or solid fuels can be used for direct drying.

Town gas is a very convenient form of fuel to use for direct heating. Reliable and efficient burners, and thermostatic control gear are obtainable, which ensure the maintenance of a steady temperature without attention. It is clean, free of dust, and the amount of sulphur is limited by statute to traces.

Fig. 244 shows a good type of direct fired gas heater with thermostatic control. The heater comprises a cylindrical steel casing with an inner refractory lined casing extending about half the length of the heater. Multiple gas burners fire into the inner brick-lined casing and air is drawn through the inner casing. Air is also drawn through the annular space between the inner and outer casings, and this air can be controlled by a hand operated shutter. The gas thermostat is fitted in the hot gas stream leaving the heater and is usually composed of an

element which, expanding and contracting with changes of temperature, opens and closes a gas relay valve. The relay valve controls a valve in the pipe supplying gas to the burners.

The presence of CO_2 is generally harmless, except in the drying of certain chemicals and for some enamel drying ovens. The quantity of diluting air

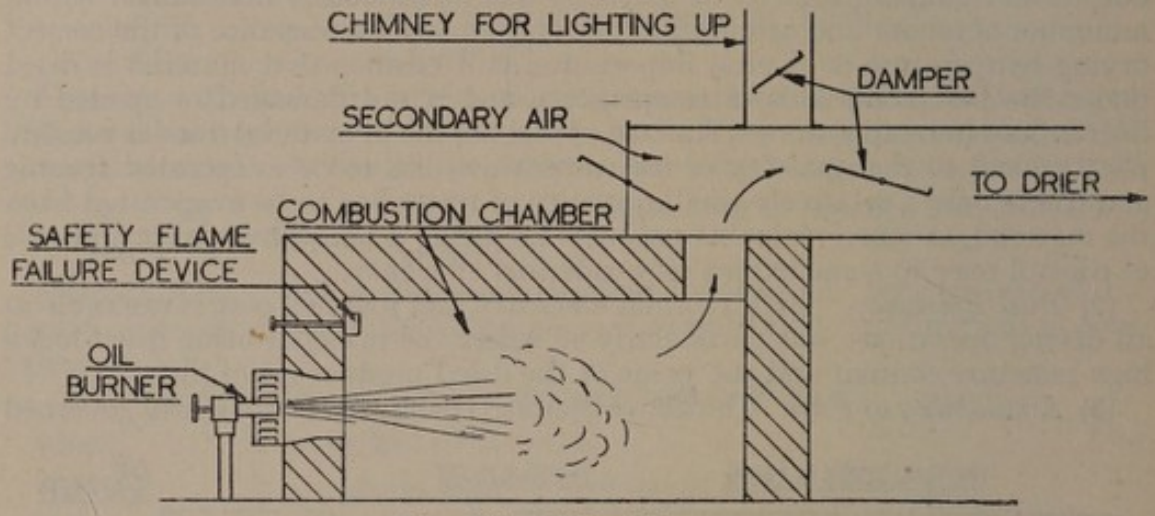


FIG. 245. Oil-fired furnace.

necessary to reduce the temperature of the gases entering the drier is comparatively high and is usually sufficient to reduce the percentage of CO_2 to a negligible amount. Domestic cooking for example is done in ovens which are direct heated by air mixed with the products of combustion of gas burners.

Oil can be used for direct heating, and is in some ways almost as convenient to use as gas, but it is inadvisable to use any but the lighter diesel oils, which will give a clean burning flame.

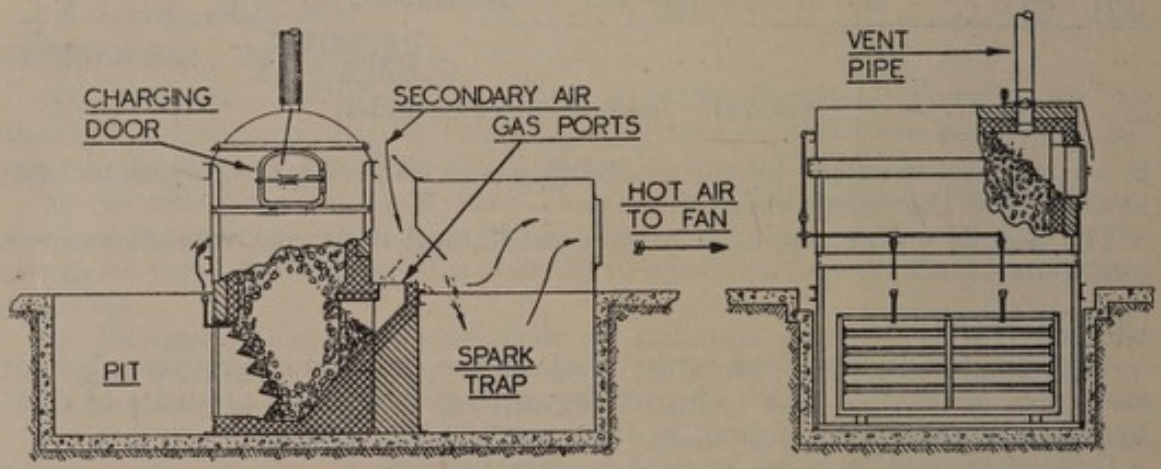


FIG. 246. Semi-producer furnace.

The combustion chamber must be of adequate size to ensure complete combustion and provision should always be made for a safety device to cut off the oil supply in the event of flame failure. This will prevent the possibility of unburned oil coming in contact with and ruining a batch of material in the drier.

A further precaution which is recommended is the provision of a flue to atmosphere, so that the whole of the products of combustion can be by-passed from the drying plant until the furnace is thoroughly warmed, and also to prevent the risk of partially burned products passing into the material under treatment.

Oil burners should be regularly examined and cleaned, as a burner becoming faulty or dirty may give off sufficient smoke, due to improper combustion, to spoil a considerable quantity of material in the drier before it is noticed.

The use of heavier oils, such as creosote pitch, is permissible as an emergency measure for drying materials other than edible products, but provision must be made for preheating the oil or pitch, otherwise it is impossible to guarantee a continuous flow of liquid fuel from the storage tanks. Also the best possible burner equipment must be used to ensure smokeless combustion. This subject is discussed in Chapters II and XXVIII.

Fig. 245 shows an oil-fired furnace comprising a firebrick-lined combustion chamber of ample size to ensure complete combustion. It also shows the chimney for by-passing gases when lighting up, the flame failure device, and the damper for admitting cold air and regulating the temperature of hot gas inlet to the drier.

The essentially suitable solid fuels for direct heating are those of low volatile content, namely, coke with less than 3 per cent. volatile, and anthracite with less than 5 per cent. volatile. These fuels give off a clean smokeless gas with such small traces of tarry or volatile matter that when diluted with air they have generally no deleterious effect on the product being dried.

Coke-burning furnaces specially designed for drying plants are obtainable. These are of the hopper feed type, with continuous fuel feed, so that a constant heat output and temperature are maintained with a minimum of attention.

The semi-producer furnace, Fig. 246, is a furnace of this type designed for drying plants. It is somewhat similar to step grate producers, but the firebed is thinner, so that there is always a flame at the gas ports, which ensures that inflammable and unignited gas cannot pass over into the drier such as could occur if a full producer furnace were used. This furnace has step grate bars with fuel hopper above with charging door. The flame from the gas ports is mixed with incoming secondary air from the damper shown above the ports. Temperature is controlled entirely by this damper: opening the damper admits more cold air, reduces the draught on the furnace and lowers the drying temperature; *vice versa*, closing the damper increases the temperature as required. A spark trap is shown between the furnace and drier, but is often not necessary.

Where drying temperatures higher than 500° F. have to be maintained, the flue or duct between the furnace and the fan is built in brickwork, or preferably insulating bricks.

The only attention these furnaces require is recharging every four hours and the cleaning of fires every eight or twelve hours. The furnace can be banked for the night and started up at full ouput in four or five minutes in the morning.

A suitable type of coke furnace is the gravity feed furnace, which comprises a firebrick-lined combustion chamber with a sloping grate. Fuel feeds by gravity from a hopper on to the sloping grate and maintains a constant depth and consistency of fuel on the grate which ensures steady heat output and temperature (see Fig. 223, Chapter XXIV).

Solid fuels with higher volatile content than coke and anthracite are not fundamentally so suited for direct drying, but can be used if precautions are taken to ensure that the volatile matter which is given off is burned in the furnace. A mechanical stoker must be used, and a high rate of combustion and large combustion chambers will assist in the combustion of volatile matter.

Medium-volatile fuels can be used for direct drying, provided adequate precautions are taken to ensure smokeless combustion and so avoid spoiling the product which is being dried by smoke or tarry volatile products given off from the fuel.

Hand fired furnaces cannot be used. The underfeed screw type of stoker is

fully satisfactory for small furnaces burning up to 2 or 3 cwt. per hour and chain-grate stokers for the larger furnaces (see Fig. 247).

FURNACES

To use medium volatile fuels for direct drying and to ensure that combustion is as complete as possible, special consideration must be given to the following points :—

(1) *High Furnace Temperature.* A comparatively high rate of combustion should be maintained to ensure a high furnace temperature, because hydrocarbon and tar vapours given off from the fuel cease to burn at temperatures below 800° C. (or 1,470° F.). If, therefore, the temperature above any part of the fuel bed falls below this temperature, carbonaceous matter and vapour will leave the furnace in the form of smoke and contaminate the material being dried. From this it will be obvious why hand fired furnaces are not suitable as when fresh coal is hand fired on to the top of the firebed the temperature above the fresh coal is bound to fall, for a short time, below the temperature

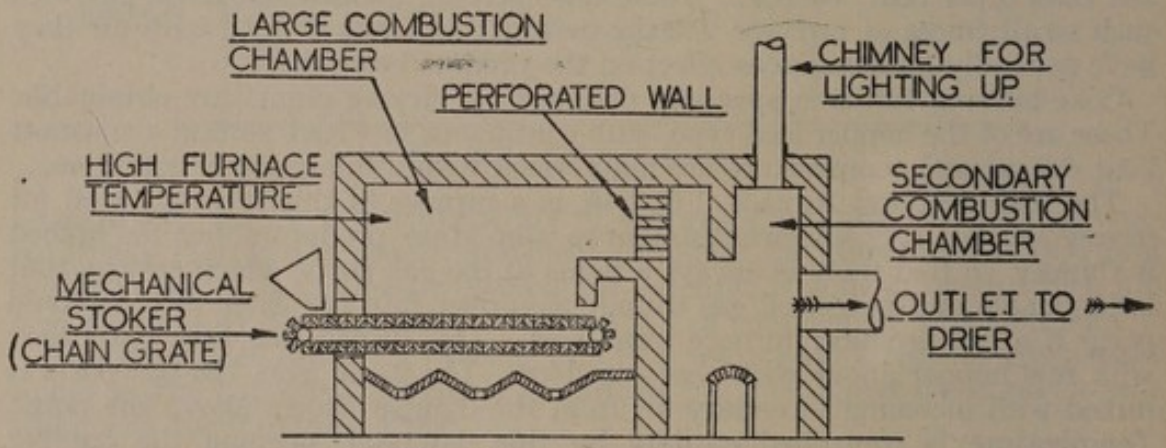


FIG. 247. Furnace for medium volatile fuel.

at which the carbonaceous vapour will ignite, and therefore smoke will be given off. Opening the fire-door is likely to give rise to similar trouble.

(2) *Ample Size of Combustion Chamber.* The combustion chamber must be of sufficient size to ensure that complete combustion of volatiles takes place. The velocity of the gases should be sufficiently low for combustion to be complete in the combustion chamber.

(3) *Firebars.* Firebars must have adequate width of air spaces between the bars to give the full amount of air required for complete combustion.

(4) *Mixing of Air with Carbonaceous Vapours.* It is essential that the gases leaving the firebed be intimately mixed with the correct quantity of air to ensure complete combustion.

A secondary combustion chamber is an advantage, as the gases are mixed in the restricted passage between the two chambers and combustion is completed in the second one. The use of a brick wall with chequered holes, between the combustion chambers, assists in ensuring complete combustion as the gases pass through the holes at high velocity and in contact with the hot brickwork.

Spark or Grit Arresters. When direct drying some materials it is necessary to ensure that no grit or spark reaches the material being dried, firstly because a hot spark may result in a fire in the drier if the material being dried is inflammable and, secondly, because it may be undesirable that there should be grit in the material to be dried.

Sparks carried over from low-volatile fuels, such as coke or anthracite, will cool in a second or less, but those from high-volatile fuels may remain hot

enough for four or five seconds to ignite material being dried, as the volatile matter in the spark continues to burn while it is carried over in the air stream. Therefore, adequate precautions must be taken to trap sparks when using high-volatile fuel or steps taken to ensure that their length of travel is such that they are cool before reaching the material.

Grit arresters may be divided into the following types :—

(1) Gravity or inertial separators.
(2) Cyclone separators.
(3) Filters.
(4) Mechanically driven rotary separators.
(5) Electrostatic precipitators.

Cyclones, to be effective, impose a resistance to the gas flow of upwards of 1½ inch w.g. Filters require to be of metal to stand the temperature of hot gases, and are liable to corrode. Electrostatic precipitators are expensive, and the same applies, to a less extent, to rotary separators.

For these reasons gravity or inertial separators, as shown in Fig. 246, are generally used ; the hot gases are deflected downwards into a spark trap or chamber, which should be of such a size that the upward velocity of the gases to the outlet to drier should not exceed 300 feet per minute ; lower speeds are desirable, but space does not generally permit of larger spark chambers being used.

INDIRECT HEATING

Indirect heating implies the use for drying of air heated by passage through an air heater.

Steam Air Heater. Where waste steam is available in sufficient quantity and at sufficient pressure to give a reasonably hot air inlet temperature to the drier, this form of heating is perhaps the most economical to adopt for drying.

With waste steam at pressures up to 40 lb. per square inch, air can be heated to 200° F. or 250° F. This is sufficient to dry many materials. If an adequate supply of live steam is available above 100 lb. pressure, air can be heated to 300° F.

There is little or nothing to be gained by using superheated steam for air heaters as 100° or 200° F. superheat adds comparatively little to the total volume of heat in the steam and the heat transmission through the heater surfaces is very low until the steam falls to saturation temperature (see Chapter XXIII).

The advantages of steam heating are that there is no necessity to employ elaborate temperature controlling equipment as, provided a steady pressure is maintained and the plant is not situated too far from the boiler, a constant air temperature will be maintained with very little attention.

If, however, the limited air temperature which can be obtained with a steam air heater is very much lower than that at which the material can be economically and efficiently dried, the use of steam air heaters is precluded or the use of a much larger drying plant is involved with consequent increase in capital cost.

Generally the use of steam air heaters is only warranted if an adequate supply of steam is available from existing boilers. The installation of new steam generating plant adds considerably to the capital and operating cost of a drying plant.

Furnace-fired Air Heaters. Where hot air is essential for drying, and steam is not available or the cost of installing steam boilers for the purpose is considered to be too high, air heaters can be used which are heated in a gas, oil or solid fuel furnace. The design of the air heater should be such that the

possibility of overheating the tube elements nearest the fire is reduced to a minimum. Overheating is generally avoided by making the tubes nearest the fire of heat-resisting metal, using these tubes as a first pass for cold air at a relatively high velocity and ribbing the tubes to increase heat transmission.

The usual types of industrial furnaces for various fuels can be used for air heaters, provided precautions are taken to avoid overheating the air heater tube elements exposed to direct radiation from the fire. This necessitates limiting the rate of combustion and preferably using a hopper fuel feed or mechanical stoker.

It may sometimes be possible to utilise waste heat from boilers or furnaces for heating the air.

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CHAPTER XXVI

THERMOSTATIC CONTROL IN STEAM AND HOT WATER PRACTICE

The importance of automatic control—Control in central heating—Hot water supplies—Process heating—Recovery of waste heat from internal combustion engines—Steam boilers—Control equipment—General technique.

FUEL economy in steam practice is too often regarded as concerned with the boiler house only, but frequently there is greater waste in utilisation than in production of steam. This theme is elaborated in Chapter XXIII.

Every degree of overheat in a space heating system, a hot water supply installation, or a manufacturing process, represents waste of heat units and consequently of the corresponding weight of fuel which was burned to generate them. The amount of fuel wasted depends on the efficiency of the boiler plant and the extent to which losses in getting the heat to the point of consumption have been eliminated; but maximum economy cannot be achieved unless temperatures at the point of usage are precisely controlled.

Improving the efficiency of combustion in the heat-generating plant or giving attention to such matters as the lagging of pipes and leaky joints, results in more heat being available at the point of consumption—perhaps more than is required. It is here that thermostatic control can help by preventing overheating and allowing just that amount of heat to be consumed as will maintain the correct temperatures.

Hand regulation of the inlet valves can never be fully efficient as a means of temperature control, because (and this is only one of the reasons) adjustment is made only after the rise in temperature has taken place. The valve must be regulated gradually and continuously as the fluctuations in temperature develop and as quickly as they can be indicated on a thermometer. To do so accurately by hand is quite impossible; such regulation can only be effected by automatic temperature control, the active element of which is the thermostat.

With manual control, temperature changes are sensed and indicated by a thermometer. They are conveyed to the attendant through his eyes (if he happens to be in the right spot at the right moment). He then operates the valve to counteract the change. He can only guess the amount of adjustment required and wait until the thermometer shows him whether or not his guess is reasonably right. There is always an inevitable and wasteful delay between the change and the action taken to counteract it, and the action taken may be inadequate.

A thermostat combines the functions of the thermometer and the attendant. Not only does it sense the temperature change before it would be perceptible to the attendant, but a compensatory movement takes place immediately which acts on the regulating valve. There is no period of delay and no guesswork as to the amount of valve movement required.

CENTRAL HEATING

In a space-heating system the demand for heat fluctuates according to the weather conditions, the heat given off by the manufacturing plant or process or by the occupants, and by the lighting system—this last being an important factor in establishments working twenty-four hours a day, often with permanent black-out. When such rooms are not equipped with thermostatic control, overheating is counteracted by opening windows and ventilators, with a consequent loss of valuable heat units (cf. Chapter XXIV). When the demand is low, overheating can only be completely avoided by placing the heating

system under thermostatic control. A thermostat placed in a suitable position within the space and arranged to operate a valve on the pipeline feeding the system will not only regulate the supply to the demand, but will afford the additional advantage of comfortable and healthy working conditions at all times.

Zone Control. Large buildings exposed on all or several sides obviously require a varying amount of heat according to aspect. Additional economies in fuel can be effected by dividing the building into zones and arranging the pipework so that the heat supply to each zone can be controlled independently according to the separate requirements, by a valve operated by one of the following :—

(*a*) a thermostat placed in a " key " room in the zone ;
(*b*) averaging thermostats placed throughout the zone ;
(*c*) an " inside-outside " control.

As an example of the economy which can be effected, a building will be considered that is to be maintained at 60° F. with an outside temperature of 30° F., i.e. 30° F. rise, the heat input being 4,000,000 B.Th.U. per hour. When working under these conditions the load will be 100 per cent. If owing to milder outside conditions or other causes the internal temperature rises 5° F. in excess of requirements, due to the heat input being uncontrolled, this represents a waste of $\frac{5}{30}$ of 4,000,000 = 666,666 B.Th.U. per hour, or over 16 per cent.

At 10,000 B.Th.U. per lb. of fuel this amounts to 66·6 lb. of fuel per hour, or nearly $\frac{3}{4}$ ton per day.

Boiler Control. Controls attached to key points in a building and controlling only the flow of heat to those sections necessitate some type of control at the boiler or combustion will continue unchecked in spite of a reduction in the heat supplied to the several parts of the building.

An alternative system of temperature control in buildings heated by hot water pipes or radiators that has been developed in recent years in connection with natural draught boilers fired with solid fuel, consists in automatic regulation of the boiler dampers by suitable thermostats. This system can be applied to most types of boiler used on central heating installations.

On stoker fired and gravity feed boilers, thermostats are fitted nowadays as standard practice to control the temperature in the boiler. An extension of the control to the system being served will result in additional economy in fuel.

With hand fired boilers it is frequently possible to show a 25 per cent. saving in fuel by fitting thermostatic damper control. There is also a substantial saving in labour, as the attendant need not watch the water temperature and adjust the dampers to meet the prevailing conditions. His duties are confined to seeing that there is sufficient fuel on the fire and to removing the ash and clinker.

Electrically operated thermostatic switches are used in this system. The check draught damper and air inlets are operated by a motor, of which there are various types on the market. The electric supply to the motor is controlled by a thermostat switch in such a manner as to open the air inlets to the fire, and close the check draught damper when the boiler temperature falls, and to return them to the no-load position when the desired temperature is reached.

The equipment should provide for automatically returning the dampers to the no-load position in the event of failure in the electric supply. This is important as a safeguard against damage to the boiler and waste of fuel through the air doors being left open.

The system control may be a room thermostat installed in a " key " position

in the building being heated, which will close the dampers when the desired temperature is reached, and open them again with a slight fall in temperature.

As an alternative to " key " room control, the building temperature can be controlled according to conditions out of doors, by an " inside-outside " control. Such instruments have two sensitive elements, one of which is mounted in a suitable position out of doors and the other inserted into the main flow pipe. The function of the outside element is to adjust automatically the setting (flow temperature) according to the conditions prevailing out of doors and the control then operates the damper motor to maintain that temperature. Provision is made for adjusting the temperature range of the control to the requirements of the building. The installation can be changed over to a lower operating temperature at night either by a hand switch or automatically by time switch.

This system of boiler control is very flexible and can be designed to meet a variety of different conditions, particularly where heating and hot water services are supplied from the same boiler.

The zone control thermostats must be linked with a master thermostat in

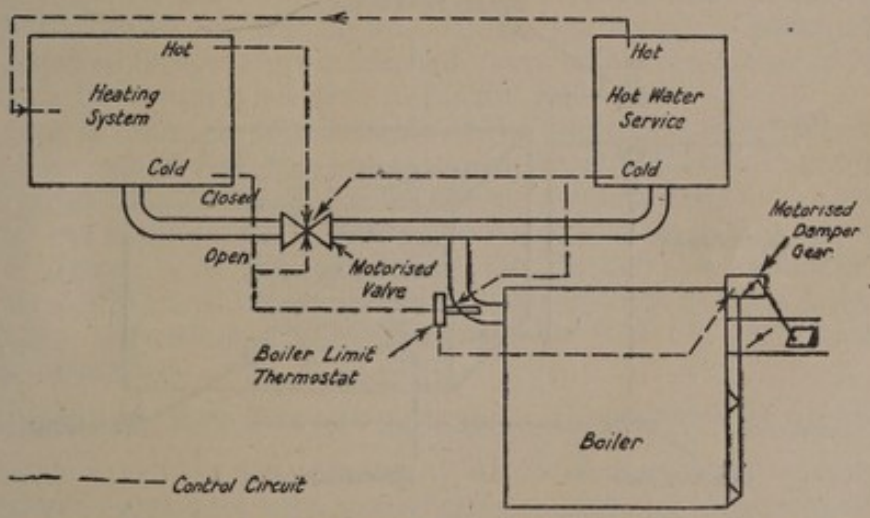

FIG. 248. System for dual control of hot water and central heating.

the boiler, which will prevent the air inlet dampers being opened if the water temperature is up to the maximum.

Fig. 248 shows a dual system, on which the controls are arranged to give preference to the hot water service. When the hot water is under temperature the motorised valve on the flow to the heating system is closed until the demand is satisfied. The hot water system thermostat then transfers control to the heating system thermostat which re-opens the motorised valve if the heating system is below temperature. Both of the system thermostats are linked with the boiler damper controls through the boiler limit thermostat, so that whenever there is a call for heat the air inlet dampers are held open provided that the maximum boiler temperature has not been reached.

HOT WATER SUPPLY

Hot water is required for a variety of purposes in offices, shops, and process work. There is a temperature below which the water is not hot enough for its purpose and above which it must be maintained. Temperatures too low lead to waste of fuel since the supply is often allowed to run to waste in the hope that the water will become hotter. Higher temperatures than are necessary are also wasteful of fuel. Precise control of the temperature within these limits by manual methods is expensive and ineffective. The waste of fuel may

be direct as in the example given below, or indirect through additional radiation and convection losses from the hot surface.

An example of possible waste is the operation of a bank of 10 shower baths using a total of 9,000 lb. of water per hour.

If this water is heated to 105° F. instead of the correct temperature of 100° F., the increase of 5° F. in temperature will be responsible for 5 × 9,000, or 45,000 B.Th.U./hr. of additional heat at the sprays.

If the water was originally at 40° F., the heat required is 60 × 9,000, or 540,000 B.Th.U./hr., so that the increased temperature causes a waste of not less than 8·3 per cent.

PROCESS HEATING

The prescribed temperature for a manufacturing process is usually determined experimentally in the laboratory and may require to be maintained within close limits. Control by thermostatic means should be investigated,

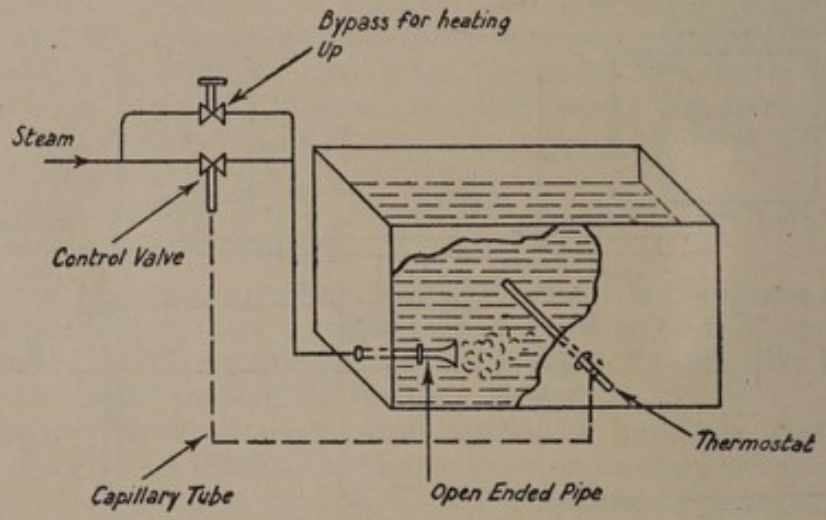

FIG. 249. Thermostatic control of process heating.

not only with a view to saving the fuel which would be wasted if the temperature rose too high, but also in order to prevent spoilage. Production is a matter of vital importance, and if a batch of the material is spoilt, not only does production suffer to that extent, but in addition there is wastage of material and of the fuel consumed in generating the heat units which have been put into it.

The boiling of liquids in open tanks and vats, particularly by the direct injection of steam and where the process is a lengthy one, is a common source of steam wastage. As the temperature of boiling point cannot be exceeded, the thermometer gives no indication of excess steam being supplied. A safeguard against this is a thermostatically controlled valve on the steam supply with the thermostat immersed in the liquid. With a little care in setting the thermostat and with the correct size of valve, the liquid can be prevented from boiling too vigorously, with a considerable saving of steam compared with uncontrolled conditions. Controls would be set a fraction of a degree (less than 0·1° F.) below the boiling point; since the boiling point depends on the barometric pressure, a barometric control would be added on the thermostat.

Where quick heating is required, the size of control valve which is correct for maintaining the liquid at boiling point will probably be too small to supply the steam at the rate necessary for heating-up. This can be overcome by

fitting the thermostatic valve in a by-pass of reduced size, around the hand valve in the main supply, and using the hand valve for heating-up. When boiling point is reached the hand valve should be closed and control left to the thermostat (see Fig. 249).

To illustrate the waste which can take place on process plant, a closed vessel is considered containing 1,000 gallons of water, which has to be brought to boiling point in one hour and kept at that temperature for a period of several hours.

(1) *Heating-up.* Heat required to raise 10,000 lb. of water through 160° F.
= 1,600,000 B.Th.U., excluding heat losses.

(2) *Maintenance of temperature.*
Surface area of vessel exposed to atmosphere = 70 square feet.
Difference of temperature between vessel and atmosphere = 150° F.
Heat losses = 2 B.Th.U./sq. ft./hr./°F. temperature difference (Chapter VIII).

$$= 2 \times 70 \times 150$$
$$= 21{,}000 \text{ B.Th.U. per hour.}$$

The heat required per hour to maintain the temperature is thus only 2 per cent. of that required for heating-up. Unless the supply is promptly checked when the required temperature is reached, there will be a wastage of 98 per cent. for as long as the steam is allowed to remain full on.

No account is taken in this example of the additional losses by evaporation from the open surface of the liquid, which would, however, not materially affect the great disparity between the steam requirements for heating-up and for maintaining temperature, which is the point it is desired to emphasise. In practice, of course, with manual control, the steam valve would be throttled down by hand after heating-up, but even if prompt action were taken it would be practically impossible to throttle down by just the right amount, and wastage is inevitable.

WASTE HEAT RECOVERY FROM INTERNAL COMBUSTION ENGINES

Fuel can be saved by the recovery of waste heat rejected by internal combustion engines, and here also temperature control plays an important part.

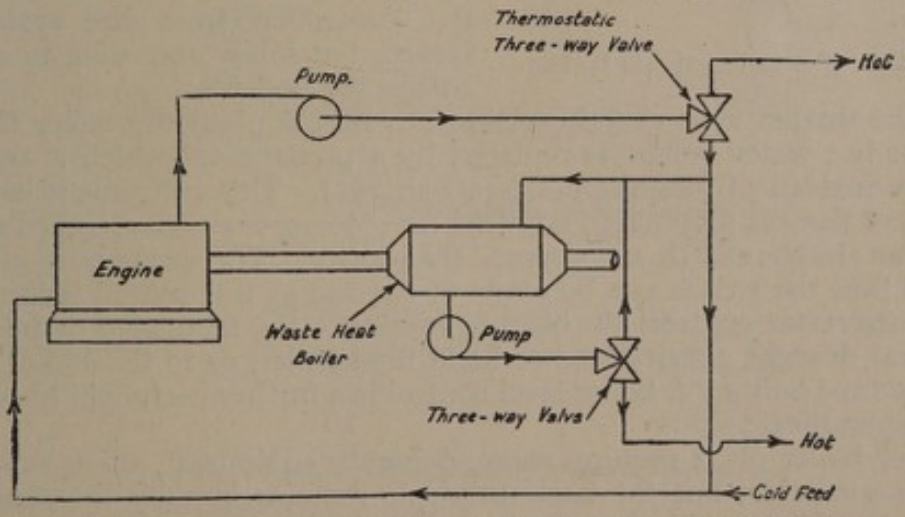

FIG. 250. Recovery of waste heat from internal combustion engines.

Approximately 60–75 per cent. of the heat energy created by the combustion of the fuel finds its way into the cooling water from the cylinders and into the exhaust gases. The whole of the waste heat from the cylinders can be recovered, provided use can be made of the temperature ranges available.

Only part of the heat in the exhaust gases can be recovered, the limitation

being the minimum temperature to which the gases can be cooled without causing condensation and consequent corrosion.

By fitting in the engine cooling water circuit a three-way valve (see Fig. 250) controlled by a thermostat, the water is re-circulated until the temperature rises above the prescribed limit, when a portion is diverted for use elsewhere and cold make-up water is admitted to the circuit. The water rejected from the cooling circuit supplemented, if necessary, by an additional volume of water can then be passed through waste heat boilers, where it is heated by the exhaust gases from the engine.

On the water outlet from the boiler is fitted another three-way valve and thermostat which keeps the water re-circulating through the boiler until it is hot enough for the purpose for which it is to be used.

On a large electric generating plant installed some years ago driven by internal combustion engines having a power of 3,550 B.H.P., the waste heat recovery scheme was designed to provide 51 per cent. of the winter heating load and the whole of the summer load. Without the heat recovery scheme it was estimated that the weight of fuel consumed would have been increased by nearly 50 per cent.

CONTROL OF STEAM BOILERS

The advantage of fitting thermostatic control devices at the points of heat consumption will be largely negatived if the boiler plant is allowed to generate more heat than is being consumed. It is useless to hold back so many pounds of steam at the consuming end if they are going to escape through the safety valve at the boiler. The supply must be regulated to the demand, and this can readily be achieved by boiler control on the lines of Fig. 248. Under these circumstances the rate of combustion is not controlled from the system being served, but solely according to conditions within the boiler.

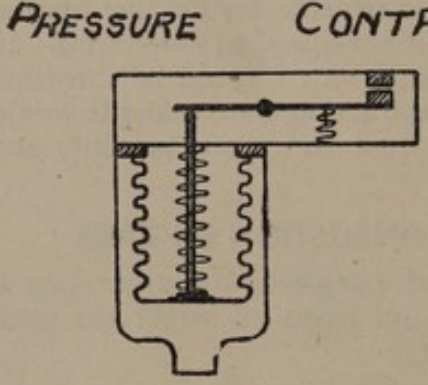

FIG. 251. Control of steam boilers.

On the smaller type of industrial steam raising plant the boiler thermostat used for hot water boilers is replaced by a pressurestat which is sensitive to pressure instead of temperature (see Fig. 251). This instrument comprises a bellows or flexible diaphragm on which the pressure operates, linked mechanically to an electric switch and opposed by a spring. The spring tension is adjustable so that the switch can be made to operate at the desired boiler pressure. The pressurestat controls the electric supply to the motorised damper gear in a natural draught gravity feed or hand fired boiler, or to the stoker motor on a stoker fired boiler. A boiler level control is a further useful aid to economical boiler operation.

Larger boiler plant requires more elaborate equipment, there being several systems available, but the basic principle remains broadly the same, except that the auxiliary operating power may be hydraulic or pneumatic instead of electric.

A note of warning may be sounded here to the effect that while boiler control systems will regulate the draught in accordance with the load, they cannot influence the management of the boiler in other directions. It remains necessary for the attendant to see that his fire is in good order, containing no holes or thin places, for the plant to be maintained in good order, and so forth.

THERMOSTATIC CONTROL EQUIPMENT

All thermostatic control appliances comprise essentially an element sensitive to temperature and a controlling device (valve, switch or lever). For the temperature-sensitive element, use is made of physical changes produced by

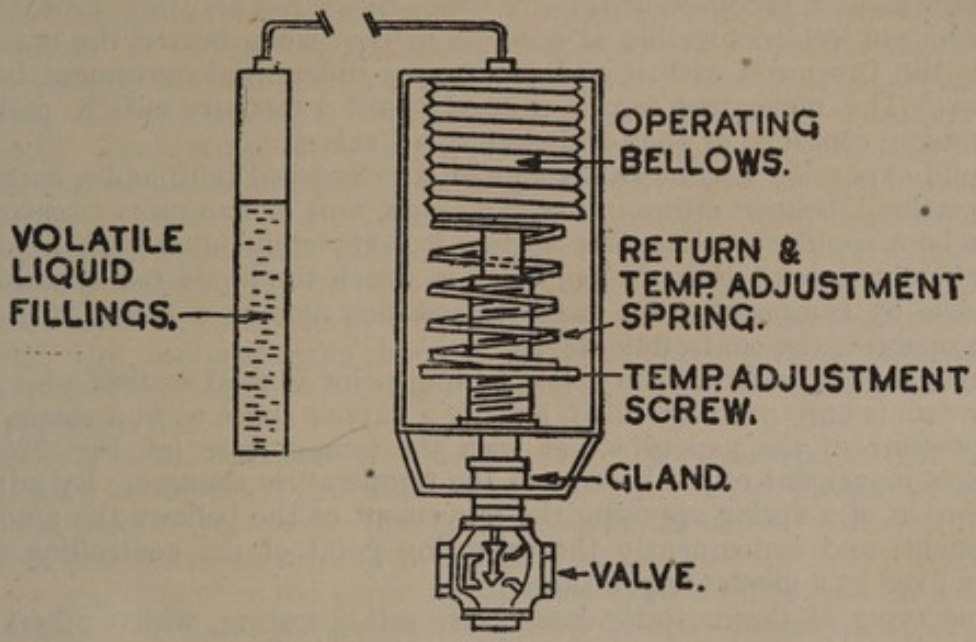

FIG. 252. Volatile liquid thermostat.

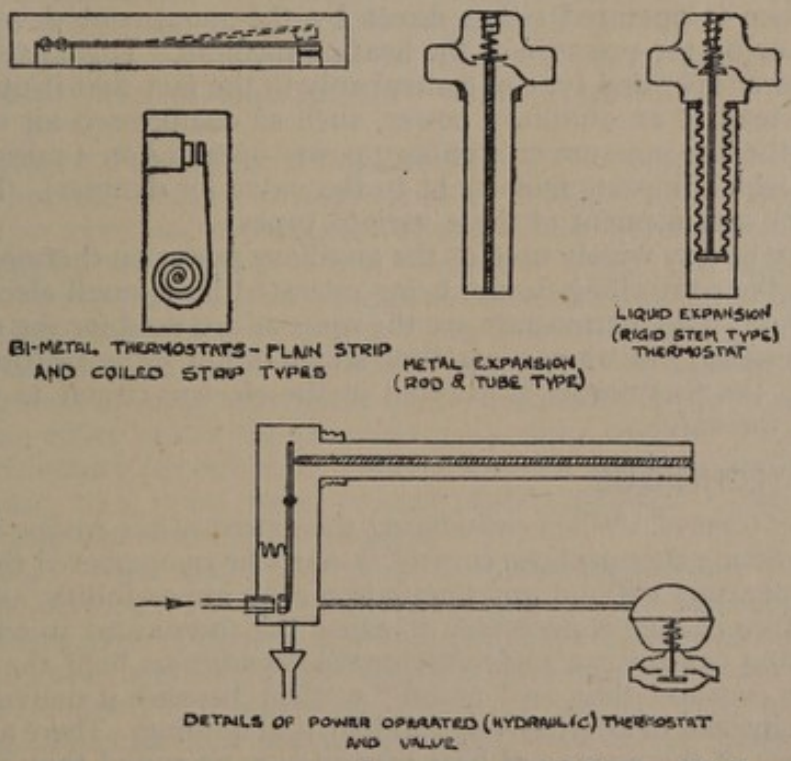

FIG. 253. Bi-metal thermostat and power operated (hydraulic) thermostat.

the application of heat, such as the expansion of metals, increase in volume of a liquid or increase in pressure of a volatile fluid. In all these methods heat produces the movement which is essential to operate the controlling device.

Thermostats constructed on the metal expansion principle comprise either a rod within a tube or a strip of " bi-metal," consisting of two sheets of metal

having widely different coefficients of expansion united face to face to form a single sheet and cut into strips. The metals generally used are brass and " Invar," a steel alloy with negligible expansion. The strip is used either flat or formed into a coil or helix, one end being fixed and the other free. When heat is applied the strip bends due to the unequal expansion of the two faces and movement is produced at the free end. In the rod and tube type, the two members are welded together at one end so that when heated the brass tube carries the Invar rod with it and produces a differential movement between the two. This movement can be utilised to tilt a mercury switch, make and break metal contacts, or open and close small valves.

Liquid expansion thermostats comprise a cylindrical bulb and a corrugated " concertina " bellows either in one rigid stem unit or two units connected by a fine bore capillary tube. The system is completely filled with liquid and hermetically sealed ; thus the only way in which the liquid can expand when heated is by compressing the bellows, and this acts as a piston, the rod of which operates the controlling device.

Where a volatile fluid with a low boiling point is used instead of a liquid, the system is only partially filled, leaving a vapour space as in a steam boiler. The pressure of the vapour varies with the temperature (cf. Fig. 225), and produces movement of the bellows as the temperature changes. By adjusting the tension of a spring opposing the movement of the bellows the amount of movement, and consequently the operating point of the controlling device, can be fixed at a given temperature.

Some types of thermostatic control are self-operating, whilst others make use of an outside source of power, such as compressed air, water, oil or electric current to operate the controlling device. In the self-operating type the controlling device is operated either direct by the movement developed in the thermostat or by the pressure of the heating medium. This latter type differs from the power-operated type of control only in the fact that it utilises the line pressure instead of an auxiliary power, such as compressed air or water. In both types the line pressure or auxiliary power operates on a pressure-sensitive diaphragm which imparts movement to the valve (or damper). Figs. 252 and 253 show the arrangement of these various types.

Electricity is very widely used as the auxiliary power on thermostatic control equipment, the controlling device being operated by a small electric motor or electromagnet. The thermostats are the same as are used for the direct control of electric heating, but when controlling steam or hot water heating, the switch operated by the thermostat is inserted in the electric circuit to the motor or solenoid on the valve.

GENERAL TECHNIQUE

Latitude of Control. When considering the extent of the saving which can be effected by fitting thermostatic control, it must be remembered that control at a fixed temperature without any fluctuation is an impossibility, as some degree of temperature change is necessary to cause the thermostat to operate.

A throttling control can under favourable conditions hold the temperature more nearly constant than an " on-off " control, because it delivers some heat all the time instead of alternatively full bore and nothing. There are operations for which one of these types of heat control is more suited than the other ; it has been stated that over all industrial operations to which automatic heat control is applied, 65 per cent. are fitted with " on-off " control. Another point to bear in mind is that a thermostat which has an operating differential of say $\pm$ 1° F. will not necessarily maintain within these limits the temperature of the air or liquid surrounding it. If the rate of heat input is excessive during the period that the thermostat is calling for heat, the control temperature will be

exceeded due to the fact that too much heat is allowed to pass before the thermostatic element has time to respond to the temperature change.

Exceeding the required temperature can be avoided by setting the thermostat to cut off the heat at a lower temperature and allowing the " over-run " to bring the temperature to the required point. The result of this adjustment will, however, be only that there will be a minus error in the control instead of a plus error. It will be seen, therefore, that closeness of control depends not only on the operating differential, i.e. the temperature change necessary to cause operation of the thermostat, but also on the heat balance in the plant being controlled. There must, of course, be some additional heat available in order to give the thermostat something to control, but there must be some reasonable relationship between the total heat available and the total heat required.

Position of the Thermostat. The thermostat can only react to the temperature immediately surrounding it. Where conditions of uneven heat distribution exist care must be exercised in selecting the place where the thermostat is to be fixed. Hot and cold spots should be avoided and a position chosen where the temperature is at an average figure representative of the conditions required to be maintained.

Effect of Plant Conditions on Control. Satisfactory automatic temperature control may be expected generally where the plant is not incapable of being controlled within reasonable limits by constant manual adjustment. If, however, it is found impossible to hold the temperature within the required limits by standing over the plant and making hand adjustments immediately fluctuations in temperature are indicated on a sensitive thermometer, it can be safely assumed that the necessary heat balance does not exist. It is useless under such circumstances to attempt to secure satisfactory control by automatic devices before the fault in the plant itself has been rectified. It is impossible to lay too much emphasis on this point, as it appears to be one of the commonest causes of unsatisfactory control, especially on steam heated apparatus.

Steam Pressure. Frequently a high pressure steam supply is taken for low temperature heating, because it happens to be available in the works and the engineer has an aversion to reducing valves. Under hand control the supply is taken through a valve which is just " cracked " open, and still the plant is overheated. The engineer then decides to install automatic temperature control and is inevitably disappointed because there is little improvement.

It should, however, be obvious that if a " cracked " valve admitted too much heat, the slightest opening movement of the automatic control valve will produce the same effect before the thermostat has time to close the valve again. The only satisfactory remedy is to reduce the steam pressure, and thereby also its temperature, to a point where a controllable quantity is required and a reasonable valve movement is permissible.

Size of Valve. With steam heating, it is of vital importance to fix the size of the control valve correctly according to the consumption and supply pressure. With a throttling type of control it is very advisable that conditions should be such that the valve operates with a good lift, and does not hover close to the seat, as this produces " wiredrawing " or wear on the faces of the valve, which in time causes leakage through the valve when in the " closed " position. A larger valve opening under working conditions can be obtained by using a smaller valve.

It is quite a common fault to install a control valve of the same size as the steam inlet provided on the appliance, without regard to the amount of steam required, it being overlooked that plant manufacturers naturally make provision for the lowest steam pressure on which the plant will operate, which may be much lower than the pressure actually being used.

An oversized valve gives much the same results as too high a steam pressure. The oversize valve admits excess volume at suitable temperature and too high a steam pressure admits a smaller volume at excess temperature (or pressure) —but the result is the same.

Type of Control. A further essential is to select the right type of control for the particular process involved. The temperature to be maintained will determine the type of thermostat, the nature of the heating or cooling medium will govern the form of the regulating device (valve or switch). The self-acting regulator, providing it fulfils the requirements, is to be preferred because of its simplicity and independence of outside sources of power, the use of which in itself means consuming fuel. Moreover, if the source of power should fail the control would be rendered inoperative.

There are many more factors to be taken into consideration in applying thermostatic control, and it cannot be too strongly emphasised that the advice of experts in this field should be sought by those interested.

Thermostatic control of gas heated ovens and furnaces is described in Chapter XXVII.

MAINTENANCE OF THERMOSTATIC EQUIPMENT

Maintenance is important in the successful operation of all instruments. Since thermostats are responsive to very small changes of temperature, they must be handled with the care due to a sensitive instrument.

The following instructions for the maintenance of control instruments were issued by the Ministry of Fuel and Power in Fuel Efficiency Bulletin No. 11, and are here reproduced :—

Thermostats. Immersion thermostats should be checked for leaks at the point of immersion ; the continual expansion of the apparatus into which they are placed may cause the holding nuts to loosen.

The thermostat should be removed from time to time for inspection for corrosion and deposit. If deposits are allowed to accumulate they may act as insulation and make the action sluggish. Terminals should be kept clean and tight. Covers should always be in place and the position of the scale pointer should be checked.

An occasional check should be made to ensure that the thermal action of the thermostat is correct ; no unauthorised person should tamper with the settings.

Thermostats should never be oiled.

Dampers and Damper Operating Motors. Maintenance of damper linkage is most important, and should be carried out at regular and frequent intervals. The motor links are disconnected and the dampers and their linkage operated by hand. Any stiffness should be immediately rectified, particular care being given to the bearings of primary and secondary air dampers.

Motors should be checked in the usual way. They should in general require no lubrication, since they are sealed and the moving parts are immersed in oil.

When the damper motor is in the " fully open " position, the damper must be checked to ensure that it, too, is fully open. When a check damper is fitted care must be taken to ensure that it opens properly when the main damper closes.

Motorised Valves. These must be examined for dirt and loose connections in the terminal box and the motor insulation must be tested. The valve gland should be examined for traces of leakage. On certain types the gland nut should on no account be tightened. The makers' instructions should be followed regarding lubrication ; in general, lubrication would only be applied if points are provided.

Relays. The interior should be kept clean and examined for loose con-

nections. It is necessary to make sure that the armature can be freely closed by hand, and that it has no tendency to stick in the " closed " position. It is very rarely necessary to clean the contacts ; on certain types of relay this may be actually harmful.

If the relay has become noisy after a period of service, the pole faces of the electromagnet should be cleaned. To do this, a clean piece of paper is placed between the pole faces, the armature is pressed down firmly with the fingers and the paper is withdrawn whilst maintaining the pressure. Usually the armature pivot does not require lubrication, but when lubrication is desirable, oil should be used very sparingly.

Covers should always be in position.

Humidity Controls. Hair elements should be washed with a camel's-hair brush and clean ether ; wetting with distilled water will moisten the hair element to its original sensitivity.

Compensated Control Systems. This type of control system as well as any other more complicated controls must be carefully checked for accuracy at all primary and secondary control points. A differential inaccuracy at any point in a line of controls may throw the whole system out of adjustment.

Differential Adjustments. It may be found that the number of operations in the plant are too many ; this may be reduced by increasing the operating differential. The operating differential should be as narrow as possible and brought to a point consistent with the heat distribution requirements. This also applies to pressurestats operating stokers, motorised valves, dampers, etc.

Differential adjustments should be checked for cut-in and cut-out against the flow thermometer and/or pressure gauge. The cut-in pressure or temperature plus the differential equals the cut-out pressure or temperature.

Low Water Controls. The blow-down valve should be opened daily and the float chamber should be opened occasionally in order to remove any sludge that has accumulated. Switch connections should be checked for dirt, etc.

Control Valves. Where the controlling device is a valve, it should be protected against foreign matter by a pipe line strainer which should be cleaned periodically as may be found necessary. The valve itself should be examined to see that the seatings are clean and not wire drawn, and that its movement is free.

Repairs. Where a control is beyond ordinary maintenance it should be removed and sent to the makers for repairs. Thermostatic controls are highly specialised instruments and major repairs should only be carried out by qualified mechanics.

Alarm Controls. Where failure of a thermostat would have serious consequences, an alarm to give a visible or audible warning that a dangerous temperature is being approached is recommended.

A test switch is advisable in such alarm circuits, or in connection with automatic safety devices which are only called upon to operate in emergency. A test should be made regularly by this means to make sure that the circuit is intact and operates correctly.

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CHAPTER XXVII

AUXILIARY PLANT FOR THE EFFICIENT UTILISATION OF GAS

Nomenclature—Industrial gas fitting—Governors—Burners—Flues—Cocks—Temperature control—Thermostats and pressurestats—Pilots, cut-offs and back pressure valves—Meters.

THE utilisation of town gas involves its distribution and combustion. The methods adopted for calculating the flow of gas in pipes have been indicated in Chapter IX.

The properties of town gas have been discussed earlier in this book, particularly in Chapters II, IV and V. The calorific value, the composition of the flue gases, the correlation of flue gas losses with temperature, the quantity of excess air required for combustion, the flame temperature, the limits of inflammability of gas/air mixtures and ignition temperatures, are all important factors in the use of town gas for industry which have been already discussed.

On the practical side of gas utilisation it is necessary to consider the transmission of gas within the works, the gas burner and the application of ancillary appliances, such as governors, thermostats and meters, which enable the gas to be utilised satisfactorily and at the highest efficiency.

These matters, which may be termed the physical basis of gas utilisation, are the subject-matter of the present chapter.

NOMENCLATURE USED IN GAS SUPPLY PRACTICE

Pipes running underground from the gas works supplying the district are known as mains.

The connection between a main and a consumer's meter is known as the service.

Connections between the meter and appliances are known as supplies, carcass or distributing pipes.

The gas pipes inside a factory must be of such capacity that the pressure loss is as small as possible. As a general rule the maximum pressure difference between the main outside the factory and any gas consuming appliance supplied from it should not exceed $\frac{5}{10}$ inch w.g., including pressure loss through the meter.

This is an ideal to be aimed at and is not difficult to secure in small installations. In larger installations, however, a somewhat greater pressure loss is permissible, up to 1 inch w.g. including the resistance of the meter.

INDUSTRIAL GAS FITTING IN FACTORIES

This section refers to matters generally dealt with by the gas undertaking, but which require also to be understood by the factory staff concerned if the gas installation is to be arranged to the best advantage.

Meter—Choice of Site. The meter should be fixed as near to the boundary wall as practicable and in a dry position, not subjected to extremes of heat or cold. If it is of a size not normally in stock and/or gas is in continual demand for essential production, the meter can be arranged with a " by-pass," that is to say, the inlet and outlet supplies can be connected with an isolating valve provided in the by-pass having a seal approved by the gas undertaking. This arrangement allows for the meter to be exchanged without the necessity of stopping production.

It should at all times be possible to remove the meter for exchange without the need for demolishing walls or other structures.

The connections around the meter must be arranged to avoid strain on the case, and the local gas undertaking should be approached for advice on this.

If air or industrial gases under pressure are used in conjunction with the town supply, non-return valves must be fixed in suitable positions, and where equipment is used which can draw gas from the undertaking's system, such as a compressor, booster, gas engine, or certain types of air blast equipment, some form of approved safety cut-off or switch must be employed to avoid damage to the meter or the introduction of air into the mains if these are broken.

Supply or Distributing Pipes. The size of the supply pipe to be laid from the meter is usually that provided on the meter, or can be deduced from flow formulæ (Chapter IX), bearing in mind the importance of reducing the pressure loss as much as possible, and the requirements of possible future installations. A size slightly larger than that calculated should be installed.

Essential tools for pipe fitting include pipe cutters and stocks and dies, or a screwing machine and foot-prints, tongs or pipe wrenches. Other tools required for pipe hangers, cutting away, etc., are assumed to be available.

As far as practicable straight pipe runs are best, and where turns have to be made wide sweep bends should be used in preference to elbows, as sharp turns increase frictional losses (cf. Table 53, page 177).

Unless it is known that additional gas appliances will not be installed later, it is desirable to leave a number of plugged tee-pieces on the line which can be used if required, without interruption of the supply.

If heavy pipes have to be supported from roof members, care must be taken that they can take the additional weight, or an alternative method of support must be found. Contact between the gas supply pipe and the electrical system must be avoided. Escapes are likely to follow any method of fixing which will result in continual vibration of the gas pipe.

To avoid the need for a complete shut down if one appliance has to be disconnected, controls are to be recommended, but as valves and cocks are in short supply owing to war conditions, wherever possible appliances could be grouped for isolation purposes.

GOVERNORS AND GOVERNING

Most gas appliances can be satisfactorily operated only when the pressure at the burner is kept reasonably constant. The combined effects of varying pressure in the mains and pressure losses in the supply pipes may cause such wide variations in burner pressure that frequent adjustment of the controls is necessary, even where automatic temperature regulators are fitted. If in such circumstances, a governor is attached, either to the appliance or to the supply pipe serving a group of appliances, then under normal conditions of supply, the performance of the appliances can be predicted and maintained, since the burner pressure will remain constant within narrow limits.

There are two types of governor. One maintains a constant pressure and the other a constant gas rate. The former are known as pressure governors and the latter as volume governors. Volume governors have a somewhat restricted use. They are usually incorporated in appliances such as water heaters and gas lamps, which require an unvarying gas rate, but since their numbers are relatively small in comparison with pressure governors, their design and characteristics need not be considered in detail.

A gas pressure governor may be defined as a device for delivering a constant and predetermined pressure from a supply in which the pressure is varying. Its function is to reduce high pressure peaks to a level value, but this value must of necessity always be lower than the minimum pressure in the supply pipe. A governor cannot increase pressures.

The advisability of using governors has been further emphasised by the war-time necessity of adjusting appliances to sub-normal pressures.

TYPES OF PRESSURE GOVERNORS

Pressure governors may be broadly divided into two main types, high pressure and low pressure ; the high pressure governors are used where inlet pressures are in excess of 2 lb. per square inch.

Pressure governors, in general, do not give an absolutely constant outlet pressure under all conditions of use. Their performance is influenced both by variations in inlet pressure and changes in gas rate.

This is particularly true of the simple low pressure governor, but a type known as the compensated low pressure governor gives a much finer degree of pressure control by incorporating a compensating device which nullifies inlet pressure variation effects.

The principles of operation and constructional details of both types are outlined later.

GOVERNOR PERFORMANCE

Performance figures for a typical simple low pressure governor, set to give a nominal outlet pressure of 2·5 inches w.g., are shown below :—

		Outlet pressure	Variation	
	Inlet pressure 3 ins. w.g.			
Gas rate	10 per cent. max.	2·8 inches w.g.	Gas rate effect	Total 0·6 inch w.g.
Gas rate	Max.	2·5 inches w.g.	0·3 inch w.g.	
	Inlet pressure 8 ins. w.g.			
Gas rate	10 per cent. max.	2·5 inches w.g.	Inlet pressure effect	
Gas rate	Max.	2·2 inches w.g.	0·3 inch w.g.	

Corresponding figures for a compensated low pressure governor are :—

		Outlet pressure	Variation	
	Inlet pressure 3 ins. w.g.			
Gas rate	10 per cent. max.	2·6 inches w.g.	Gas rate effect	Total 0·2 inch w.g.
Gas rate	Max.	2·4 inches w.g.	0·2 inch w.g.	
	Inlet pressure 8 ins. w.g.			
Gas rate	10 per cent. max.	2·6 inches w.g.	Inlet pressure effect	
Gas rate	Max.	2·4 inches w.g.	nil	

Distributing pressures vary between different undertakings and are not normally so high as the 8 inches shown above ; although this value gives outlet pressure variations slightly higher than those likely to be obtained in practice, this figure is chosen in order to allow a margin of safety.

APPLICATION OF GOVERNORS

Governors may either be fitted to the supply pipe feeding a group of appliances or attached to individual appliances. The first practice is known as " service governing " and the second " point governing."

Service Governing. The service governor is usually of the compensated type, as its duties are more exacting than those demanded from the point governor. Where service governing is employed, relatively large variations in outlet pressure are to be expected owing to the intermittent gas demands of the individual appliances in the group, and the pressure loss in the supply pipe between the governor and the appliances.

Service governing is, in general, found to be cheaper and simpler to apply, but in choosing this method care must be taken to ensure that the pressure loss

in the supply pipe between the governor and the most remote appliance in the group does not exceed 0·3 inch w.g.

Point Governing. Point governing is to be preferred to service governing as it is less susceptible to the external influences (varying gas rate demands of individual appliances and supply pipe pressure loss) outlined under " service governing." For this purpose, the simple type of governor is used, as it is cheaper than the compensated type, although the compensated governor may be used where more accurate pressure control is required.

From the foregoing and the table of performance figures it will be seen that accuracy of governed burner pressure control is in the following order :—

(1) Point governor, compensated type.
(2) Point governor, simple type.
(3) Service governor, compensated type.
(4) Service governor, simple type.

GOVERNOR CAPACITIES

In selecting a governor, care must be taken to ensure that it is large enough to meet the maximum gas demands of the appliance or group of appliances. As a guide to the size to be used, Table 115 of governor capacities will be found to be useful, but if there is any doubt about the exact maximum gas rate requirements, a larger size should always be used.

TABLE 115. GOVERNOR CAPACITIES

Size	Capacity
½-inch B.S.P.	100 cubic feet per hour
¾ ,, ,, ...	200 ,, ,, ,, ,,
1 ,, ,, ...	350 ,, ,, ,, ,,
1¼ ,, ,, ...	600 ,, ,, ,, ,,
1½ ,, ,, ...	900 ,, ,, ,, ,,
2 ,, ,, ...	1,500 ,, ,, ,, ,,
2½ ,, ,, ...	2,000 ,, ,, ,, ,,
3 ,, ,, ...	4,400 ,, ,, ,, ,,
4 ,, ,, ...	14,000 ,, ,, ,, ,,
6 ,, ,, ...	40,000 ,, ,, ,, ,,

LOW PRESSURE GOVERNOR DESIGN—BASIC PRINCIPLES

A conventionalised diagram of a simple low pressure governor is shown in Fig. 254. Gas enters at I, passes between the valve V and its seating S, where

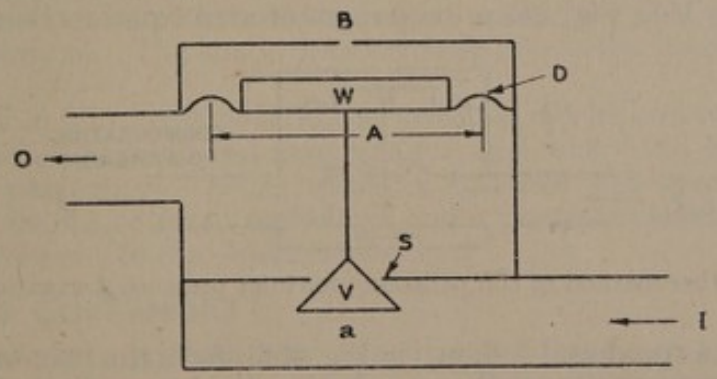

FIG. 254. Simple low pressure governor.

it drops in pressure before passing to the outlet O. This reduced pressure exerts an upward thrust on the diaphragm D and balances the loading W of the diaphragm. If the pressure below the diaphragm falls the diaphragm and its

attached valve will fall and allow more gas to pass to restore the pressure. Conversely an increase in pressure will cause the diaphragm and valve to rise and so decrease the amount of gas passing until balanced conditions are again re-established.

The pressure on the upper side of the diaphragm is maintained atmospheric through the breather hole B. It will be realised by a closer examination of this diagram that this simple type of governor cannot give a constant outlet pressure with varying inlet pressures since the balance of the valve and diaphragm is influenced by the inlet pressure acting on the underside of the valve and tending to close it.

A large ratio of area of diaphragm to area of valve reduces this effect, but there are practical limitations to this process. On the one hand the valve seating must be sufficiently large to pass the maximum required amount of gas at the minimum pressure loss, and on the other hand a diaphragm cannot in the interests of compactness be made indefinitely large. To eliminate the unbalanced forces in the valve due to inlet pressure a number of compensating devices have been developed, two of which will be described.

METHODS FOR COMPENSATING FOR INLET PRESSURE

In the method shown diagrammatically in Fig. 255 a second valve of the same size as the first is attached to the same valve stem and works against a second valve seating. Gas enters at the inlet I and then passes in either direction from between the valves V_1 and V_2 which are of equal size, to the space beneath the

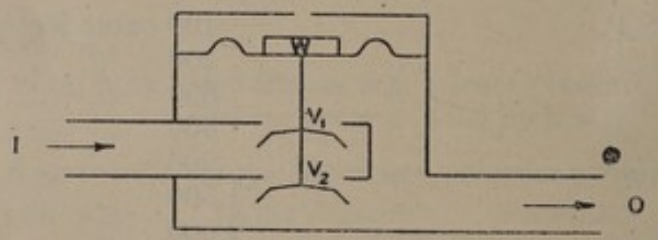

FIG. 255. Method of compensating for inlet pressure fluctuation on low pressure governor.

diaphragm to the outlet O. The principles of operation are the same as for a simple governor, but here the upward force on V_1 due to the inlet pressure is balanced by an equal force acting downwards on V_2, thus neutralising the effects of inlet pressure variations.

Another method of compensation which is widely used is to substitute for the upper valve V in Fig. 255 a diaphragm of area equal to that of the valve.

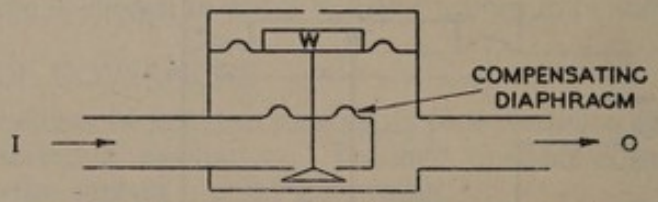

FIG. 256. Another method of compensating for inlet pressure fluctuation on governors.

The governor thus produced is shown in Fig. 256. As in the twin-valve governor, the upward force upon the small compensating diaphragm due to inlet pressure, is balanced by a similar force acting downwards on the valve.

Variations in gas rate also affect the performance of a governor. As the gas rate increases the outlet pressure tends to fall and conversely as the gas rate diminishes the outlet pressure increases. Reference to Fig. 257, which illus-

trates the configuration of a leather diaphragm in two extreme positions, will make the reason for this clear. When the valve is almost closed against its seating to pass a small gas rate the diaphragm centre is higher than when a large gas rate is passing. This vertical movement has the effect of decreasing the effective area of the diaphragm when it is high and increasing it when it is low. When the effective area is greater, the pressure required to support the loading is naturally reduced and consequently the outlet pressure tends to fall.

Governors which incorporate an accurately machined metal piston or a metal bell floating in mercury in place of the diaphragm are sometimes used to avoid

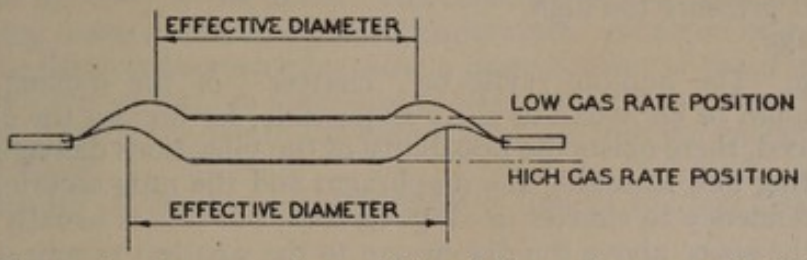

FIG. 257. Illustrating the reason for variation of the performance of a governor with varying gas rate.

these variations of outlet pressure with variations in gas rate, but the degree of pressure control obtained from the conventional flexible diaphragm type of governor is sufficient for all normal requirements.

GOVERNOR MAINTENANCE

Under normal conditions of use, a governor should require little maintenance. It is desirable, however, periodically to examine the diaphragm and, if it is found to be dry, to rub in a little good quality mineral oil. Diaphragms are usually made from East Indian Persian sheepskin, and unless they are kept supple by the application of suitable dressings they are likely to become stiff and so to affect the performance of the governor. In extreme conditions of dryness they may become porous. Long experience has shown that annual inspection is sufficient, but if the governor is unavoidably fitted where it is likely to get warm, more frequent inspections are recommended.

GOVERNOR INSTALLATION

Before fitting a governor, care should be taken to ensure that no packing material has been left inside. Governors are nearly always so packed to prevent jolting and damage to the valve, etc., during transport. Paper packing or other absorbent material should never be used, since this will soak the oil from the diaphragm. The small breather hole above the diaphragm must be open.

The position in which the governor is fitted should be one which will afford easy access for adjustment and maintenance, and where the governor will be free from the possibility of being heated or splashed with grease, liquids, etc. The governor should be fitted truly level and a pressure testing point inserted as near as convenient to the outlet connection.

LOADING OF GOVERNORS

Governors are usually loaded by removing the cover cap and adding discs of lead to the diaphragm pan until the required outlet pressure is obtained. This operation must be carried out when the full gas rate is passing to the appliances. The outlet pressure is conveniently determined at the pressure testing point fitted in the outlet. The top cap must be replaced securely. Loading by means of springs is sometimes employed, and although this method

is less satisfactory than weight loading from the viewpoint of governor performance, it has the advantage that the governor can be used on its side or even upside down.

GOVERNOR FAULTS

A modern governor which has been correctly installed and carefully maintained should give many years trouble-free service, but the following faults may sometimes be encountered.

(1) Chattering.
(2) Outlet pressure too high.
(3) Leakage.

Chattering. The audible rattle or "chatter" of the moving parts of a governor should be corrected as early as possible, for besides the objectionable noise produced, there exists the possibility of the vibrations damaging the valve and its seating and loosening the diaphragm and the nuts securing the valve. There is a tendency to chatter in all governors, but this is usually checked by restricting the space above the diaphragm to the smallest practical dimensions and by using a small breather hole. The chattering is most noticeable when the valve is near its seating when small gas rates are passing under conditions of high inlet pressure. Chattering can usually be traced to a loose or imperfectly sealed cover cap, and a cure is commonly effected by tightening the securing screws. In obstinate cases, it may be necessary to reduce the size of the breather hole, but care must be taken to ensure that the hole is not completely closed. A governor will not work unless this hole is open.

Outlet Pressure too high. Outlet pressures in excess of that to which the governor had previously been adjusted may be due to any or a combination of the following causes :—

(*a*) Dirt or foreign matter under the valve seating.
(*b*) Damaged valve.
(*c*) Choked breather hole.
(*d*) Stiff or leaky diaphragm.

Dirt may be removed by dismantling and cleaning. Damaged valves are uncommon, but those with leather coverings are more susceptible to damage than the wholly metal type and replacement of the leather is all that is usually required. Choked breather holes are usually traceable to blockage by paint or grease which can be removed by means of a piece of wire.

Leakage. Leaks at the diaphragm edge are caused by a slight shrinkage of the leather under compression or by a loosening of the securing nuts and bolts. A leak through the diaphragm centre may be due to a loose securing nut in the spindle. In both instances a cure can be effected by tightening the nuts and bolts. Leaks through the diaphragm may be due to absence of dressing and this defect may be remedied by working in a little good quality mineral oil. Cut or ruptured diaphragms should be replaced.

TOWN GAS BURNERS

The methods of burning town gas in industrial appliances may be divided conveniently into three main groups :—

(1) The simple jet, ejecting gas into the air and giving a luminous flame.
(2) The atmospheric or bunsen burner which ejects a mixture of gas and air giving a flame of which the luminosity decreases with increasing quantity of air used per cubic foot of gas.
(3) The air blast burner in which mechanical means are employed to obtain an air and gas mixture in any desired proportions ; the flame varies from luminous to non-luminous as the proportion of air increases.

The three types are illustrated diagrammatically in Fig. 258.

Whichever type of burner is used the amount of heat liberated by the complete combustion of a given quantity of gas is the same, but the shape, volume, oxidising and reducing qualities, and the flame temperature are widely different, and it is these properties that decide what type of burner shall be used in a given set of circumstances. The characteristics and uses of the three types are therefore given in greater detail.

(1) *The Luminous Burner.* When gas is burned with no previous admixture of air (i.e. no primary air) the air for combustion (i.e. secondary air) mixes by diffusion into the body of the flame. The combustion process is slow, giving an undefined long flame of relatively low temperature. In some burners the lack of form in the flame is overcome by using a large number of small jets (pin-hole burner) with a high discharge rate causing eddy currents in the air surrounding the gas stream which accelerate the diffusion of air into the gas and produce a well-defined flame.

The luminosity of the non-aerated flame is due to the fact that the gas is

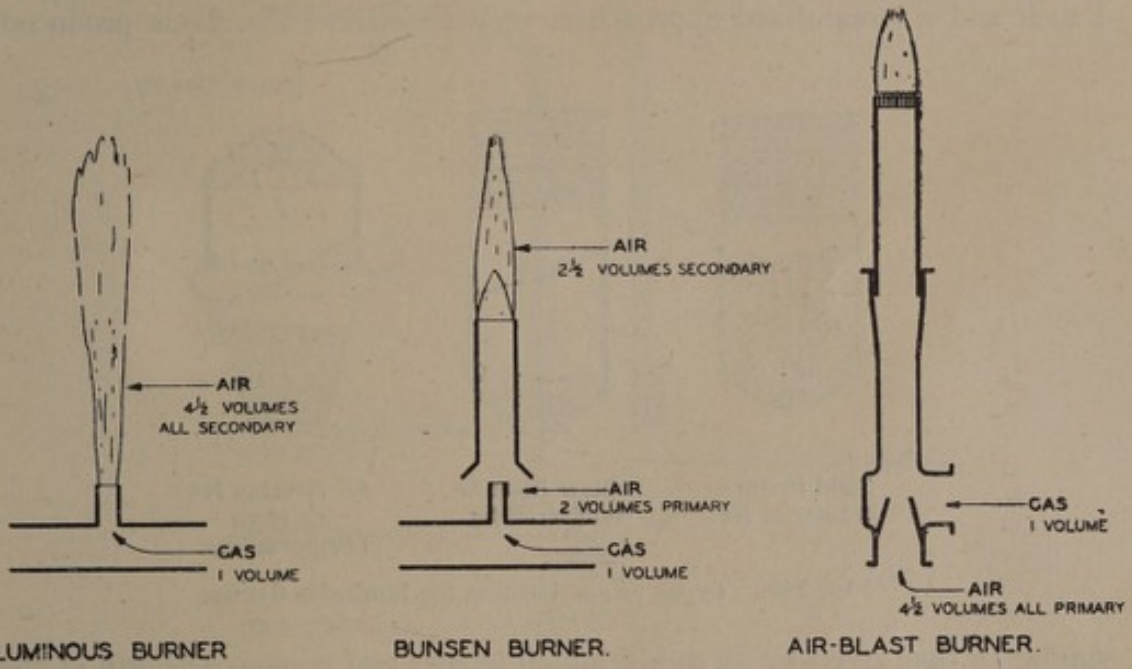

FIG. 258. Types of industrial gas burners.

subjected to a sufficiently high temperature before mixing with air to " crack " the hydrocarbons, particularly the unsaturated ones, producing particles of carbon which are raised to incandescence. If the flame is uninterrupted these carbon particles combine with oxygen to form carbon dioxide, giving out heat, but if the flame is broken by a cold surface the carbon particles are deposited and sooting takes place on the cold surface. In practice, therefore, it is obvious that a luminous flame should not be permitted to impinge on a cold surface, otherwise soot will be deposited and combustion will not be complete.

The most common luminous burner used is known as the industrial jet, shown in Fig. 259, three types of which are illustrated.

The burner is provided with a fireclay head in which the jet is so arranged that the flame is spread out in the form of a thin fan. This type of flame is very stable and offers a large surface area for diffusion to take place. The burners are intended for use at pressures between 1 tenth inch and 25 tenths inch water gauge. This pressure range is below the normal pressure in the supply pipe, and it is therefore necessary to use governors or otherwise check down the full pressure before it reaches the burner jet, the size of which must

be selected to pass the amount of gas required. This does not mean, of course, that governors are only required under these conditions. At the lower limit the flame tends to lose its shape while above the upper limit the flame is liable to become unstable. The jets can be obtained in ten sizes covering the range from 2 cu. ft. per hour to 14 cu. ft. per hour per jet. Burners are made for any designed gas rate by arranging the jets on a conveniently shaped burner float, usually straight or circular. The jets should be so spaced that the flame will run from burner to burner, but without flame impingement, as this destroys the true fan shape and interferes with combustion. Two characteristics of the industrial jet are :—

(1) That any dirt falls off to the side without blocking the jet.
(2) That the jets cool easily and can be used in close formation without overheating.

Apart from the small luminous jets there are also large luminous flames used widely in industrial work. In some applications it has been found that the flame obtained with a simple circular jet having a diameter of the order of ½ inch and a streamlined approach is very effective. The flame produced is

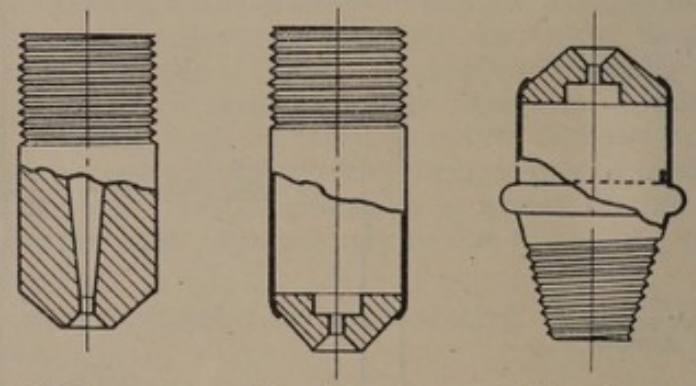

FIG. 259. Types of gas burners for luminous flames.

similar to that given by an atomising oil burner and reaches a length of about 4 feet with a gas pressure of 3 inches w.g. This type of burner is used for the conversion of side flue ovens, oil-fired equipment in general and coke- and coal-fired boilers. There are also many large furnaces in which luminous flames are used, the gas being admitted to the combustion space at relatively low velocity through circular or rectangular ports. An advantage of this arrangement is that it allows the whole of the combustion air to be preheated.

A special form of luminous burner is the type consisting of a central gas jet with a concentric air stream or alternatively of a central air jet with a spiral baffler and a concentric gas stream. This type of burner minimises the turbulence in the air and gas streams and produces a flame in which gradual combustion occurs so that in long furnaces very uniform heating can be obtained.

The general characteristics of luminous burners are :—

(1) Simplicity.
(2) Cheapness.
(3) Ease of control, stability and flexibility.
(4) Ease of application to thermostatic control, as they enable the gas to be cut down to a very low rate without the flame being extinguished or lighting back.

THE ATMOSPHERIC OR BUNSEN BURNER

The simple type of burner shown diagrammatically in Fig. 260 forms the basis of all aerated burners, and is the type most frequently encountered in town gas-fired industrial equipment. Its characteristics are :—

(1) The space required by the flame to liberate an equal quantity of heat is smaller than the luminous flame, hence the heating capacity of a given volume of combustion space is greater.

(2) The flame can be used for the direct heating of solid bodies without soot deposition, while as long as the inner blue cone remains unbroken by contact with the object heated and secondary air is uninterrupted combustion will be almost complete without the formation of more than traces of carbon monoxide.

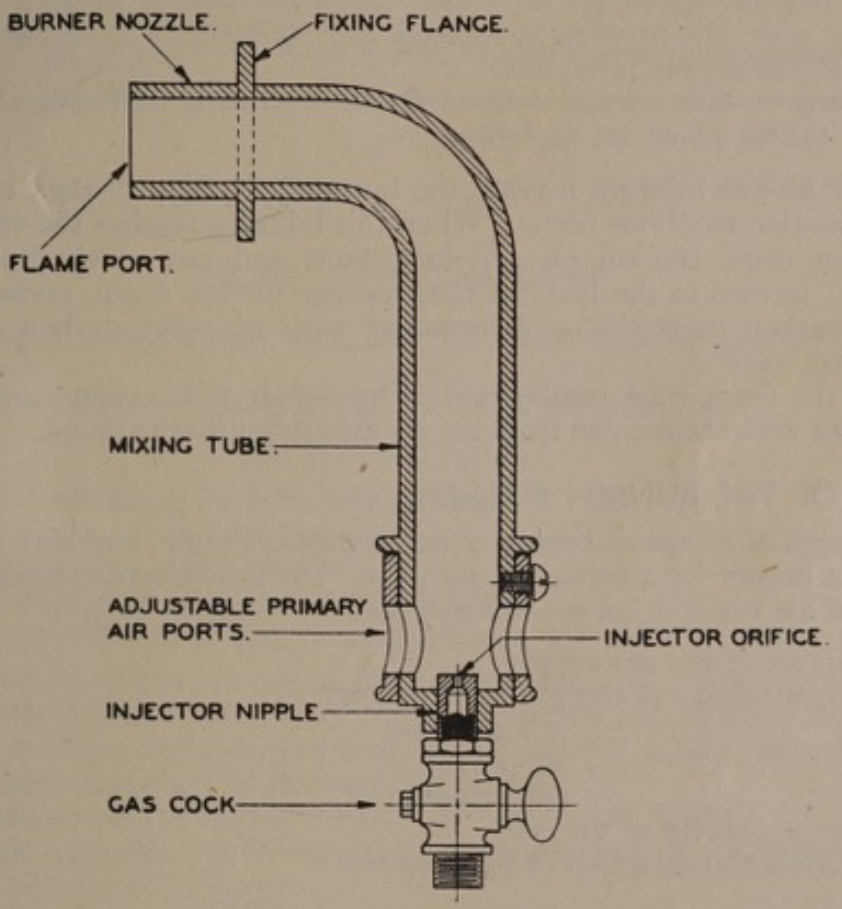

FIG. 260. Bunsen furnace burner.

(3) Objects heated in the outer colourless zone are under oxidising conditions, while the inner blue cone provides reducing conditions. This is a well-known phenomenon, and is used in qualitative analysis for the various bead tests for metals, and is particularly important in such applications as brazing and tempering with an open flame.

Considering the simple burner illustrated in Fig. 260, when the primary air control is completely closed, the flame is luminous and yellow with no defined shape ; as the air shutter A is gradually opened, air is drawn in at the base of the mixing tube and the flame takes a more finite shape, turning reddish blue in colour. As the proportion of primary air increases, the flame divides into two zones, an inner cone of greenish blue and an outer cone which remains reddish blue. It can be shown that the inner greenish blue cone is hollow and

that combustion is occurring only at its surface, while throughout the outer cone, combustion and heat liberation are occurring. If the air shutter is opened still further, the inner cone shortens and finally may light back at the injector. This lighting back needs further explanation, and can be illustrated by the example of a man walking down an upward-moving escalator. The man represents the speed at which a flame can travel in the air-gas mixture, while the escalator represents the speed at which the gas is passing up the mixing tube. When the man is descending at the same rate as the escalator is ascending, he remains relatively stationary; if he increases his speed he will ultimately reach the bottom of the escalator; in the same way, increasing the proportion of air in the gas increases the flame speed which, when it is greater than the mixture speed up the tube, causes the burner to light back. This may occur when the primary air exceeds a proportion of about $2\frac{1}{2}$ to 1 of town gas at ordinary supply pressures.

Returning to the normal bunsen flame with two well-defined zones, the reactions taking place are as follows :—

(1) The air-gas mixture leaving the burner port is preheated, and primary oxidation reactions occur. When this mixture reaches the surface of the inner cone, the complex hydrocarbons and any aldehydes, peroxides, etc., formed in the body of the cone are broken down, giving a mixture of carbon monoxide and hydrogen with nitrogen, carbon dioxide and water vapour.
(2) In the outer cone combustion of hydrogen and carbon monoxide takes place with the oxygen from the air diffusing into the flame.

DESIGN OF THE BUNSEN BURNER

The design of a bunsen burner is by no means simple, and care is needed in selecting a burner for a particular purpose. The following factors influence the amount of air which a burner can entrain :—

(1) Pressure of the gas supply.
(2) Size and shape of the gas jet (injector).
(3) ,, ,, ,, ,, ,, mixing tube.
(4) ,, ,, ,, ,, ,, air ports.
(5) ,, ,, ,, ,, ,, flame ports.
(6) Specific gravity of the gas.
(7) Position and direction of the injector.

Space does not permit a detailed analysis of these factors, and reference should be made for such detailed information to standard works on the subject.

A properly designed burner should, however, possess the following characteristics :—

(1) High maximum aeration of at least 2 air to 1 gas.
(2) Stability of flame over a wide range of working pressures.
(3) Ease of control.

High maximum aeration is desirable for most applications and essential to many. It gives a high temperature flame and can be used without preheating the secondary air for producing temperatures up to 1,400° C. with 500 C.V. gas. The flame can be brought into contact with relatively cold surfaces with only slight formation of carbon monoxide, and can be used in a confined space; this is important, as surface loss of heat by convection and radiation increases rapidly as the temperature rises.

Stability over a wide range of working pressures is particularly necessary where automatic devices are in use to control gas rate with temperature. Ease

of control is of especial importance in operations requiring burner adjustment.

These properties are achieved in practice by :—

(1) Reducing friction and turbulence in the mixing tube to a minimum.
(2) Correctly proportioning the sizes of mixture tube, burner ports and air ports.

The design of bar burners, Fig. 261, is a frequent necessity for such purposes

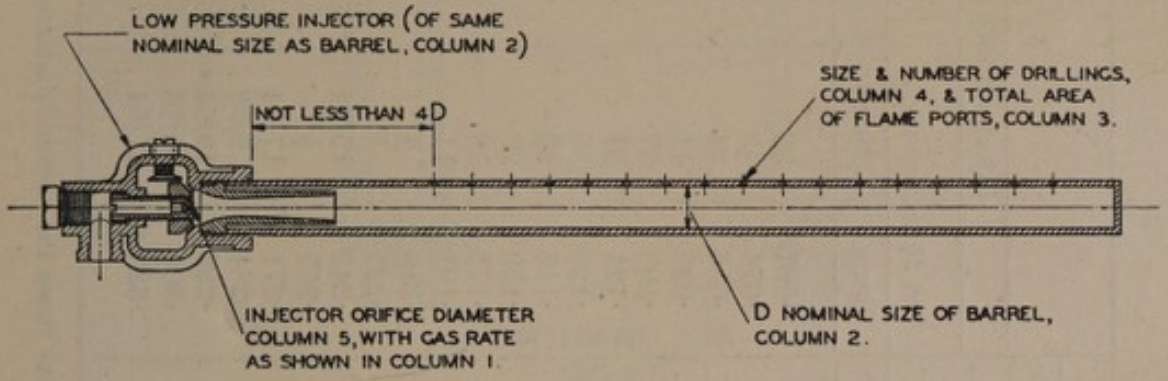

FIG. 261. Town gas bar burner.

as heating liquid tanks, ovens, etc., and the following information is given for this purpose.

The injector orifice diameter is derived from the formula :—

$$Q = 1658{\cdot}5\ Ak\sqrt{h/s}$$

where Q = discharge in cubic feet per hour.
s = specific gravity (air = 1).
h = pressure in inches w.g.
A = area of orifice in square inches.
k = orifice constant, varying between 0·7 and 0·9.

The sizes of the orifices shown in Table 116 are strictly applicable only if the channel length is from 1 to 1½ times the diameter and has an angle of approach of 120° (i.e. ordinary twist drill cutting angle), but the values given are within normal experimental error of all normal injectors.

The optimum value of burner throat = 0·1 sq. in./cu. ft./ hr.

For specific gravities other than 0·45, the diameter of the injector orifice needs to be altered as follows :—

Specific gravity ..	0·35	0·4	0·5	0·55	0·6
Percentage diameter	94·0	97·0	102·7	105·0	107·5

SPECIAL AERATED BURNERS

The Meker Burner. This is essentially a bunsen burner with high aeration of 3 : 1 or even higher. It is provided with a flame port designed to prevent back-firing which would occur if an open flame port were employed. The flame port used is generally a channel grid giving a number of small flames close together, resulting in a solid flame, the multiple cones being short and sharp and the hot oxidising zone extended. This principle is frequently used in ribbon bar burners for heating liquid tanks and for burners in very restricted spaces such as platten heating. For industrial purposes it is important that such burners shall be kept clean and that the burner head should not suffer distortion through overheating.

Semi-bunsen Jet. This jet, shown in Fig. 262, provides a means of building up burners of almost any size giving partially aerated flames. The jets are usually arranged in pairs, the two flames impinging forming an inverted Y. The

TABLE 116

Burner drillings and spacings												
1	2		3	4					5			
Gas rate cu. ft. per hour	Size of barrel		Total area of flame ports square inch	No. of drillings					Injector orifice diameter			
				3/32 inch	1/8 inch	5/32 inch	3/16 inch	7/32 inch	(a) at 2½ inches w.g.		(b) at ½ inch w.g.	
	Min^m inches	Opt^m inches		Max. permissible spacing :								
				3/4 inch	7/8 inch	1 inch	1⅛ inch	1¼ inch	inch	Morse	inch	Morse
10	3/8	1/2	0·15	22	12	8	5	4	·061	53	·090	43
15	1/2	3/4	0·225	33	18	12	8	6	·074	49	·111	34
20	1/2	3/4	0·3	43	24	16	11	8	·086	44	·128	30
25	1/2	1	0·375	54	31	20	14	10	·096	41	·143	27
30	3/4	1	0·45	65	37	23	16	12	·105	37	·157	22
40	3/4	1¼	0·6	87	49	31	22	16	·121	31	·181	15
45	3/4	1½	0·675	98	55	35	24	18	·128	30	·192	11
60	1	1½	0·9	131	73	47	33	24	·148	26	·222	2
70	1	1½	1·05	152	86	55	38	28	·159	20	·239	15/64
80	1¼	1½	1·2	174	98	63	43	32	·171	18	·256	1/4
100	1¼	2	1·5	217	122	78	54	40	·191	11	·286	9/32
120	1½	2	1·8	261	146	94	65	48	·210	4	·313	5/16
140	1½	2½	2·1	304	171	110	76	56	·225	1	·339	11/32
160	2	2½	2·4	347	196	125	87	64	·243	15/64	·362	23/64
180	2	2½	2·7	391	220	141	98	72	·257	1/4	·384	25/64
200	2	3	3·0	435	245	156	109	80	·271	17/64	·405	13/32
220	2	3	3·3	478	269	172	120	88	·284	9/32	·424	27/64
240	2	3	3·6	522	293	188	131	96	·296	19/64	·443	7/16
260	2	3	3·9	565	317	203	142	104	·309	5/16	·461	15/32
280	2½	3	4·2	608	342	219	152	112	·320	21/64	·479	31/64

Note. The distance between the injector and the first flame port should not be less than four times the nominal diameter of the barrel.

primary aeration is low, about 1·5 : 1, but by the process of mixing the flames a certain turbulence is created so that the effective flame length is not excessive. It is important that the jets should not be allowed to become overheated,

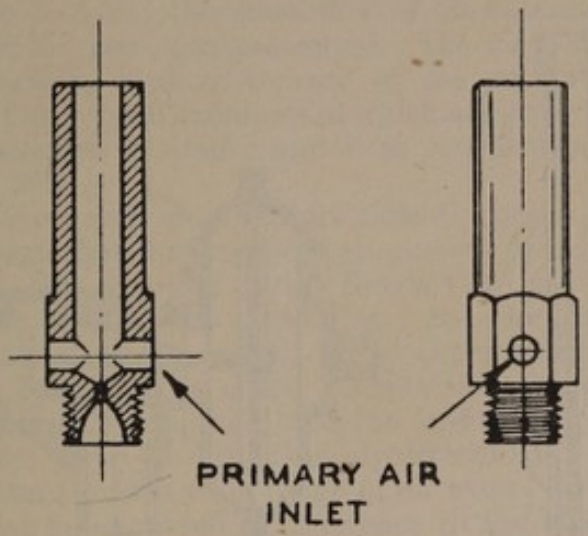

FIG. 262. Semi-bunsen jet—aerated burner.

otherwise corrosion and back-firing are liable to occur. The jets are made in six sizes giving gas rates between 4 and 10 cubic feet per hour at 2 inches water gauge.

HIGH PRESSURE GAS BURNERS

Where burners are required to operate in a very restricted space, such as a vulcanising press or a laundry iron, where there is little space for the removal of products of combustion or the introduction of secondary air, it is necessary to introduce all the air required for combustion primarily. This can be achieved by using gas at pressures up to 3 lb. per square inch (83 inches w.g.) and a modified injector, similar to the low pressure injector, but very carefully made to enable a high gas velocity at the injector to entrain the required amount of air. The burner ports for such burners require special nozzles with fine orifices, and they are frequently provided with retaining flames in order to keep the flame in contact with the flame port. Burners of this type have a wide application in internally heated soldering irons, platten heating and high temperature furnaces. The advantage of the high pressure system is that only one supply pipe is required while the chief disadvantage of the system is the initial and running cost of the compressors, and the limited stability range of the flame.

AIR BLAST BURNERS

As an alternative to high pressure gas burners there has been developed another method of producing air-gas mixtures in the theoretical proportions, using air under pressure and low pressure gas. The air pressures commonly employed do not exceed 1 lb. per square inch, so that the cost of compression is not as great as when a high pressure gas system at 3 lb. per square inch is employed, but it should be remembered that at least four times the volume of air must be compressed to the lower pressure. An advantage of the method, however, is that higher mixture pressures (up to 10 inches w.g. with air at 1 lb. pressure) enable high mixture velocities to be maintained and burner rates to be cut down to a fraction of the full gas rate without back-firing, while the volume of the flame is small, giving a high heating intensity. On the other hand, at full gas rates the high velocity may require special attention to burner

housings to prevent the flame from blowing off, and there is always the problem of flame erosion to be countered.

A " theoretical air-gas mixture " is highly explosive, and it is inadvisable to

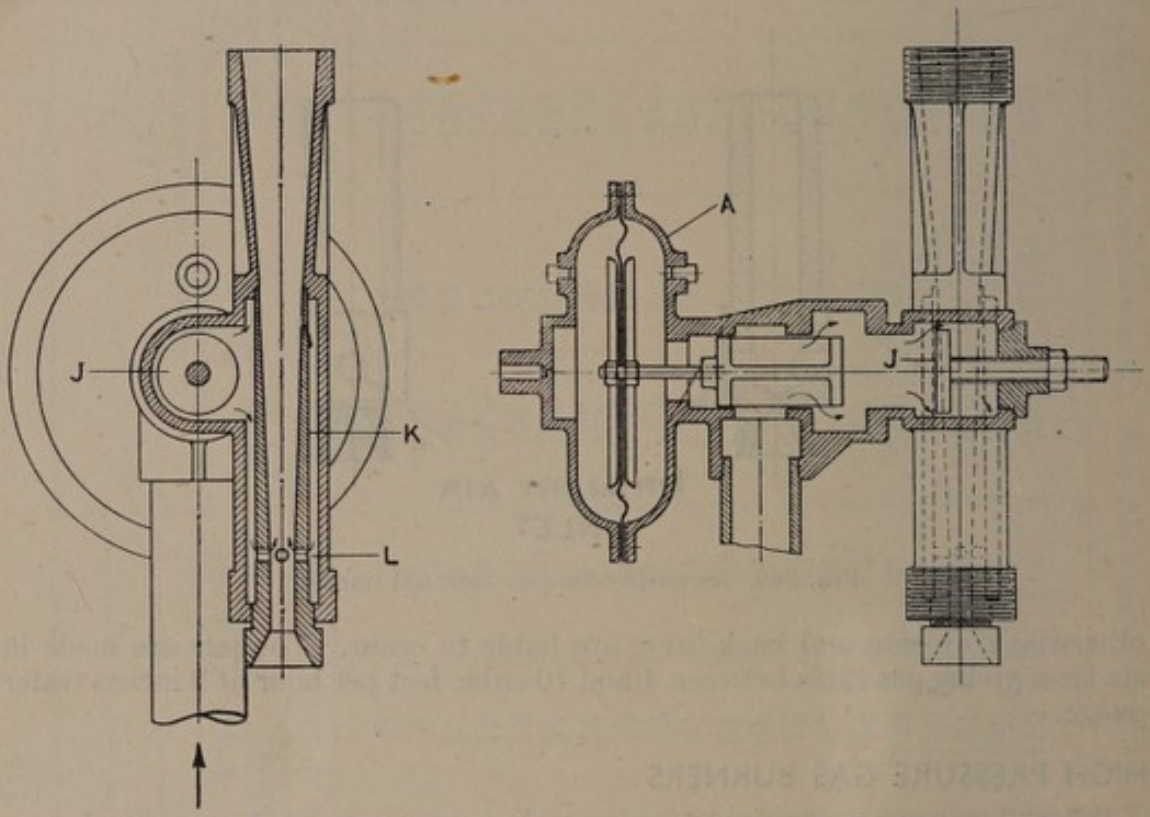

FIG. 263. Automatic air-gas proportioning device.

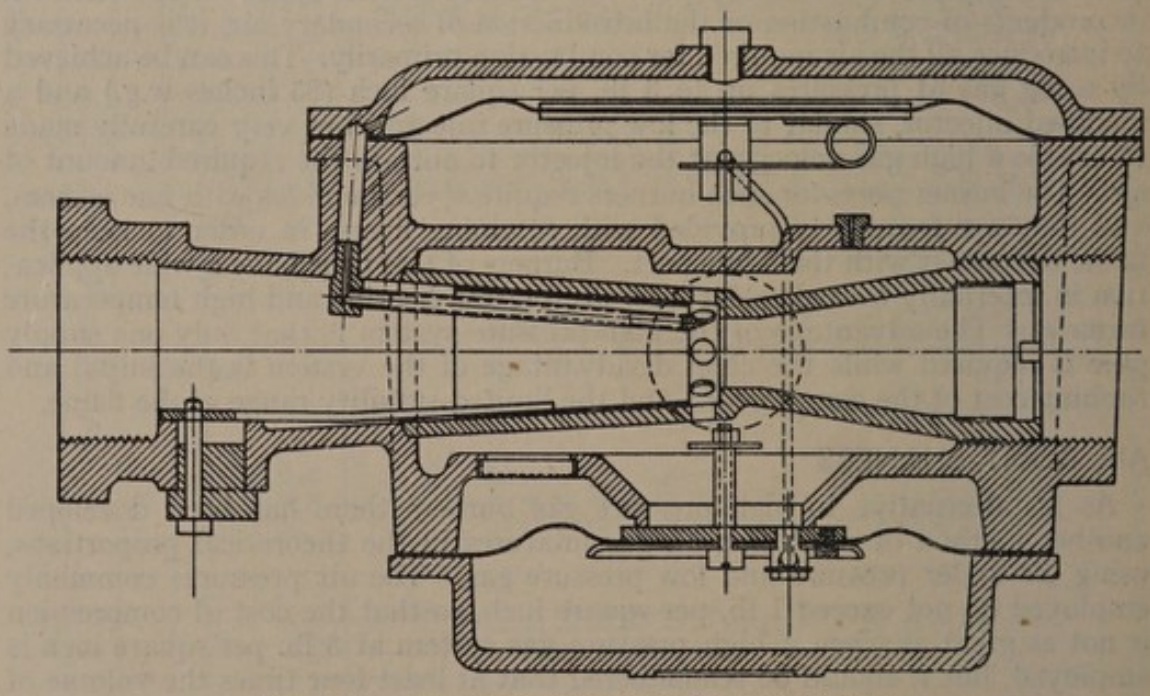

FIG. 264. Automatic air-gas proportioning device.

run such a supply more than a few feet, so that mixing is usually arranged to take place immediately before the burner.

The simplest air blast application is to pass the air through a tube surrounded by a concentric gas supply so that mixing takes place in the flame ; this is the

normal gas blow pipe, and the mixture and flame size are controlled by individual cocks.

If it is desired to pre-mix the air and gas, this is frequently done by means of a mixing tee or other suitable mixing chamber, the gas line being protected by a non-return valve to avoid the possibility of air entering the gas supply, a danger which must be rigorously prevented. The method is crude and is not recommended; instead, it is customary to use an automatic air-gas proportioning device, two typical examples of which are given in Figs. 263 and 264. Their mode of operation is similar and it is proposed to describe only that illustrated in Fig. 263.

The proportioner consists of a Venturi tube K perforated at the throat by holes L in such a way that the stream of air passing through the tube draws in a quantity of gas. The gas first passes through the governor A, the vertical diaphragm of which reduces it to atmospheric pressure, so that the amount of gas entering the holes L at the venturi throat depends entirely on the velocity of the air stream. The efficiency of the Venturi is so arranged that it will inject more gas than is normally required, and the air-gas ratio can be adjusted by means of a restrictor valve J. Once this adjustment has been made the air-gas ratio remains constant over a wide range of air rates. When using devices of this type care must be exercised by reference to the makers' published data that the size of the Venturi throat is appropriate to the size of the burner port openings, otherwise it will be found that mixture adjustment is not possible.

PROPORTIONING WITHOUT MIXING

The disadvantage of pre-mixing air and gas by means of the proportioning devices already referred to, is that on account of the danger of back-firing and

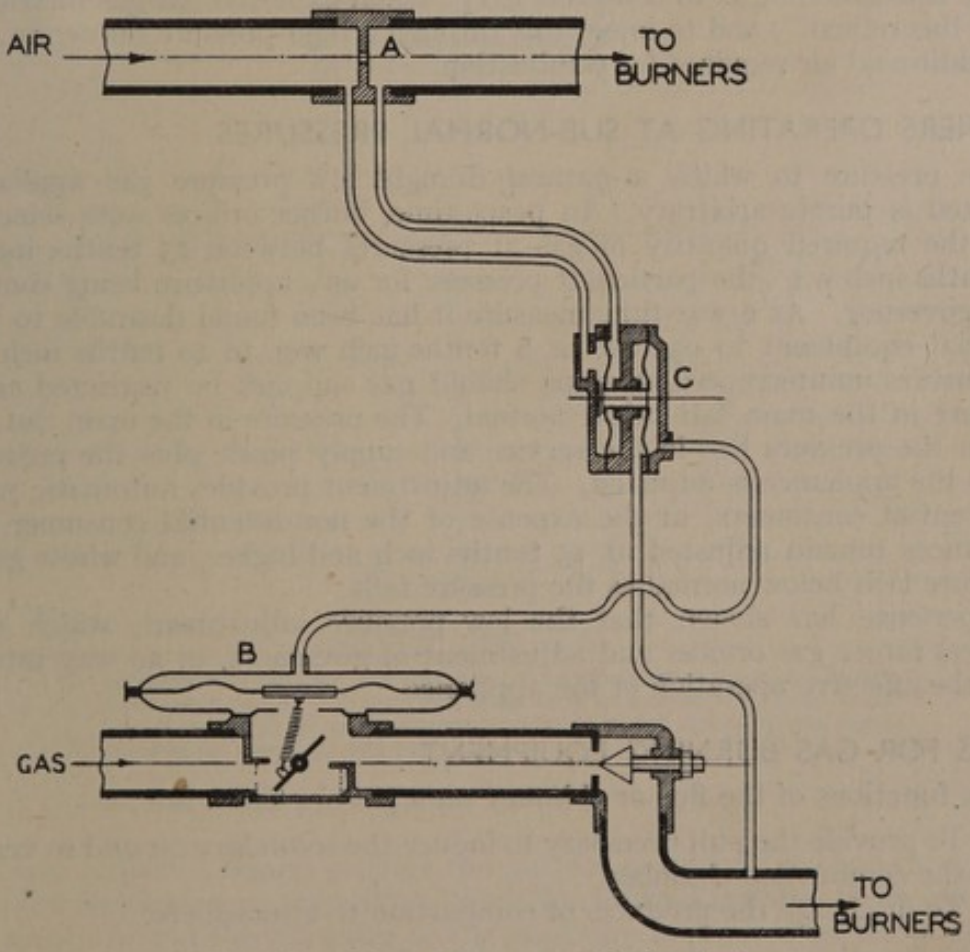

FIG. 265. Non-mixing proportioning device.

the difficulty of using hot air in the Venturi tubes, preheating is impossible. In order to overcome this difficulty the non-mixing proportioning device shown in Fig. 265 is used. The pressure drop across the orifice A operates the valve C, which controls the pressure loading on the gas pressure operated governor B. Increasing the air velocity increases the pressure difference across the orifice A, moving the control valve to the right, equalising the pressure difference across the Venturi and the loading pressure minus the back pressure from the downstream of the constriction in the gas line. By varying either of the two restrictions any desired air-gas ratio can be obtained.

BURNER STABILITY WITH AIR BLAST FLAMES

It has already been mentioned that the high velocity of air-gas mixtures using air blast may give rise to flame instability. This rarely causes difficulty, however, since the burners are usually directed into brickwork channels or actually on to brickwork ; the back pressure exerted when they are cold keeps the flame stable, while as soon as the immediate brickwork becomes red hot, a matter usually of only a few minutes, the flame is automatically established.

COMPRESSOR MIXING

Mention must be made of a further mixing system for air and gas. It consists of mixing air and gas, carefully proportioned, in a compressor and distributing the mixture. In some installations the " theoretical mixture " is used, but elaborate precautions are necessary to prevent explosions while each burner must be provided with a flame trap and fusible link safety cut-off. The application to immersion heating, however, has important possibilities.

The usual method is to compress a $1\frac{1}{2} : 1$ non-explosive air-gas mixture (i.e. not " theoretical ") and to inject this through a high pressure burner to obtain the additional air required for combustion.

BURNERS OPERATING AT SUB-NORMAL PRESSURES

The pressure to which a natural draught low pressure gas appliance is adjusted is purely arbitrary. In peace-time, burner orifices were selected to pass the required quantity of gas at pressures between 15 tenths inch and 30 tenths inch w.g., the particular pressure for any appliance being controlled by a governor. As a war-time measure it has been found desirable to adjust essential equipment to operate at 5 tenths inch w.g. to 10 tenths inch w.g. ; this ensures uninterrupted working should gas supplies be restricted and the pressure in the main fall below normal. The pressure in the main can fall to within the pressure loss in the service and supply pipes, plus the pressure to which the appliance is adjusted. The adjustment provides automatic priority to essential consumers, at the expense of the non-essential consumer whose appliances remain adjusted at 15 tenths inch and higher, and whose gas rate therefore falls below normal as the pressure falls.

Experience has shown that the low pressure adjustment, which simply involves larger gas orifices and adjustment of governors, in no way interferes with the effective operation of the appliance.

FLUES FOR GAS BURNING EQUIPMENT

The functions of the flue or chimney on a gas appliance are :—

(1) To provide the pull necessary to induce the secondary air and to ventilate the combustion chamber.
(2) To discharge the products of combustion to atmosphere.

These functions are, of course, those of a flue on any fuel burning appliance,

but there are important differences in the flue requirements for town gas products and, say, the products from a solid fuel installation, and it is essential that these different requirements shall be recognised. Natural draught

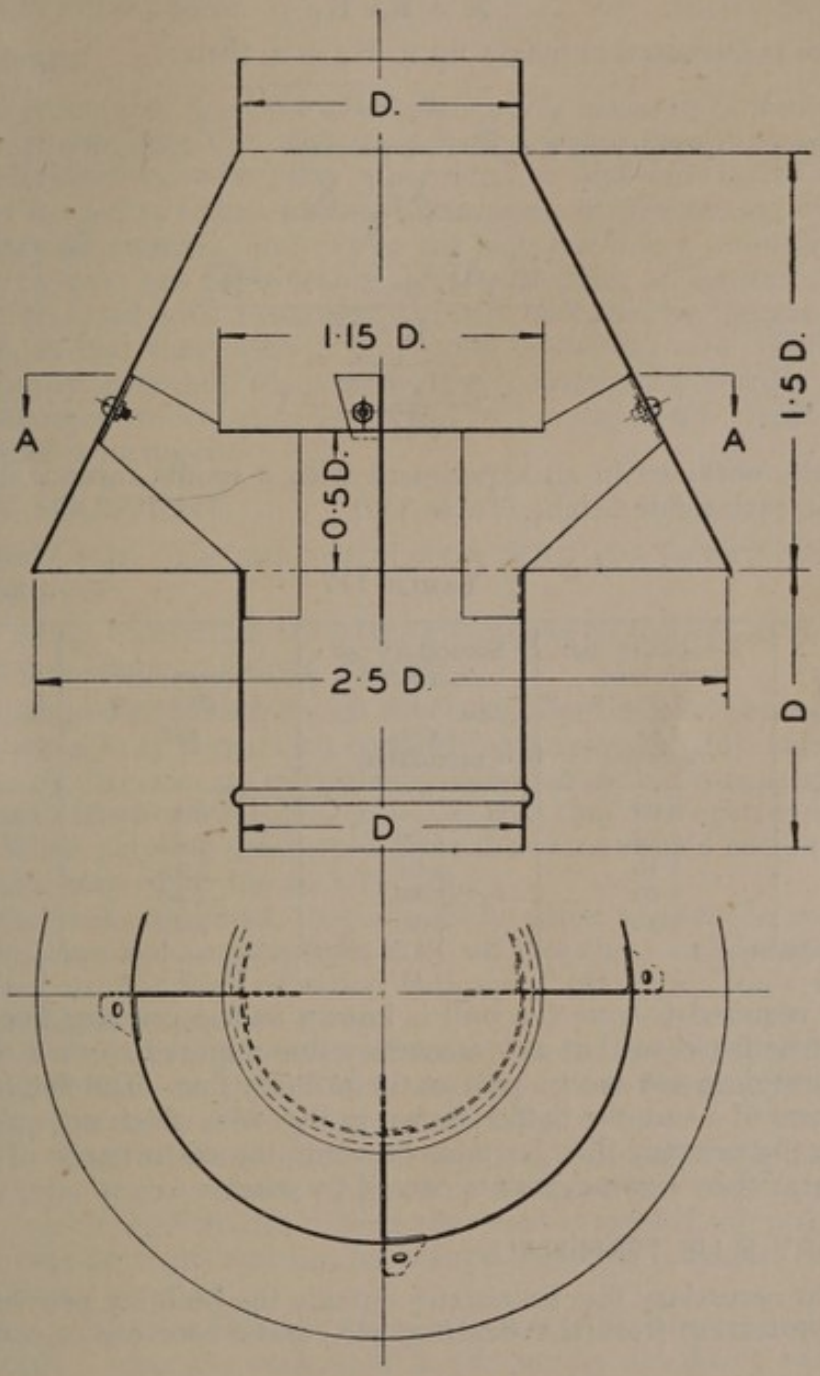

FIG. 266. Flue baffler.

appliances require a flue to provide the pull for secondary air, but rarely more than a few feet are necessary ; if too long a flue is fitted then too much secondary air is drawn in and the efficiency of the appliance is lowered by the increase in flue loss.

The effect of flue length may be deduced as a first approximation as follows :—

X = volume of secondary air entrained by the flue—cu. ft./cu. ft. gas.
H_1 = the height of the flue—ft.

then $$X = K\sqrt{H_1}.$$

If the flue is increased in height up to $H_1 + a$, then

$$X_2 = K\sqrt{H_1 + a}$$

the proportional increase in the air entrained

$$= \frac{X_2 - X_1}{X_1}$$

$$= \frac{K\sqrt{H_1 + a} - K\sqrt{H_1}}{K\sqrt{H_1}}$$

$$= \frac{\sqrt{H_1 + a}}{\sqrt{H_1}} - 1$$

The results obtained in an experiment with a muffle furnace illustrate the effect of increasing flue height (Table 117).

TABLE 117

Length of flue in feet	Secondary air entrained cu. ft./cu. ft. gas observed	Secondary air entrained cu. ft./cu. ft. gas calculated	Excess air cu. ft./cu. ft. gas	Flue loss per cent.
1	2·90	—	0·9	42
2	4·15	4·09	2·15	47
3	5·15	5·01	3·15	51
4	5·60	5·80	3·60	56
5	6·36	6·50	4·36	58

The flue required to give the pull is known as the primary flue, and a flue break must be fitted so that any secondary flue required to vent the products to atmosphere does not exert a pull on the primary flue. The flue break should be in the form of a suitable baffler shown in Fig. 266, which not only limits the flue pull to the primary flue, but also prevents the disturbance of combustion in the apparatus by down-draughts caused by wind.

SECONDARY FLUE TERMINALS

Where the secondary flue terminates outside the building provision must be made for protection from the weather, etc., and a cone cap or cowl is usually fitted.

An alternative method of discharging the products to atmosphere by forced ventilation is frequently employed in which the primary flues terminate under a hood from which the products are exhausted by a fan. This method ensures positive ventilation and freedom from wind effects, while no baffler is required on the primary flue.

AREA OF FLUE

It is the usual practice to allow 1 square inch of flue area for every 5,000–7,500 B.Th.U. per hour maximum input (10–15 cubic feet per hour of gas consumption) depending on the furnace design.

FLUES FOR AIR BLAST EQUIPMENT

Air blast equipment does not depend on flue pull to provide secondary air for combustion, and a flue is provided simply to discharge the products of combustion to atmosphere.

FLUE DAMPERS

When an adjustable flue pull is necessary it is usual to fit damper control. This does not eliminate the need for a baffler which should always be fitted when an individual secondary flue is provided to vent directly to atmosphere. The damper is used to adjust furnace atmosphere in conjunction with primary and secondary air controls, and to prevent high flue losses when the furnace is working at by-pass gas rates. Large gas consuming appliances, particularly boilers, can be fitted with automatic dampers operated by the gas pressure at the burners, so that they open in proportion to the gas rate. Furnaces fitted with electrically operated temperature gas controls are usually fitted with automatic dampers which close as the gas rate is cut down. Such devices are important fuel economisers.

AUXILIARY EQUIPMENT

Gas Control Cocks. The gas control cocks fitted in a factory can be divided into two groups :—

(1) Main cocks, controlling the whole gas supply to sections of the factory.
(2) Appliance cocks on individual units.

Both groups should be of such design that they offer the least possible resistance to gas flow when fully open, such cocks being known as " full bore " cocks.

Group (1). Main cocks are infrequently operated so that care should be taken by regular maintenance inspection to ensure that they will open and close when required. When painting gas supply pipes the cocks should not be painted, as a film of paint may cause the cock to jam.

When lever cocks are fixed, they should be fitted so that the weight of the lever will not turn a loose plug from the " off " to the " on " position. Levers must be fitted so that their direction indicates " off " and " on " setting, and each cock should have its own lever.

Gate valves with wheel hand control should be clearly marked with the " off " and " on " directions, so that if an emergency arises there shall be no doubt as to whether the valve is being opened or closed.

Group (2). Cock controls fitted on appliances should work smoothly and be given regular maintenance ; they must be kept free both internally and externally from foreign matter ; particular care is needed when fitted on salt baths, wax heating tanks and similar plant. The best lubricant for cocks is a grease containing graphite.

As a safety precaution it is always wise to use cocks which cannot be turned on accidentally ; also any cock used in gas supply should be provided with positive stops at 90° so that " off " and " on " position are quite definite.

TEMPERATURE CONTROL

Not all processes lend themselves to automatic temperature regulation, on the other hand others require strict adjustment. From the point of view of ease of control and to a great extent of fuel economy, automatic temperature regulation should be adopted whenever possible.

With gas-fired equipment automatic temperature control is simple and the equipment cheap and reliable.

Thermostats have been discussed in Chapter XXVI. Their application to gas-fired equipment only will be here considered.

DIRECT ACTING THERMOSTATS

Fig. 267 shows a conventional bi-metal thermostat applied to a gas-fired appliance.

When below the required control temperature the non-expanding rod forces

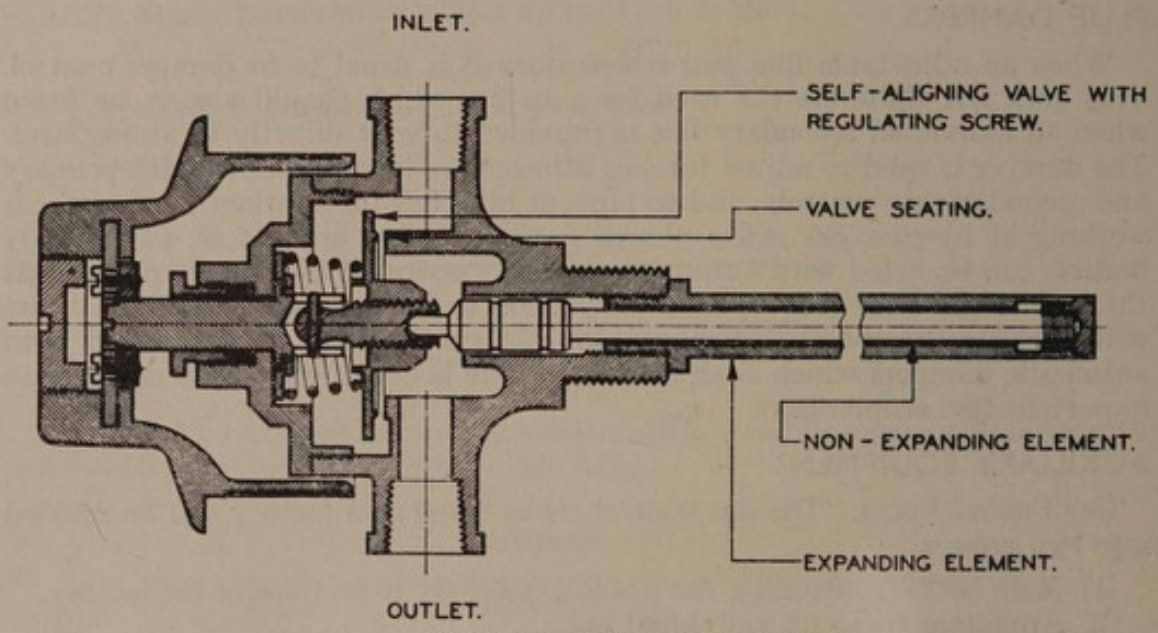

FIG. 267. Bi-metal thermostat.

the valve away from its seat against the spring. On heating, the expanding element—a brass sheath—expands more than the non-expanding element which, being free to slide, permits the spring to close the valve, shutting off the

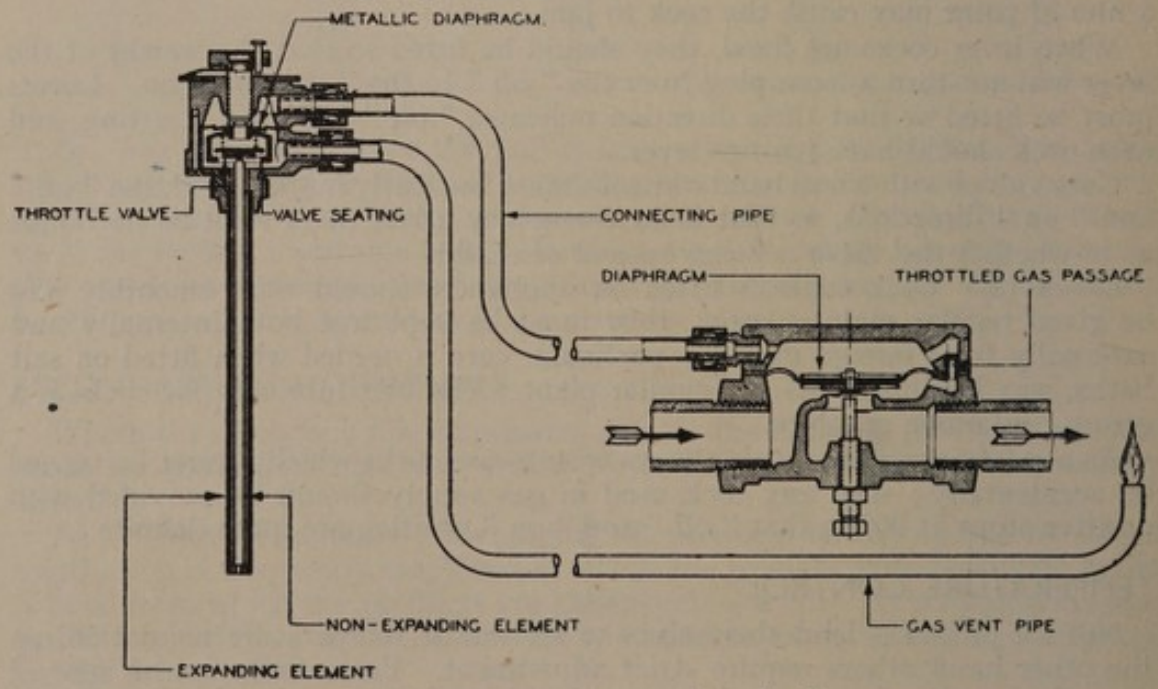

FIG. 268. Thermostat in relay circuit.

gas supply to the burner. A small by-pass is fitted so that the burners are not extinguished, and on cooling, the valve opens and restores the full gas rate. Direct acting thermostats of this type have a high resistance to gas flow and are used only for low gas rates up to 100 cubic feet per hour as for rates higher than

this they tend to become bulky. The difficulty is easily overcome by using the thermostat in a relay circuit as shown in Fig. 268.

INDIRECT ACTING THERMOSTATIC CONTROL

The relay valve shown in Fig. 268 has a connecting weep between the main gas passage and the upper chamber, which is sealed except for a gas passage through the thermostat. If the thermostat is open the pressure above the diaphragm is lower than below, and the valve is raised allowing gas to flow in the direction of the arrow. If the thermostat is closed, pressure builds up above the diaphragm by passing through the weep until the weight of the valve causes it to fall and shut off the gas, a by-pass being provided to prevent the valve closing completely. The relief connection from the thermostat can be connected as shown giving a balanced equilibrium condition with gradual reactions and control or it can be left open to atmosphere to give snap action control.

Some features of indirect thermostats are :—

(1) They can be used for any gas consumption without appreciable loss of pressure.
(2) They enable greater rate of heating up since control only operates when the thermostat is shut completely.
(3) They enable higher recovery rates after a cooling off period.
(4) They can be used in connection with time controls and safety cut-offs which are fitted in the outlet weep of the thermostat.

In selecting and using a thermostat of this type the following points should be considered :—

(i) The detecting rods must be of material that will withstand the temperature required without fatigue or permanent physical change.
(ii) The rod should be protected by a suitable sheath if there is any danger of corrosion or mechanical injury.
(iii) The thermostat must be placed in a truly representative position, the length of stem, etc., being chosen for this purpose.

In practice the maximum temperature for such thermostats of these types is 900°–1,000° C., but if the temperature may exceed 500° C. the stem may bend unless fitted in a vertical position.

THERMO-ELECTRIC THERMOSTATS

In such systems the detector is a thermo-couple which operates a milli-voltmeter calibrated for temperatures against the thermo-element used. An adjustable contact in the milli-voltmeter enables a circuit to be closed when the indicator needle reaches the pre-set temperature, and the current operates a reversing electric motor system controlling the gas valves. Such systems can be elaborated to operate dampers, air controls and warning signals, as desired.

Thermo-couple control can be used for temperatures up to 1,400° C.

PRESSURESTATS

Thermostats cannot be used directly for steam boilers, and control of gas rate with steam requirements is achieved by means of a pressurestat ; this is simply a gas valve arranged to close by steam pressure against an adjustable spring loading, a by-pass being provided to maintain sufficient gas to keep the burners alight.

PILOTS

When an appliance is automatically controlled, a fixed pilot fed from the inlet side of the gas controls, including the main cocks, must be provided.

SAFETY CUT-OFFS

Gas appliances fitted with automatic controls, such as steam raising plant drying ovens, etc., should be provided with a safety device which will shut off the gas supply if the burner or lighting pilot fails to ignite. This is a most important safety precaution, as gas accumulating in combustion chambers and flues may lead to an explosion when an attempt is made to light the burner.

It is usual to make use of a thermal effect produced by the pilot flame to operate such devices. The simplest is a bi-metal element operating the main gas valve, the valve being closed when the element is cold and open when hot. This device can be used in the relief weep from a relay valve giving indirect control. By the development of suitable solenoids the thermo-electric effect of heating the junction of two dissimilar metals is made use of to hold the main gas valve open against a mechanical load ; if the pilot is extinguished no current passes through the solenoid and the valve closes. The current generated is insufficient to open the valve which must be hand set, ensuring personal attention and additional safety. If desired, the thermo-generated current can be made to operate through a relay controlling motorised or magnetic valves —thus eliminating manual control.

PRESSURE CUT-OFFS

If an interruption of the gas supply takes place and the pressure falls unduly, burners may be extinguished ; if the gas pressure is then restored apparatus could become filled with gas and a potentially dangerous condition result. To prevent this danger a type of gas cut-off valve known as a pressure cut-off can be fitted. With this cut-off if the gas pressure falls below the safety limit, the gas is cut completely off and cannot be restored until the valve is re-set by hand.

BACK-PRESSURE VALVES

When air pressure is used in conjunction with town gas, provision must be made under section 25 of the 1934 Gas Undertakings Act to prevent air passing back into the gas supplies. The gas undertaking must approve the precautions taken, and their advice should, therefore, be asked before completing the installation of the equipment.

METERS

Meters fitted in industrial premises are usually of the conventional dry type familiar to most people, but necessarily of larger capacity.

Usually they are fitted with top connections up to a capacity of 700 cubic feet per hour, and above this are provided with flanges on the sides. It is common with the smaller sizes to make the connections between the adjacent service and outlet pipes and the meter in lead piping with " blown " joints. This avoids straining the case and makes for ease when the meter requires to be exchanged, but connection in wrought iron or mild steel tubing is sometimes seen and in present conditions is frequently justified to save lead piping, brass unions and solder, and the practical skill required to make " blown " joints satisfactory.

Meters over 700 cubic feet per hour capacity, with side flanges, are connected by mild steel tubing and fittings, and the provision of a by-pass may be necessary. It is desirable to obtain from the local undertaking a sketch giving details of the method of fitting approved by them. An important point to consider when connecting large meters is the need to provide for some small variation in dimensions if exchange is necessary. This can be provided for laterally in the side connectors, and packing pieces underneath take care of a variation in

flange height as well as ensuring that the base is not in direct contact with a damp floor.

Wet meters are much less used than formerly, but a new high capacity type using oil as the sealing medium is being employed in larger numbers and is made in standard sizes to pass volumes from 7,000–80,000 cubic feet per hour.

Inferential meters are not generally approved by the Board of Trade as a basis of charging for the gas supplied by an undertaking, but are useful as " check " meters.

Meters of 700 cubic feet per hour capacity and above are fitted with oiling devices. Some devices are automatic in action and merely require oil to be added to a reservoir periodically, but the majority are oiled by means of a high pressure oil syringe applied to nipples situated at the front of a meter.

Meters should never be allowed to become noisy by neglect of lubrication.

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CHAPTER XXVIII

OIL AND ITS UTILISATION

Nature of petroleum—Distillation and cracking—Fuel oils—The utilisation of fuel oils ; pre-treatment ; burning technique ; burner maintenance—Criteria for fuel oils ; testing.

THE general properties and method of utilisation of oils derived from coal tar were described in Chapter II. In this present chapter will be given some account of petroleum oils, including the methods by which they are burnt in industrial practice. The information here given on utilisation is also generally applicable to coal tar oils, and should be read in conjunction with Chapter II.

THE NATURE OF PETROLEUM

Mineral oil as it issues from its subterranean formations is a mixture of solids, liquids and gases. The gas ("natural gas") is, in the main, methane (CH_4), the first member of the paraffin series, but also present to a lesser extent are ethane (C_2H_6), propane (C_3H_8), the butanes (C_4H_{10}), together with small amounts of hydrogen sulphide (H_2S), carbon dioxide, nitrogen and, in some few oils, helium. Typical analyses of natural gas are given in Table 118.

TABLE 118

	A dry gas from Pittsburgh	A West Virginian "wet" gas	A Middle East gas after removal of gasoline
Specific gravity (air = 1) ..	0·65	1·38	1·03
Analysis :—			
Methane .. per cent.	84·7	18·0	40
Ethane .. .. ,, ,,	9·4	—	21
Propane .. ,, ,,	3·0	—	18
Butanes .. ,, ,,	1·3	78·7	9
Nitrogen .. ,, ,,	1·6	3·3	—
Carbon dioxide ,, ,,	—	—	—
Hydrogen sulphide ,, ,,	traces	—	12

Natural gas contains a greater or lesser quantity of liquifiable constituents, as will have been evident from Table 118. These form the so-called "natural gasoline." This is removed by simple compression of the gas, by means of a solvent, or by adsorption as, for example, in active carbon. Natural gasoline is a very light spirit and may be incorporated in straight-run petrol from the distillation units ; it is characterised by high volatility and considerable anti-knock properties. Furthermore, the lower paraffin gases (except methane and ethane) are readily condensed into liquids by moderate pressure. The resulting liquids can be stored in cylinders and form an increasingly important source of gas of high calorific value.

The hydrocarbon constituents of petroleum from all and every source fall into three main classes :—

Classes	Generic formula	Example
Paraffins	C_nH_{2n+2}	Butane, C_4H_{10}
Aromatics		Benzene, C_6H_6
Naphthenes	C_nH_{2n} to C_nH_{2n-20}	Cyclohexane, C_6H_{12}

Petroleum must, therefore, be considered as built up of numerous members of these three main groups of chemical bodies, but there is the further com-

plication that small amounts of sulphur, nitrogen and oxygen derivatives are frequently present, together with some mineral ash, though the ash content is usually extremely small. It can readily be seen that the isolation of any particular hydrocarbon may be extremely difficult, and perhaps impossible.

In effect the petroleum refineries separate crude oil by distillation into :—

A. A range of motor gasolines (petrols) of various degrees of volatility.
B. A range of solvent spirits—turpentine substitutes and paint compounds.
C. A range of burning oils (so-called paraffin oils and kerosines) from the familiar household illuminant to lighthouse and signal oils.
D. A range of gas oils for the enrichment of water gas and a most valuable fuel for the compression-ignition engine.
E. A range of lubricating oils from the lightest spindle oil to the heaviest steam cylinder oil.
F. A range of solid paraffin waxes from the lower melting material for water-proofing paper, etc., to the high melting component of candles and night-lights.
G. A range of bitumens or pitches for road surfacing and road aggregates, and for general water-proofing purposes, such as the impregnation of felts.
H. Distillates and residual oils from heavier fractions, used as fuel oils.

THE DISTILLATION OF PETROLEUM

Distillation is the obvious way of separating mineral oil into its broad and basic groups of components. It is also the oldest method of refining. In the earliest days of the industry, the segregation of burning oil fractions was the first to be undertaken. The simple act of boiling crude oil in a kettle or still and condensing the vapours was practised many years ago. Clearly such a crude method was ineffective if it was desired to make a clean cut. So next was devised a sequence of connected stills each heated to a higher temperature than the one before, and from each successive unit of the series there was derived a fraction of higher boiling point.

Some forty years ago the light (petrol) fractions were merely a nuisance, the desired product being lamp oil. With the greater and greater demand for motor and aviation fuel (particularly around 1914–18) and the lesser call for kerosine, stills began to be equipped with fractionating devices whereby the various groups of hydrocarbons might be better separated. Amongst these was the bubble cap tower which was already in use in the alcohol and coal tar industries.

In a modern distillation unit capable of handling 3 million tons a year (or 3 million gallons a day) of crude petroleum, the industry has progressed from the days when distillation meant nothing more than evaporation and condensation, to the complete elimination of re-distillation and to the perfection of plant that by means of accurate operation and control achieves a sharp separation of the various commercial grades of mineral oil derivatives.

This result is due to scientific combination of the various components—the pipe heater, the primary flash tower in which waste heat is economically used to remove from the crude oil its lightest components, the main fractionating column in which the various grades of gasolines, white spirits, kerosines and gas oils are cleanly and effectively separated, and finally a high vacuum system in which heavy oils and waxes, lubricating oils and the like, are distilled off from the ultimate residue of bitumen.

Furthermore, the physical separation of hydrocarbons of various boiling points and molecular weights can be extended to the lightest components and, as has been indicated, can be applied even to the segregation of the lower

gaseous hydrocarbons. Modern distillation methods applied to mineral oil effect a broad differentiation between its multitudinous components and effectively group them into commercially usable materials. Many of these groups, of course, need chemical and physical refining before they can be used industrially.

The refining methods applied to petroleum fractions will not be here described because the distillates and heavy residual fractions used as fuel oil are generally in no need of treatment. It seems possible, however, that special cuts used in the compression-ignition engine will require treatment at some future time to improve their performance in the engine. Some heavier oils are treated to remove solids and ash, while diesel oils are often treated to remove naphthenic acids.

THE CRACKING OF PETROLEUM OILS

It has long been known that mineral oils suffer serious decomposition when raised to high temperatures, and in the limit are decomposed into their elements —carbon and hydrogen. By the middle of the last century oil gas had been made, and gas oil had been cracked into the lighter kerosine. The interesting fact was discovered in Russia that aromatic hydrocarbons (benzole, toluole, and the like) could be produced by thermal treatment of heavy oil.

Decomposition is not by any means limited to the formation of lighter material. Cracking implies to a greater or less extent the simultaneous production of gas, petrol, heavy oil, pitch and carbon, and this is true whatever is the initial material. Even if this is so simple a substance as methane, there are obtained yields of hydrogen, benzole and other aromatics, pitch and carbon. Thus cracking, which is better termed pyrolysis, implies simultaneous simplification and complication—cutting down and building up. To use a convenient simile, pyrolysis begins by pulling down the hydrocarbon structure into " chemical bricks," and these are then used to form new structures.

High temperatures are not always essential. Even in the simple distillation process some cracking occurs, and it is to avoid this that vacuum is used to reduce the temperature in the still. The following effects occur in the temperature ranges indicated :—

Between 400° and 500° C. and at appropriate pressures, the products are normally and mainly material of lower boiling point.

Above 500° C. gases tend to be in excess and up to 700° C. are the main product.

Up to 1,000° C. the " chemical bricks " form the aromatic structure, whilst in the limit carbon and hydrogen are the resultants.

CRACKING PLANT

The essential components of a typical cracking plant are a pipe heater and a reaction chamber in which the products can remain for a time sufficient to allow the reactions to proceed to the required extent. Not all systems embody a reaction chamber. The velocity of the oil through the pipe heater is great enough to prevent any local overheating and deposition of carbonaceous matter. Turbulent flow is essential. After the time factor (the so-called " soaking coils " provide this in sequence to the main pipe heater) has operated the product is passed to separators where a pressure drop occurs and heavy residues are removed ; the vapours then pass to fractionating columns in which the usual division takes place, petrol passing overhead, and a gas oil residue going back to the plant as re-cycling stock. In passing it may be mentioned that even straight-run petrol is to-day a convenient cracking stock and its product distinctly better than the original petrol from the anti-knock point of view.

FUEL OILS

For the present purpose the main emphasis must be placed upon fuel oils, and upon the problems arising from their use for steam raising, and for furnaces. In this general sense "fuel oil" includes distillates of the gas oil type and residual and heavy oils from the processes just described, as well as blends of distillate and residual oils.

The correct use of fuel oils involves many important factors. The oils are frequently too viscous (cf. Chapter IX) at ordinary temperatures to flow to the burner or to atomise satisfactorily; they must then be heated to the right temperature to bring them into the desired condition. Attention to the burner is necessary, and following that, attention to the combustion.

It should be made clear at this point that the basic principles of combustion and furnace operation described in other parts of this text-book are equally applicable to oil burning; here particular attention is devoted to special considerations which relate to the burning of oil fuel.

OILS AVAILABLE FOR FUEL PURPOSES

During the war the resources of the oil industry and of the coal tar industry have been pooled. The various grades of tar oils available have been described in Chapter II; reference will here be made only to creosote/pitch mixture, which is referred to in Chapter II as C.T.F. 200.

Table 119 shows the approximate specifications of pool grades of petroleum fuel oils.

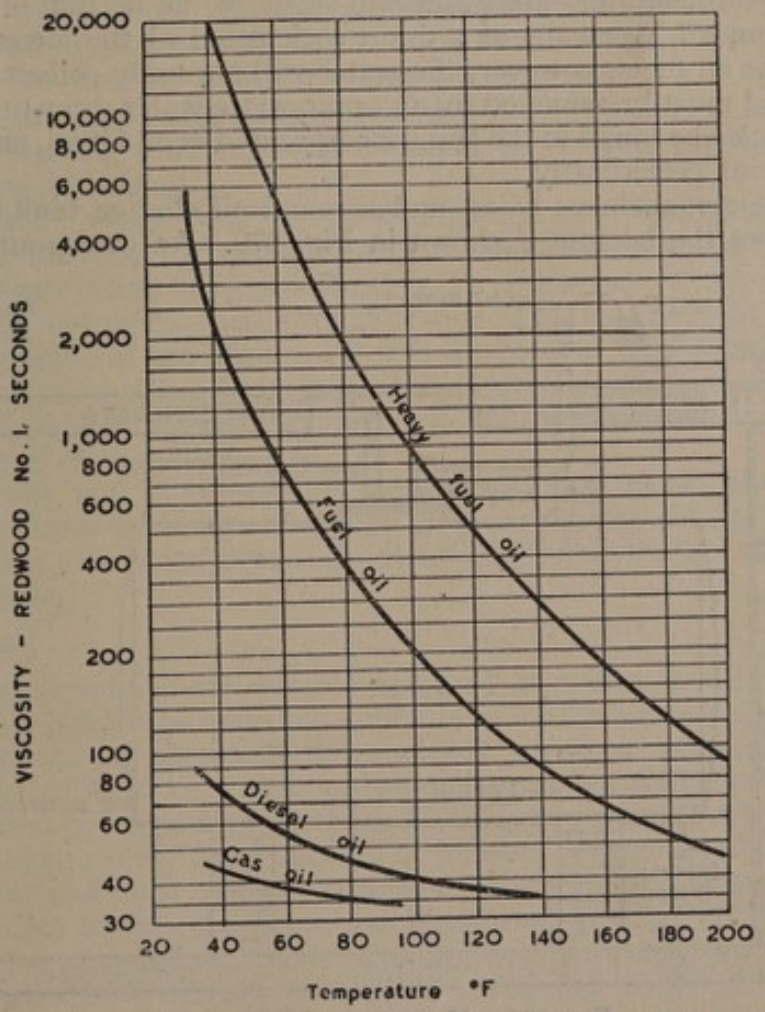


FIG. 269. Approximate viscosity curves of pool gas, diesel and fuel oils.
(N. L. Hudson, R. J. Bressey and T. C. Bailey, *J. Inst. Fuel*, XIII, 80.)

TABLE 119. APPROXIMATE SPECIFICATIONS OF POOL GRADES

	Pool gas oil	Pool diesel oil	Pool fuel oil	Pool heavy fuel oil
Viscosity Redwood I at 100° F.— secs.	35	40	220	950
Pour point (maximum)	20° F.	20° F.	35° F.	—
Gross calorific value, B.Th.U./lb.	19,600	19,350	18,900	18,750

The effect on their viscosity of heating these oils is illustrated in Fig. 269; the curves here given may not be exactly accurate for every delivery of oil, but they show the order of magnitude of the effect.

PRE-TREATMENT OF LIQUID FUEL OILS

Oil is brought to the works by rail tank car, road tank waggon or barge. It is then unloaded into the main storage tank. Care is needed in this operation to avoid spillage, and proper equipment and supervision are necessary.

In storage the oil may collect water and this must be removed at regular intervals. Water in oil may delay combustion and produce heavy smoke. When present in large amounts it may cause the flame to leave the burner nose and either be totally extinguished or burn at the back end of the furnace and in the flues.

In most petroleum oils the water will settle to the bottom of the tank, and should be run off by means of a drain cock fitted at the lowest point of the tank. In tar oil fuels, however, the water will normally collect on the surface of the oil and must be removed by an appropriate water draw-off arrangement. Water may also be found in the traps incorporated in oil filters, and these should be cleaned out periodically.

A suitable arrangement of an underground oil storage tank which permits drainage from the bottom is shown in Fig. 270. An overground tank would

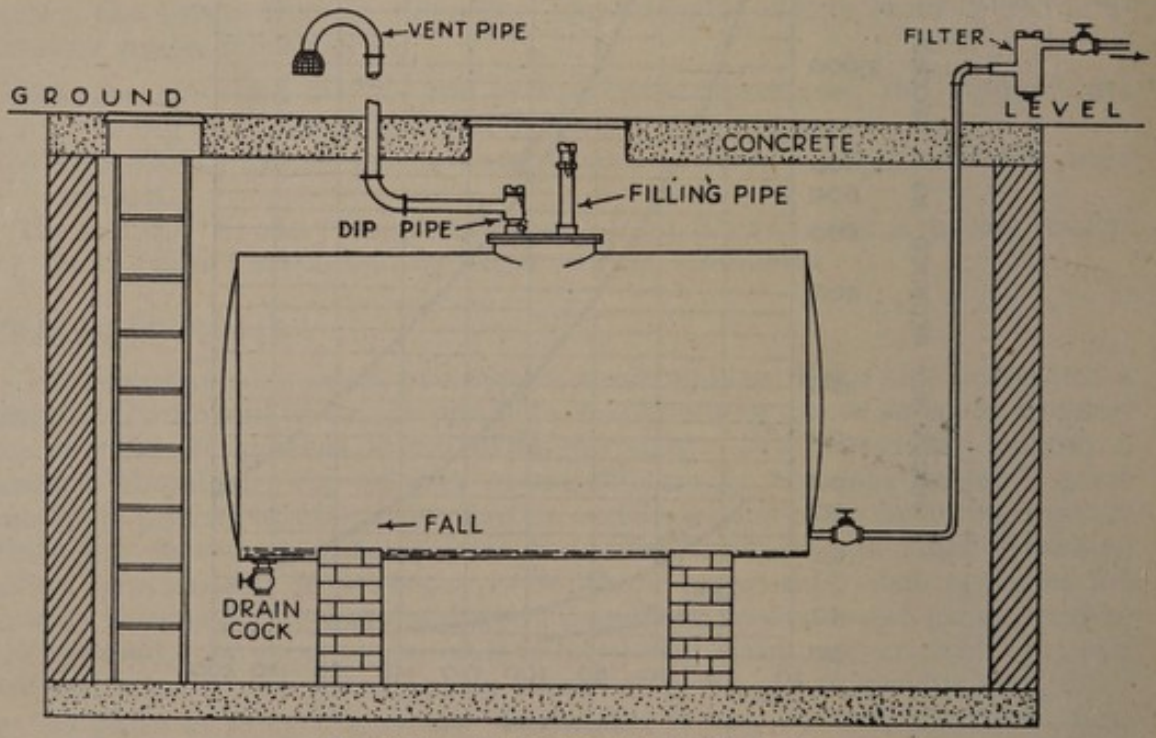


FIG. 270. Underground storage tank.
(N. L. Hudson, R. J. Bressey and T. C. Bailey, *J. Inst. Fuel*, XIII, 97.)

be exactly similar except for such modifications as are involved in the run of the pipes.

The heating of liquid fuels is usually carried out in two stages :—

(1) In the main storage tanks ; the oil is here raised to such a minimum temperature as will ensure a proper flow to the pumps, service tanks, etc.

(2) Nearer the burners ; the oil is in this stage raised to a sufficiently high temperature to ensure good atomisation by the burner.

The amount of heat required in stage 1 depends upon the grade of oil, the size, arrangement and exposure of the storage tanks and pipe-work, and the general layout of the plant. The temperature required in stage 2 depends mainly upon the grade of oil, the type of burner in use and the application. For the grades described in Table 119, the temperatures suggested are given in Table 120.

TABLE 120. PREHEAT TEMPERATURES FOR VARIOUS OILS

Grade	Stage 1	Stage 2
Pool gas oil	Atmospheric temp.	is normally sufficient
Pool diesel oil	do.	do.
Pool fuel oil	45° F.	120°/140° F.
Pool heavy oil	70° F.	175°/200° F.
Creosote/pitch mixture	80°/90° F.	160°/200° F.

The temperatures given for stage 2 in Table 120 are generally applicable to burners of the medium and low pressure air, and steam jet types ; with the pressure jet burner a greater degree of preheat is usually required. Whatever the type of burner, however, these temperatures are necessarily only a rough guide and the exact temperature which gives the best results on the individual oil-burning unit in practice should be determined by experiment under operating conditions.

It is preferable, where possible, to arrange for the heating to be thermostatically controlled and thermometer pockets should be arranged in the system to allow the temperature to be watched.

The oil, as has been explained, should be heated to just the right temperature for proper operation of the burner. If it is overheated trouble may be caused through vaporisation and coking. On the other hand, too low a temperature will lead to difficulties in getting the oil to the burners and in securing the correct degree of atomisation.

Petroleum oils and creosote/pitch mixtures should never be mixed as a deposit will be formed. If for any reason it is necessary to change from the one fuel to the other, all tanks, pipe-lines, etc., should first be thoroughly cleaned out with creosote.

OIL PIPE-LINES

The sizes of pipe-lines can be calculated as described in Chapter IX. In general they should not be less than 1 inch bore for the heavier grades of oil.

Oil lines should be placed in the warmest positions, preferably near the furnaces or boilers, and kept clear of draughts, or alternatively in trenches in the floor covered by protective plating. Hot oil lines should be lagged.

When a comparatively large number of burners is distributed throughout a works, the best and most efficient arrangement is usually a ring main system. A typical system of this character was illustrated in Chapter II. The oil is drawn from the main storage tank by a power pump, delivered through the heaters and circulated through an oil line round the works and back to the storage tank or to the suction side of the pump. Tappings are taken from the ring main to the various burners.

A circulating system of this sort permits of accurate control of oil pressures and temperatures, and of reduction in the sizes of pipes and fittings with a resultant economy in capital outlay, compared with gravity feed systems. If the oil in the mains and branches is likely to cool to too low a temperature during shut-down periods the whole system should be drained, and suitable cocks and vents should be installed for this purpose.

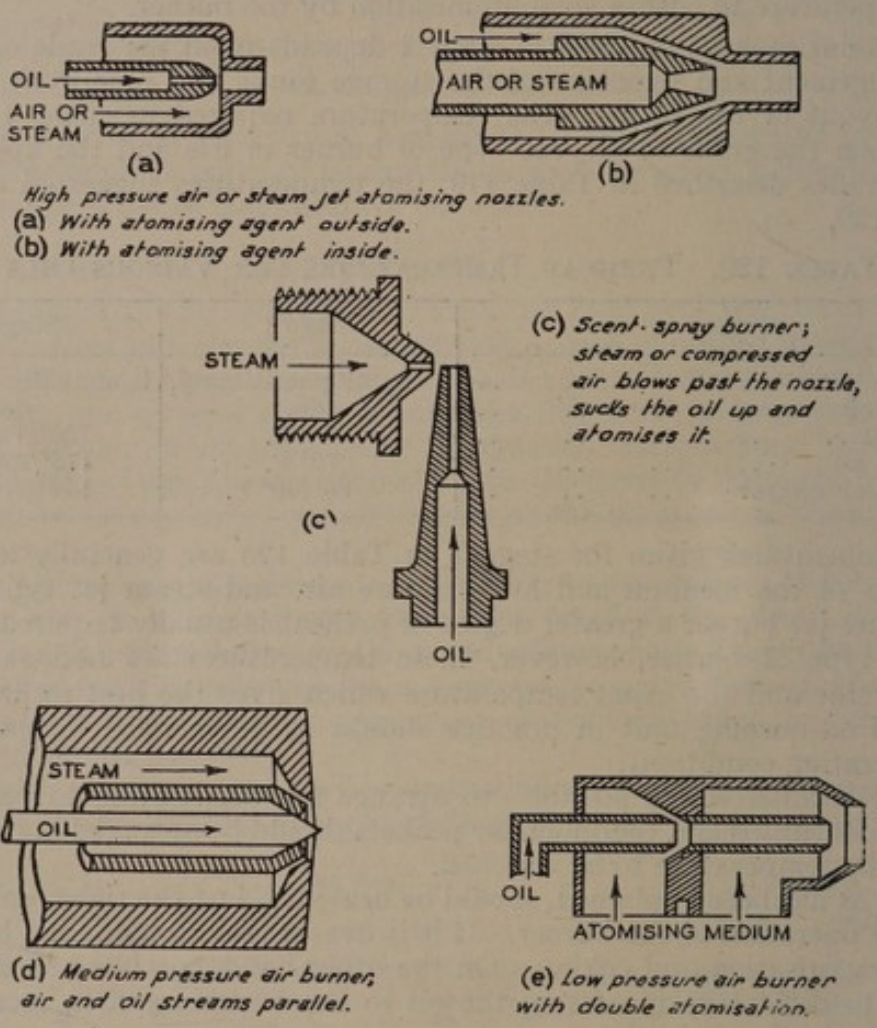


FIG. 271. Diagrams illustrating types of oil burners.

(F. J. Battershill, I. Lubbock and R. H. B. Foster, reproduced by permission of the publishers from "The Science of Petroleum," Vol. IV (Oxford Press).)

OIL BURNERS

For burning fuel oils of the types mentioned in Tables 119 and 120 atomising burners are used. There are several types of these, but for the most part they can be classified into :—

(1) Pressure jet burners which are mainly used for steam boilers. The oil is forced through a mechanical atomiser at a high pressure, and is given a rotational velocity. The oil thus issues from the nozzle in a fine spray in the form of a hollow cone.

(2) Steam jet. This type is also used in steam raising for the most part, and in principle is almost identical with the high pressure air burner.

(3) Air jet burners, which are those most usually applied to industrial furnaces. Atomisation is effected by air under a pressure which may range from 0·25 lb. per square inch to 100 lb. High pressure burners operate at over 5 lb. ; medium pressure at 5 lb. down to 1½ lb. ; while below this the burners are referred to as low pressure.

Fig. 271 shows a number of diagrams illustrative of these types of burners.

The steam jet burner requires between 0·5 and 0·85 lb. of steam per lb. of oil, and when used on oil-fired steam boilers absorbs at least 4–5 per cent. of the total steam production. Steam reduces the flame temperature, though it may assist in the subsequent heat transfer. The wear that always takes place on steam jets may cause the consumption to increase materially if the jet is not inspected and tested for size at frequent intervals.

The low pressure air system is used very largely for oil-burning industrial heating operations, and with this system the combustion air can be heated to 500° F. before passing through the burner. A higher degree of preheat can, of course, be applied to the secondary air.

The air supplied for the purpose of atomisation is, of course, the primary air. Secondary air is always necessary, except when low pressure air burners are arranged to take the whole of the combustion air for atomisation. Fig. 272 shows a burner arranged for injecting (*a*) all and (*b*) part of the combustion air.

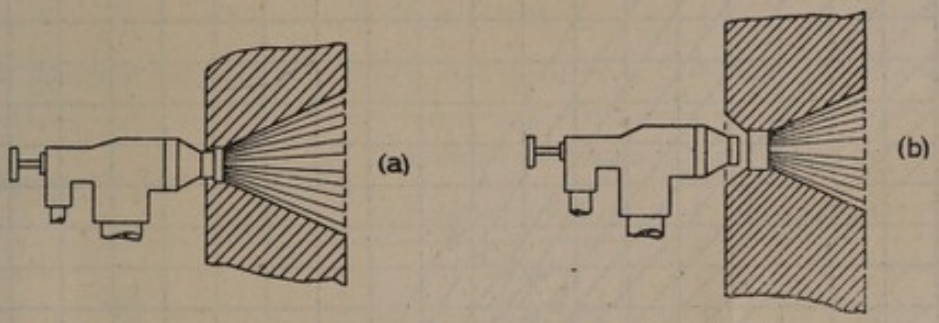


FIG. 272. Burner arranged for injecting (*a*) all; (*b*) part of the combustion air.

(F. J. Battersbill, I. Lubbock and R. H. B. Foster, reproduced by permission of the publishers from "The Science of Petroleum," Vol. IV (Oxford Press).)

In burner (*b*) some at least of the secondary air is drawn in by the action of the burner. Where highly preheated secondary air is used, it is drawn in by the furnace draught in the usual way through separate ports.

OIL BURNING

Success in burning oil fuel depends on three factors which are the primary concern of the designer, namely :—

(1) A suitable burner.
(2) A combustion chamber of the right shape and dimensions.
(3) Correct design of furnace, or whatever other type of plant is used for transferring the heat generated by combustion.

The skill and attention of the operator are required to see that the burner is functioning properly and that air is supplied for combustion in the right amount and in the right places. The burner cannot function properly unless it is provided with oil of the right viscosity.

For the most efficient utilisation of the fuel, it is necessary :—

(1) To approach in the outlet flue gases the maximum possible content of CO_2 subject to there being no smoke or unburnt gases (CO) present.
(2) To secure the lowest practicable outlet flue gas temperature at the chimney base.

Fuel oil, being composed essentially of carbon and hydrogen yields on combustion CO_2 and water vapour accompanied by the nitrogen of the combustion air. The composition of the flue gases depends to some extent upon the character of the oil, but for petroleum oils, the theoretical figure with no excess

air is generally between 15 and 16 per cent. CO_2. Fig. 273 shows combustion data for petroleum oils and indicates the losses of heat that arise with high

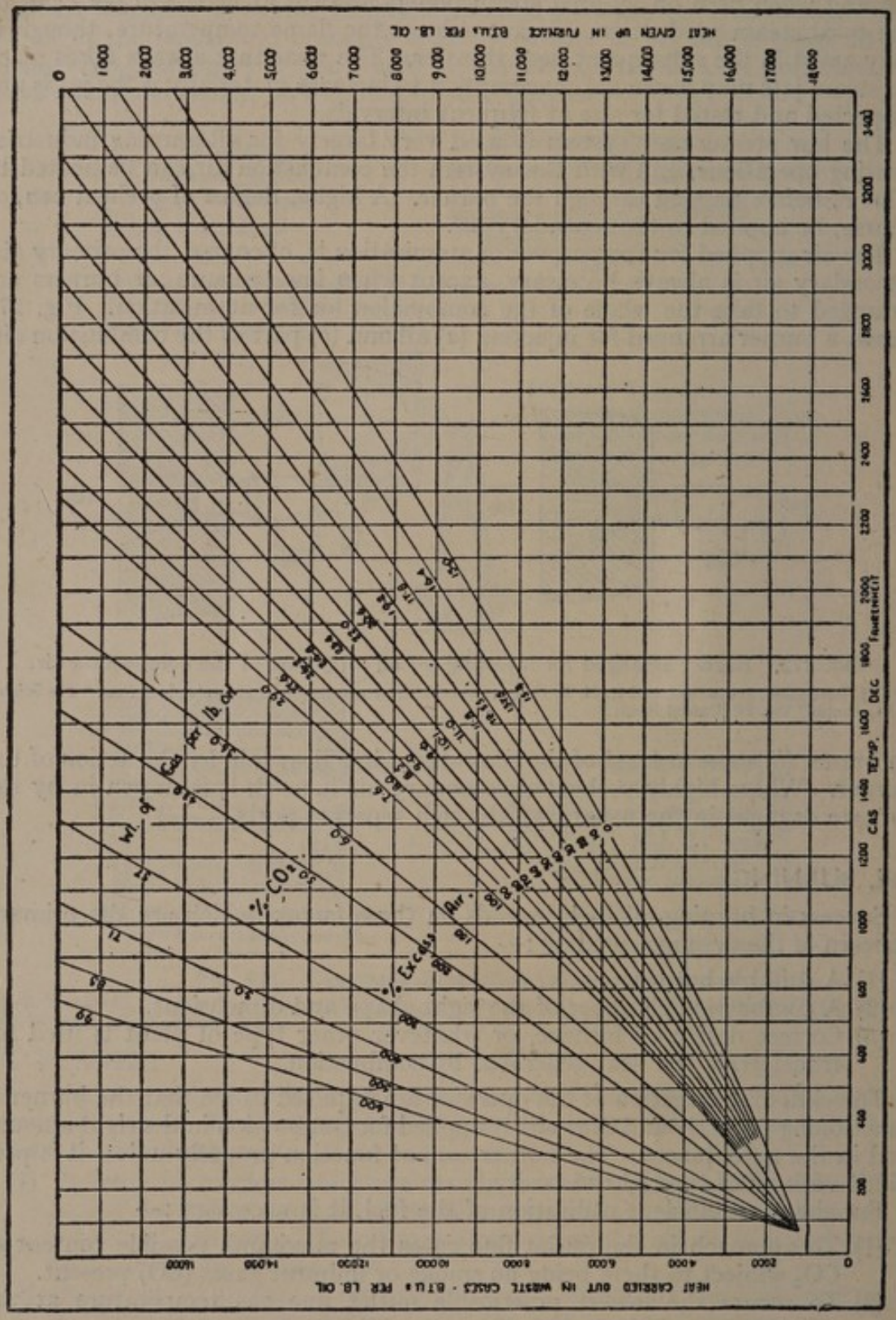


FIG. 273. Combustion data for petroleum oils.

(F. J. Battershill, I. Lubbock and R. H. B. Foster, reproduced by permission of the publishers from "The Science of Petroleum," Vol. IV (Oxford Press).)

flue gas outlet temperatures. Generally some 30 per cent. of excess air must be allowed for effective combustion, so that the flue gas should contain between

11 and 12 per cent. of CO_2, and as previously mentioned, no more than traces of CO.

Coal tar oils, being higher in carbon and lower in hydrogen content than petroleum products, give a theoretical CO_2 content of about 17 or 18 per cent., and the CO_2 content of the flue gases for efficient working is generally between 13 and 14 per cent.

The recommendations here made with regard to CO_2 content presuppose a reasonably steady load and normal operating conditions ; with a fluctuating load and with very low rates of burning, a lower CO_2 may be unavoidable. Furthermore, the CO_2 content attainable in practice may be governed by the process being carried out if that demands a controlled percentage of oxidising or reducing gases in the furnace atmosphere.

The relationship between excess air and CO_2 content of the flue gases is given in Table 121. This table is subject to the proviso just made that the exact figures will depend upon the composition of the fuel.

TABLE 121

RELATIONSHIP BETWEEN EXCESS AIR AND CO_2 CONTENT OF FLUE GAS

Excess air per cent.	Theoretical maximum CO_2 content of dry flue gases.	
	Petroleum fuel oil.	Creosote/pitch mixture.
	Per cent.	Per cent.
Nil	15·4	17·6
10	13·75	15·9
20	12·55	14·6
30	11·8	13·4
40	11·0	12·4
50	10·1	11·5
60	9·6	10·8
70	9·0	10·2
80	8·5	9·6
90	8·0	9·1
100	7.6	8·6
150	6·0	6·9
200	5·0	5·7
300	3·8	4·3
400	3·0	3·4

PREHEATING COMBUSTION AIR

It is not generally recommended that air should be passed through an oil burner at a temperature above 500° F. and, where a higher degree of preheat is to be used, it is advisable to pass a small proportion of the total combustion air through the burner for atomising purposes at a comparatively low temperature, and the remainder as high temperature air through secondary air ports.

The advantages of using preheated air for combustion have been stated in general terms elsewhere. For present purposes these advantages may be thus summarised :—

(1) The flame temperature is raised, resulting in increased heat transfer.
(2) The rate of combustion is increased, thus further raising the flame temperature.
(3) The amount of excess air necessary for combustion can be reduced.
(4) The heat abstracted from the flue gases by the combustion air is returned to the furnace.

The approximate savings in oil fuel which can be made by preheating the air are indicated in Table 122 (N. L. Hudson, R. J. Bressey and T. C. Bailey, *J. Inst. Fuel*, XIII, 90).

TABLE 122. APPROXIMATE SAVINGS IN OIL CONSUMPTION USING PREHEATED AIR. CALCULATED ON A BASIS OF 20 PER CENT. EXCESS AIR FOR COMBUSTION AND A FLUE GAS TEMPERATURE OF 800° C. (1,472° F.)

Passing all combustion air through burner at a temperature of	200° F.	300° F.	400° F.	500° F.
Approximate saving in fuel .. .. per cent.	4·8	7·9	10·9	13·7
Passing 25 per cent. of combustion air at 200° F. through burner and remainder as secondary air at a temperature of	700° F.	800° F.	900° F.	
Approximate saving in fuel per cent.	15·9	17·8	19·7	
Passing 5 per cent. of combustion air cold through burner and remainder as secondary air at a temperature of	700° F.	800° F.	900° F.	
Approximate saving in fuel .. .. per cent.	18·2	20·5	22·7	

MAINTENANCE AND OPERATION OF THE BURNER

It is manifestly important to keep the oil burner in good order. The maintenance depends to some extent upon the type of burner used, but most of the details given here apply equally to steam jet and air jet burners.

It is important to prevent the burner from becoming overheated. When in use the passage of air or steam and oil through the nozzle cools the metal to some extent. When the oil is shut off, however, the radiant heat may cause damage and may coke the fuel left in the burner. One widely practised method of avoiding this with steam jet burners is to keep a little steam passing through during shut-down periods while the furnace is hot. This practice is not recommended as it is wasteful of steam.

The correct method is to remove the burner as soon as the oil supply is turned off. If this should not be possible the burner should be protected by a piece of sheet metal interposed between the burner nose and the furnace.

The size of the burner nozzle, the oil pressure and temperature, and the pressure of the steam or air (if used), should all be adjusted and properly balanced. The burner manufacturers calibrate nozzle capacity against pressure with fuel oil at a given viscosity. From these calibration curves the correct size of nozzle can be determined.

The temperature to which the oil should be heated to attain the required viscosity can be obtained from the temperature-viscosity characteristics. As shown in Fig. 269, different oils have different curves, and so also have oils of the same class derived from different sources.

Oil filters are usually provided, e.g. at the inlet to the fuel pumps and at the outlet of the fuel oil heaters. The water that may collect here must be removed periodically by drain cocks. The periodic cleaning of the filters is essential as the burner operations can be upset by solid particles lodging in the small passages of the nozzles, and equally, of course, by the clogging of the filters.

It is usual to change burner nozzles every shift, so that any carbon deposit on the nozzles may be removed. The used burner is soaked in kerosine or gas oil to facilitate cleaning when the nozzle is dismantled. Creosote is used for cleaning burners handling tar oils. No tools which are likely to enlarge the orifice or the small oil passages should be used when cleaning the parts which comprise the nozzle.

When lighting a burner, care must be observed that a flame from a torch, or other source, is established in front of the nozzle before turning on the oil supply to the burner. The application of a flame to a combustion chamber charged with oil mist will cause a serious back-fire.

FURNACE AND BOILER OPERATION

The technique of boiler operation has been discussed in earlier chapters and furnace operation is dealt with in Chapter XVIII. Whatever is there said in regard to boilers and furnaces in general applies with equal force to oil-fired plant. The need for insulation, the avoidance of leaky brickwork, the need for keeping doors and openings closed as far as operation permits, and for loading furnaces to the maximum extent, all come under this heading.

There are, however, certain special characteristics of oil firing, to which reference must be made.

Each liquid fuel burner should incorporate where possible some form of simple graduated indicator attached to the valve which shows visually the degree to which the valve is open. The rate of oil supply to some furnaces is thermostatically controlled according to the furnace temperature. Alternatively, with hand regulation graduated valves enable the operator to regulate the oil flow to the burner to a predetermined amount. The dampers should be similarly graduated so that the oil and air supply are correctly adjusted during periods when the load is varying.

The combustion should be controlled by flue gas analysis ; the best method is a CO_2 recorder checked periodically by chemical analysis. When an instrument is not available for the purpose, the air supply is adjusted to the minimum which will ensure reasonably smoke-free combustion. A very faint appearance of light smoke at the chimney is an indication of correct firing conditions, provided the proper degree of atomisation is being obtained at the burner.

A frequent sign of inefficient conditions is the formation of carbon in combustion chambers. This is generally an indication that either (*a*) the burner needs adjustment, or (*b*) that the furnace brickwork needs re-designing to avoid impingement of the oil spray on the burner quarl or furnace walls. Carbon formation is an indication of wasted oil, it leads to shorter life of the refractories, and upsets operating conditions in the furnace.

CRITERIA FOR FUEL OILS

Several characteristics, taken in conjunction with one another, serve to indicate the nature and properties of any particular fuel oil.

Viscosity. The most important of these from the practical point of view is viscosity. The theoretical aspects of viscosity have been discussed in Chapter IX, and its importance in relation to oil-firing earlier in this chapter. The viscosity of oils is measured by the time taken for a certain volume to pass through a jet of defined dimensions under such conditions that the head of oil, i.e. the pressure at the jet, is similar for all determinations. This is essentially a convenient method of comparing viscosities, but by similarly treating in the same apparatus a liquid of known viscosity, the measurements can readily be converted into absolute units. The effect of temperature on the viscosity of some fuel oils was shown in Fig. 269.

The apparatus used for the commercial determination of viscosity in this country is the Redwood Viscometer ; in this apparatus is measured the number of seconds required for 50 millilitres of the oil to issue through the orifice at a given temperature. In America the Saybolt apparatus is generally used.

Data regarding the absolute viscosity of fuel oils, which would be required for calculating pipe sizes (Chapter IX), are given in Table 123.

TABLE 123

KINEMATIC VISCOSITY OF POOL FUEL OILS IN C.G.S. UNITS (STOKES)

Temperature ° F.	Gas oil	Fuel oil	Heavy fuel oil
32	0·110	14·82	74·5
70	0·056	1·41	8·65
100	0·037	0·52	2·35
140	0·024	0·205	0·73
200	0·015	0·075	0·21

The kinematic viscosity of water at 70° F. is 0·01 stokes, and at 200° F. 0·003 stokes.

Conversion tables (e.g. " Technical Data on Fuel," 4th edit., pp.269 and 270) enable the readings of these apparatus to be mutually converted and to be converted into absolute units. In practice, however, the viscosity of an oil in this country is stated as being so many seconds Redwood at some prescribed temperature.

Specific Gravity. The specific gravity of an oil is defined as the ratio of the mass of a given volume of oil to that of an equal volume of water at 60° F.

Broadly speaking, the higher the specific gravity, the greater is the heating value in B.Th.U. per gallon, because the higher the specific gravity the more weight of combustible will there be in the gallon. For most practical purposes, hydrometers are used for determining specific gravity, and in effect the value of this function is in relation to the calculation of weights and volumes.

Flash Point. The flash point is the temperature at which the air space above the oil under standard conditions contains sufficient oil vapour to be ignited by a flame. In this country the usual limit of flash point is 150° F. minimum.

Pour Point. The pour point is the lowest temperature at which the oil will flow when chilled under prescribed conditions. If the oil contains wax, anomalous results may be found. For this reason previous heat treatment is specified, though it must be admitted that actual flow through pipes in practice is predictable from the pour point only in conjunction with the viscosity of the oil at the temperature considered.

Gross Calorific Value. The calorific value of fuels was discussed in Chapter V, and further information regarding oils has been given in Table 120.

The net C.V., which assumes that the water is not condensed, is for petroleum oils about 1,100 to 1,200 B.Th.U. below the gross value.

Carbon Residue. The amount of carbon residue remaining on heating a fuel oil under certain specified conditions gives some indication of the relative carbon-forming propensities of oils in practice. It should be emphasised, however, that the formation of carbon deposits when oil is burnt is generally due to faults in plant or in operating conditions.

Ash. Ash is determined by the complete combustion of a weighed amount of oil, the inorganic material forming the ash remaining unburnt. In general the ash content of petroleum fuel oils is so small as to be of little practical importance, but there are a few applications in which the amount and/or nature of the ash may be of some moment.

General. Details of the tests performed on oils will be found in " Standard Methods for Testing Petroleum and Its Products," published by the Institute of Petroleum.

Attention is also drawn to the following British Standard Specifications :—

No. 209—1937. Fuel Oils for Diesel Engines.
No. 742—1937. Fuel Oils for Burners.

CHAPTER XXIX

GRAPHICAL HEAT BALANCES

General principles—The construction of graphical heat balances—Heat flow diagrams—The heat requirements of different systems as shown by heat flow diagrams.

GENERAL PRINCIPLES

THE performance of a heating unit, as has been pointed out in earlier chapters, may be expressed in terms of fuel used per unit of output. Thus the performance of a billet heating furnace can be assessed in pounds of coal per pound of billets heated ; of a boiler in pounds of coal per pound of steam raised ; or of a coke oven in cubic feet of gas burnt per ton of coal carbonised. This method is satisfactory so long as it is desired only to compare the performance of furnaces using the same fuel and heating similar material to the same temperature, but it gives no absolute method of comparison. The relative performance of two cement kilns, for example, is not in any way assessed by the statement that one uses 30 lb. of coal A to burn 100 lb. of clinker from a slurry containing 43 per cent. moisture and the other uses 20 lb. of coal B to burn 100 lb. of clinker from a slurry containing 33 per cent. moisture.

An absolute standard of comparison is necessary. This can be obtained by ascertaining as a basis the quantity of material that could be treated by unit quantity of the fuel in a perfect appliance. This computation is based on the law of conservation of energy which is a summary of numerous experiments well known to physicists whereby it is proved that heat and energy are mutually convertible, nothing being lost in the process. These experiments show that when a combustible of constant composition, e.g. carbon, is burnt a fixed and definite amount of heat is generated in the process (Chapter V), that when 1 kWh of electricity is absorbed in a resistance, 3,413 B.Th.U. of heat is liberated, that a mass of 1 lb. moving with a velocity of 1 foot per second releases $1/(778 \times 64{\cdot}4)$ B.Th.U. when brought to rest, that 1 B.Th.U. corresponds to 778 ft.-lb. of work, and so on.

When more than one process is carried on simultaneously, if all the energies in the reacting materials before the process are expressed in one set of units—e.g. B.Th.U., or ft.-lb.—they precisely equal the sum of the energies after the processes, expressed in the same units. For example, if electricity is used to lift weights, the total energy of the electricity used is exactly equal to the sum of the energy absorbed by lifting the weights and of that appearing in the form of heat in the bearings of the motor, pulleys, etc.

It follows from this law, that any given process requires a certain and definable minimum of energy. To heat 100 lb. of steel billets from 25° C. (77° F.) to 1,100° C. (2,012° F.) requires, to a close approximation, 31,500 B.Th.U. As explained in Chapter V this figure is given by

$$\text{weight} \times \text{temperature rise} \times \text{specific heat.}$$

No furnace which is required to heat billets through this range of temperature can possibly use less than this amount of heat. In terms of weight of fuels used, this means that the fuel consumption cannot be less than 2·5 lb. of standard coal having a C.V. of 12,500 B.Th.U. per lb. Similarly, the production from water at 16° C. (60° F.) of 100 lb. of steam saturated at 121° C. (250° F., 30 lb. per square inch absolute) requires 113,700 B.Th.U., and hence not less than 9 lb. of standard coal can possibly be used to carry out this task.

This method of comparison of appliances from the purely thermal angle is of much greater value than the simple statement of pounds of fuel per 100 lb.

of product. Appliances are compared by the ratio of fuel ideally necessary to fuel used in practice, reduced to fuel of standard C.V.

Useful as this figure is, it is not sufficient, however, for it is only a measure of overall performance, whereas detailed analysis is necessary to determine where the heat is lost, or why one appliance is better than another. It is therefore necessary to construct a heat balance, again using the principle of the conservation of energy. Heat balances are nothing more than an attempt to trace what happens to all the energy at any given stage; in other words, an attempt to balance all the energy entering the stage against that leaving.

The forms of energy considered in a heat balance can be classified as follows:

(1) Chemical energy of fuel and of incompletely burnt gases, and electrical energy of mechanical work.
(2) Sensible heat of hot gases and steam.
(3) Sensible heat and heat absorbed by endothermic reactions in the materials. ("Endothermic" reactions are chemical reactions resulting in the absorption of heat, in distinction to "exothermic" reactions which evolve heat.)
(4) Heat flowing to the surroundings, including cooling water.

A heat balance may be drawn up on the basis of measurements for a furnace that is working, or may be deduced from appropriate assumptions and technical data for a furnace that it is proposed to build. A complete heat balance can be drawn up from a knowledge of the quantities of heat in the first two of these forms supplied to a stage of the process, and in all four forms after the stage. The performance of the appliance can then be assessed or analysed.

THE CONSTRUCTION OF GRAPHICAL HEAT BALANCES

The first step is to determine how detailed the heat balance should be. An overall balance, comprising nothing more than the calculation of the total energy entering and leaving the system is relatively simple, but gives no information of the relative effectiveness of the different parts of the system, the fuel bed, the heating chamber, the recuperator, and so forth. At the other extreme it is possible to subdivide the furnace into a large number of zones and make a complete heat balance for each. This will require many more measurements, but when it is possible to make the measurements will give much more detailed information. Practical conditions generally prevent this being done. Therefore, it is usually best to make a compromise and divide the system into two or three main units, and make a heat balance for each of these units. Thus, a boiler may be divided into main combustion chamber, superheater, water tubes, economiser and air heater, if the gas and steam or water measurements are made at the entry and exit of these parts. Similarly, an open hearth furnace can be divided into producer, mains, regenerators, and working chamber.

The second step is to obtain the necessary measurements. To assess the chemical energy of the fuel its calorific value must be determined. Similarly, the chemical energy of incompletely burnt gases, e.g. producer gas, can be obtained from a complete gas analysis. The term "incompletely burnt gases" does not refer to the gases leaving the combustion zone, but to the gas used for heating, the assumption being that coal or coke is the primary fuel fed into the producer, and that by the incomplete combustion of this fuel, producer gas (or water gas or blast furnace gas), is formed which is burnt completely in the combustion chamber. The C.V. of any tar and hydrocarbons present in the heating gas must be added to the C.V. of the gas, as these bodies have a high heating value (cf. "Producer Gas," Chapter II). If tar is condensed en route to the furnace, as in a main carrying crude producer gas, an estimate must be

made of the extent of condensation and this heat debited to the mains. Electrical energy and mechanical work can be obtained by measurement of wattage or horse-power.

The sensible heat of the hot gases and steam is calculated as shown in Chapter V. The quantity of these gases may be obtained from calculation as there described. If, however, hot, crude producer gas is used, it may be possible only to measure the weight of fuel gasified in the producer. The quantity of producer gas can then be deduced from the amount of carbon consumed in the producer (i.e. carbon charged less carbon in ashes and dust and tar deposited in the mains) and the amount of carbon in 1 cubic foot of the gas as fed to the furnace.

The heat capacity and heat absorbed by endothermic reactions in the material being heated is obtained from a measurement of its temperature at the appropriate points and from laboratory measurements of the heat required per unit weight for heating or cooling between the various temperatures at which each stage commences and finishes. These total heat differences are, of course, multiplied by the weight of material processed. As an example, in a furnace treating 100 lb. of material as the unit, it may be found that the heat required to raise 1 lb. of stock from 60° F. to 500° F. is 200 B.Th.U., and to raise it from 60° to 900° F. is 600 B.Th.U. (the temperature rise of 500° to 900° being one stage). The total sensible heat plus endothermic heat is thus 400 B.Th.U. per lb. for that stage, or 40,000 B.Th.U. for the process.

Finally, the heat losses to the surroundings must be assessed. It is usual to calculate these by difference ; that is to say, the other three heat quantities are measured, and by the law of the conservation of energy, the difference between the sum of these quantities and the total heat supplied to the system must be the heat lost to the surroundings. Although it is difficult, it is more satisfactory, where possible, to make a direct estimate of this quantity. The heat absorbed by water cooling can be readily calculated from measurements of the quantity of water passing (Chapter IX) and its temperature rise, providing precautions are taken to obtain the true mean temperature of the water. This measurement is always well worth making, even where the other heat losses cannot be measured, in order to see whether the system is excessively cooled. Heat losses by conduction through the furnace structure, and by storage in the structure followed by loss during periods when the furnace is not in operation can best be assessed by measuring the surface temperature of the structure at a number of points with a contact thermocouple, and multiplying the mean of the heat losses corresponding to these temperatures obtained from tables, by the surface area (cf. Chapter VIII). Alternatively, it can be calculated from the inside temperature and the thermal conductivity of the wall.

Let it be supposed that a complete heat balance for any one stage of a process can be made. The heating gases arrive at this stage with a known content of chemical and sensible energies, and leave with other known values. A known amount of heat is given to the material being heated, and a known amount to the furnace structure. The results can then be expressed graphically by means of a heat flow diagram, sometimes termed a " Sankey " diagram. An outline diagram is made of the furnace system, showing the various parts and their interconnection. The flow of energy is then represented on this diagram by a stream flowing through the system. The width of the stream at any point is drawn proportional to the amount of energy flowing at that point.

It is suggested that the best units in which to express the energy are pounds of standard coal, or cubic feet of standard gas, or gallons of standard oil per 100 lb. of product. This method has the advantage over the use of percentages that if two diagrams are made for different furnaces producing the same material, the relative fuel consumptions are shown directly. It is, of course,

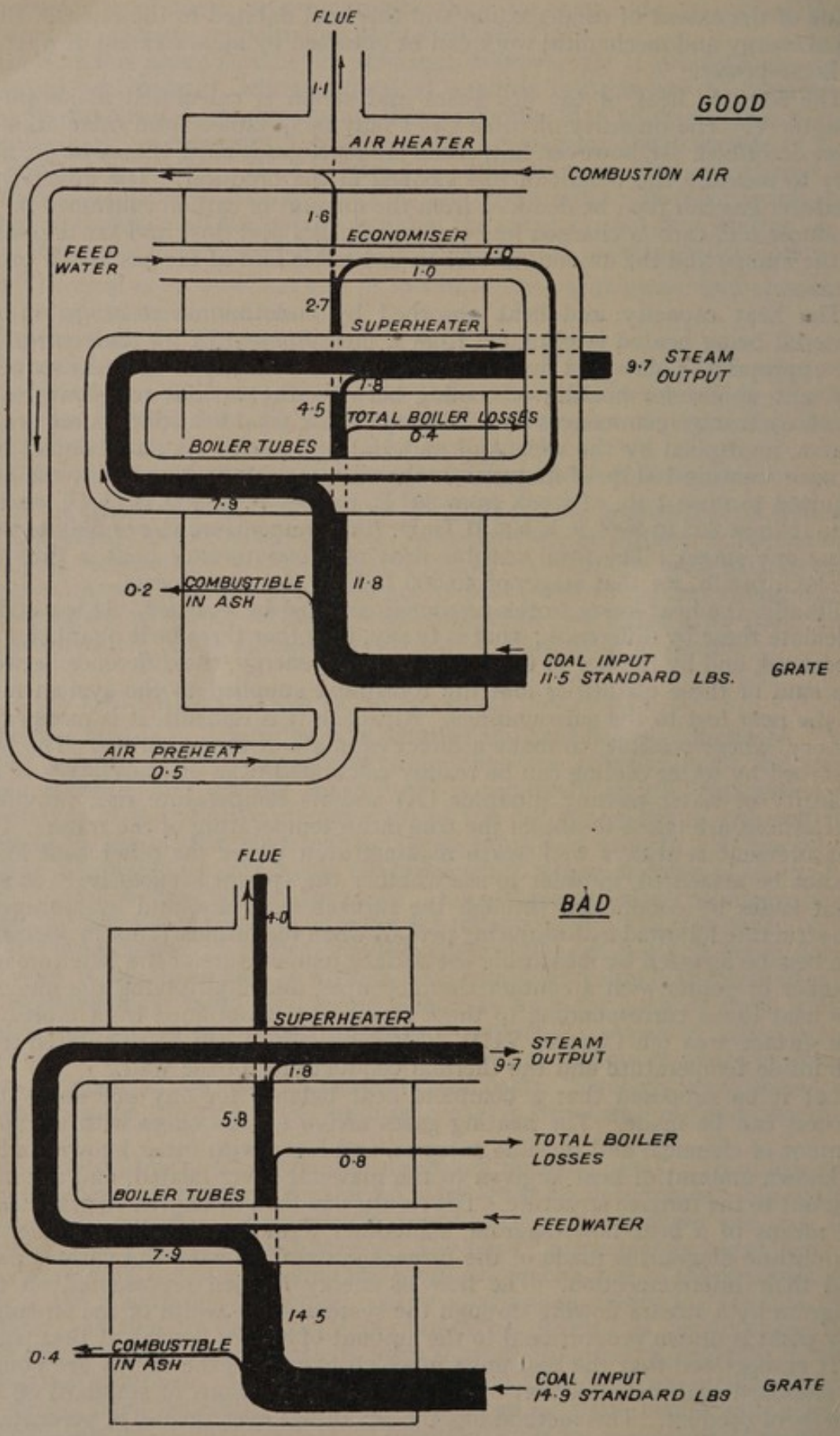


FIG. 274. Heat flow diagram for steam boilers.

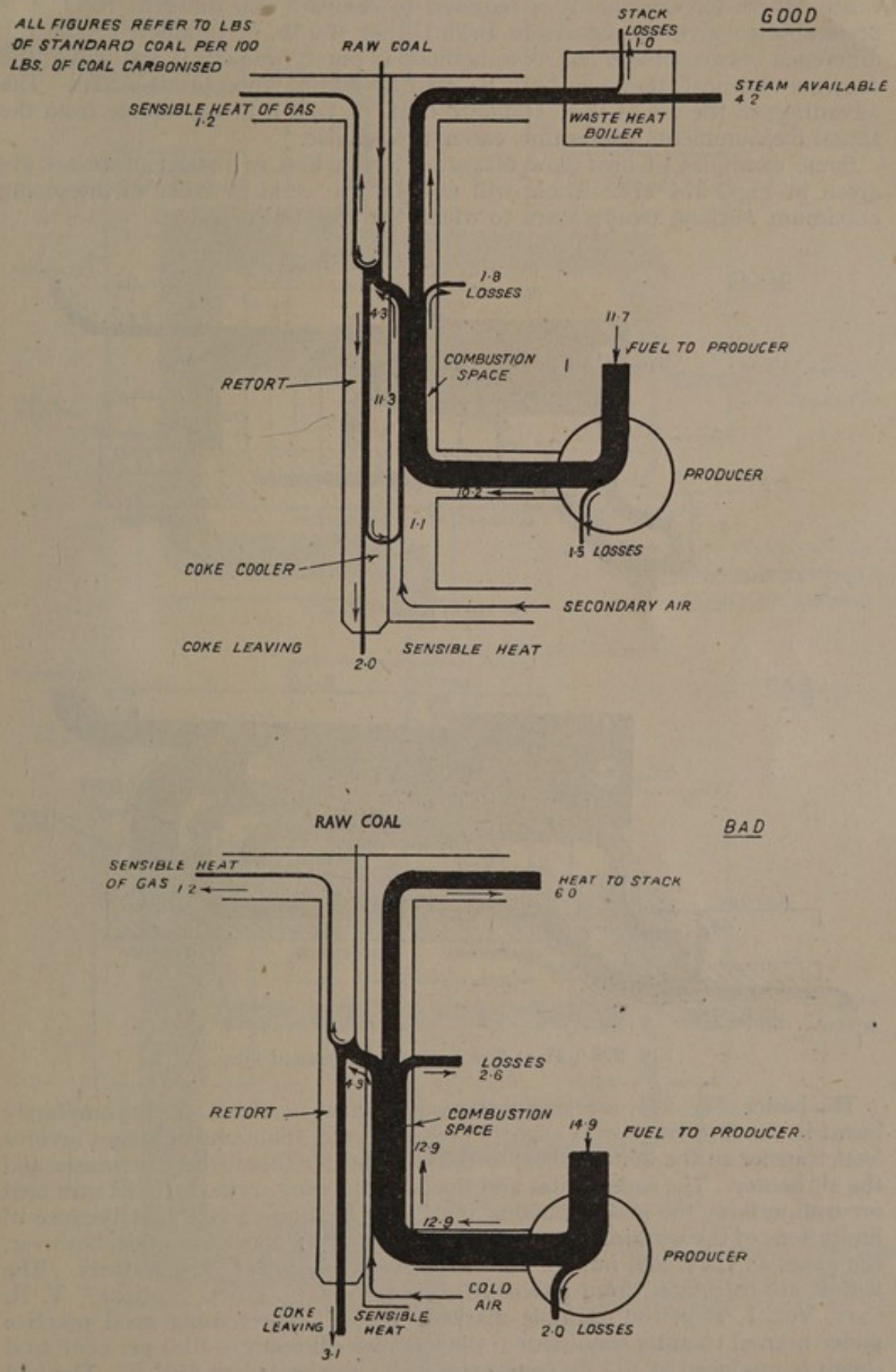


Fig. 275. Heat flow diagram for continuous vertical gas retorts.

exactly equivalent to the use of heat units per 100 lb. of product, and sometimes, where for example it is required to compare two different fuels, it is preferable to give the figures in B.Th.U. per 100 lb. of product. Since the difference between these last two methods is purely numerical and leaves the diagram identical, they can both be used by adding appropriate scales. The advantage of the use of the " standard coal " is that it is less remote from the actual measurements and is thus easier to visualise.

Some examples of heat flow diagrams for various industrial processes are given in Figs. 274–278. These will now be discussed in order of increasing maximum working temperature to which the material is heated.

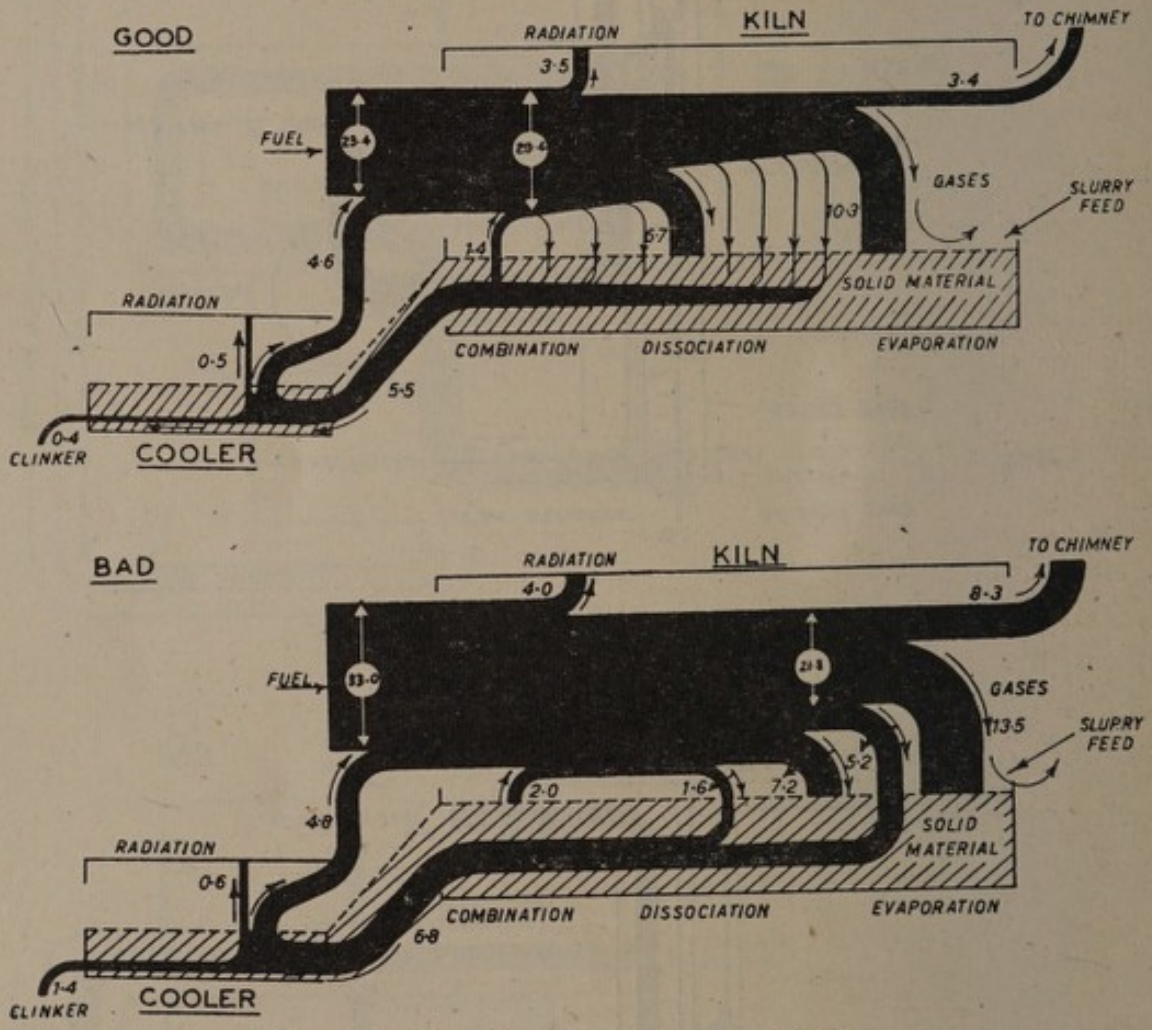


FIG. 276. Heat flow diagram for cement kilns.

The boiler, Fig. 274, is a straightforward system in which coal is completely burnt in the first stage with slightly preheated air. Subsequent stages involve heat transfer to the boiler tubes, to the superheater tubes, the economiser and the air heater. The superheater and the boiler heating surface are not arranged for counterflow, the gases reaching the boiler heating surface first because of limitations of the working temperature of the metal ; apart from this, however, the gases as they cool impart heat at successively lower temperatures. The figures are calculated from published data (" Electric Power Stations," T. H. Carr, Vol. I, 258), the example marked " good " representing good practice under normal running conditions ; the thermal efficiency is 84·5 per cent. and steam is produced at 625 lb. per square inch superheated to 850° F. The bad boiler has an efficiency of 65 per cent., the main difference being in the greater

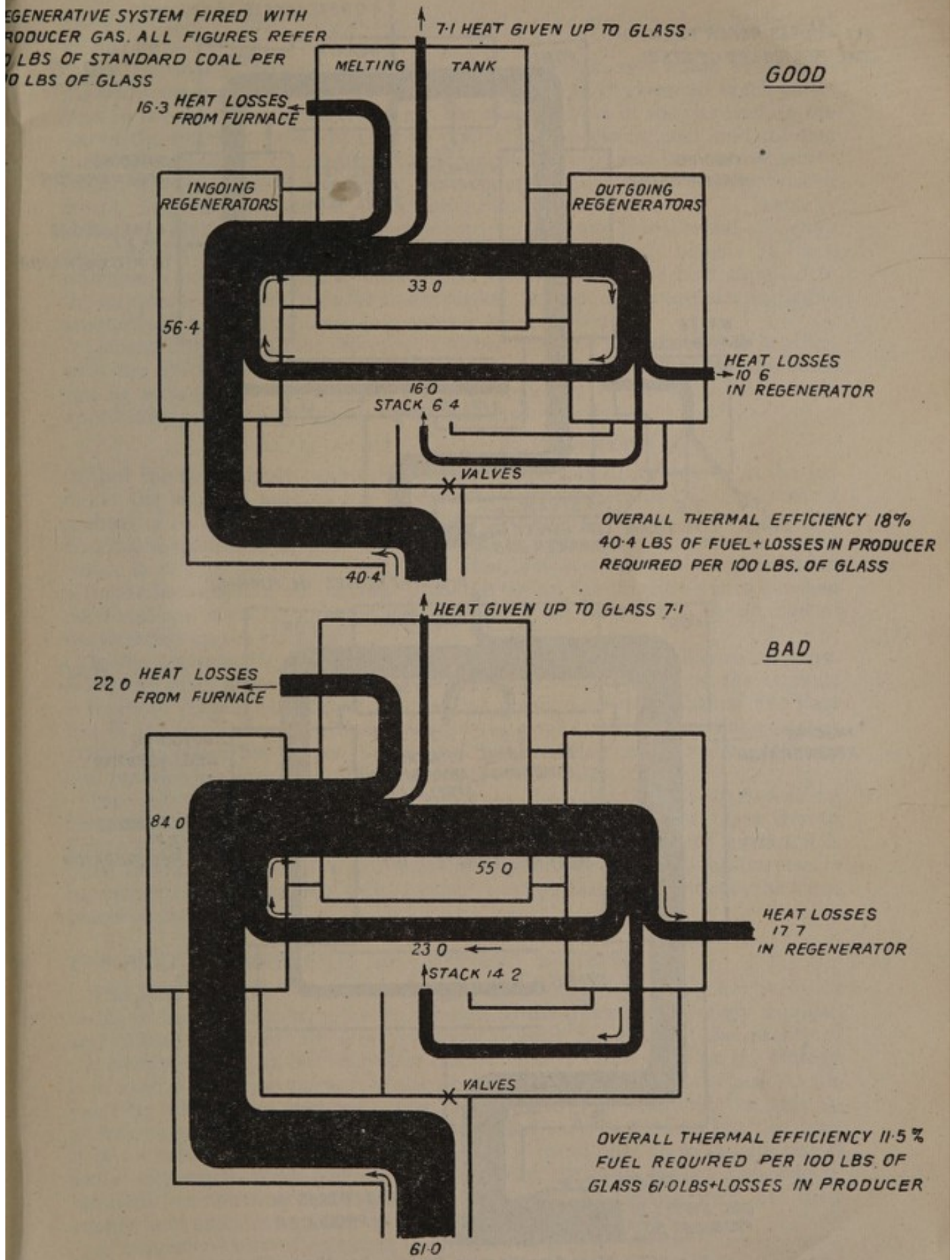


FIG. 277. Heat flow diagram for glass tank.

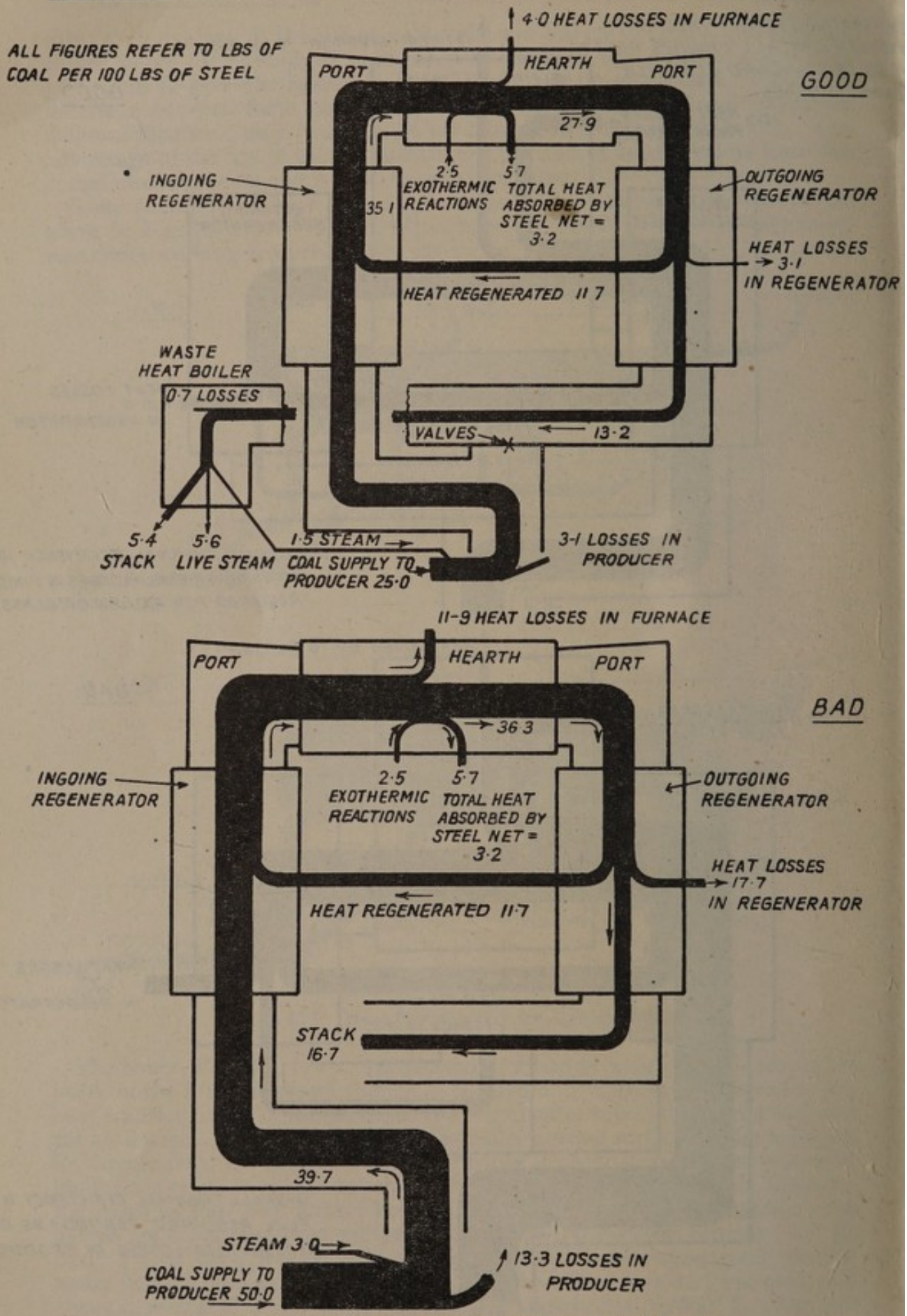


FIG. 278. Heat flow diagram for open hearth steel furnace.

flue losses due to lower CO_2 in the gases and to the absence of air heater and economiser.

The continuous vertical retort, Fig. 275, represents one phase of modern gasworks practice in which the coke leaving the retort gives up much of its heat to the secondary combustion air, the sensible heat in the gas leaving the top of the retort is partly utilised to preheat the incoming coal, and the flue gases are passed in counterflow to the incoming coal and are subsequently passed through a waste heat boiler. The figures are calculated from published data (" Fuel Requirements of Carbonising Plants," S. Pexton, *Gas J.*, January-February, 1943). The chief difference between high and low thermal efficiency arises from the absence of a coke cooler and a waste heat boiler. If these principles could be carried to their logical conclusion, all the heat supplied to the retort would be available for steam raising, as most coals require a negligible amount of heat for endothermic reactions in carbonising.

Cement kilns, Fig. 276, represent another example of counterflow heating, with the difference that heat transfer is here much simpler because the gases can be mixed with the material being heated. In this process there is an appreciable heat absorption by the endothermic reaction :

$$CaCO_3 = CaO + CO_2 - 42{,}700 \text{ calories},$$

so that the heat supply cannot be reduced below this figure, but as in the gas retort the sensible heat of the fired clinker can be returned in the form of preheat of combustion air. A cement kiln working with a wet feed (slurry) is also called upon to evaporate a large quantity of moisture, the sensible heat of which is not at present recoverable. This, however, does not represent an ultimate limitation of the process. Like the boiler, the chief difference between good and bad practice in the cement kiln lies in the variation in the amount of heat carried up the chimney.

In the regenerative glass tank (Fig. 277) (data from W. A. Moorshead, priv. com.) the gases leave the furnace at a very high temperature and the attempt is made to recycle as much of this heat as possible by regenerators. The heat lost in the furnace and regenerators is of the same order of magnitude as that utilised in these two stages ; consequently the difference between good and bad practice lies mainly in differences in the heat losses.

Precisely similar considerations apply to the open hearth steel melting furnace shown in Fig. 278 (data from Sarjant and Barnes, Trans. Open Hearth Conf., 1938, p. 238). In both the glass tank and the open hearth furnace it is often desirable to install waste heat boilers because owing to limitations in regenerator size and rates of heat transfer the regenerators do not cool the gases below some 700°–800° C.

THE HEAT REQUIREMENTS OF DIFFERENT SYSTEMS

The heat flow diagram shows whether a system is operating effectively, in terms of heating efficiency, i.e. in relation to the quantity of energy supplied and utilised. It does not, however, show directly the effect of the quality of the energy concerned, but this effect can be derived indirectly from it. Energy is in itself of no value unless it is also of sufficiently high quality to perform the work for which it is required. The Sahara Desert, for example, contains an enormous quantity of heat measured in B.Th.U. above 32° F. ; but this heat is of no value for the generation of power ; it could only be made effective if some very much colder body of large heat capacity were placed alongside it. Similarly, combustion gases at 800° C. are useless for melting steel, but their energy is of sufficient quality to be valuable for raising steam.

A convenient way of assessing the quality of a given source of energy for a given purpose is to calculate the fraction of the energy which is ideally available.

If combustion gases (assumed to be of constant specific heat) at a temperature T_1 are required to heat material to a temperature T_2, one fraction of the energy, A, of magnitude $\frac{T_1 - T_2}{T_1}$ is available for the purpose and the remaining fraction, B, of magnitude, $\frac{T_2}{T_1}$ is not.

The heat flow diagram can be used to investigate for various processes the conditions governing the best use of these two fractions. The conditions governing the best use of fraction A are as follows :—

(1) The fraction A should be made as large as possible by obtaining complete combustion with a minimum of excess air.

(2) The heat transfer in the primary region should be as complete as possible so that the gases are cooled as nearly as is possible to T_2 in this region. How well these conditions are fufilled is shown in the diagram by comparing the ratio of the heat leaving the primary region to the heat retained in it, with the ideal ratio : $\frac{T_t - T_2}{T_t}$ where T_t is the theoretical temperature of combustion.

(3) The heat retained in the furnace must be put to useful purpose by avoiding heat leakages ; i.e. by the use of insulation (where possible), reduction of radiation through openings and of other unnecessary heat losses as described in Chapter XVIII. The success with which this is done is shown in the diagram by the ratio of the heat lost to the heat usefully applied in the furnace.

Fraction B cannot be used for heating at the maximum temperature of the process, but the diagrams show three satisfactory applications for it. In the steel and glass furnaces it is used to preheat the ingoing air and fuel gas, and in the boiler and gas retort it may be used to preheat the air, thus raising the theoretical temperature of combustion, and increasing the value of fraction A. Here, just as in the furnace itself, the conditions whereby the best use may be made of the regenerator, recuperator or air heater are adequate heat absorbing surface, good insulation and minimum air infiltration.

The second method of using fraction B is the counterflow system typified by the vertical gas retort, cement kiln, and to some extent by the boiler. Here the material to be heated requires heat below the maximum temperature and this amount is of the same order of magnitude as the amount required at the maximum temperature. The low grade heat can therefore be partially used for the first stages of the heating of the material.

The last method of using fraction B is for lower grade purposes relatively unconnected with the main process. This is exemplified by the waste heat boilers on the steel melting furnace or gas retort.

Furnaces using any of these three methods for the utilisation of fraction B are thermodynamically sound, since they use heat of successively lower grade for work of correspondingly lower grade ; they are an industrial approximation to the ideal reversible process.

In this way a study of the heat flow diagrams can be made to reveal indirectly the success with which the quality of energy is being utilised just as it also reveals similar information regarding the utilisation of the quantity of energy. It will repay the time spent to construct such a diagram for any appliance in which energy is of major importance, to indicate promising lines for improvement as well as to show outstanding deficiencies.

CHAPTER XXX

INSTRUMENTS

Measurement of composition of flue gases and the instruments used—Measurement of draught—Temperature measurements ; thermometers, thermocouples and pyrometers—The accurate measurement of flue gas temperature : fine wire thermocouples, screened thermocouples, suction pyrometers—Measurement of surface temperature of external walls—Heat flow gauges—Measurement of smoke—Boiler feed water measurement—Maintenance of instruments.

IN chapters dealing with the conditions of combustion and steam raising, reference has been made to the usefulness of certain measurements, such as those for ascertaining temperature, chimney draught, composition of flue gases, etc., as providing data on which it is possible to secure economic control. For all these measurements, instruments of the spot reading, indicating or recording types are required, and this chapter deals with the principles on which the various instruments work and some of the precautions which are necessary in their use.

Such instruments as boiler pressure gauges, water level gauges, etc., which are required by law, and are subject to specifications as to construction and accuracy, are not included in this account.

Full descriptions and diagrams are not given of the construction of proprietary makes of instruments. These will be found in their makers' instructions, which should be carefully followed. The makers also usually give information on the theory of the working of their instruments in their pamphlets, where this is not generally understood, and on servicing.

It is not possible to give any guidance as to whether portable or permanently installed instruments should be used nor whether they should be of the indicating, recording or integrating types. This will depend on the individual circumstances and size of the plant concerned, but in general the larger the plant the more economical it becomes to install the more elaborate and permanent equipment.

Furnace and boiler control starts, of course, with the control of quality and quantity of the coal or other fuel supplied. This subject is discussed elsewhere in this book, especially in Chapter X.

MEASUREMENT OF COMPOSITION OF FLUE GASES

Flue gases usually consist of nitrogen, oxygen, carbon dioxide and water vapour, with some sulphur oxides. When combustion is incomplete, carbon monoxide, hydrogen and smoke will also be present. A high CO_2 content and the absence of CO and H_2 indicates good combustion with little excess air.

The instruments used can be divided into two general classes : (1) those in which the measurement is made by the chemical absorption of the gases, and (2) those of an inferential type in which some physical property of the gas which changes with the concentration of CO_2 is measured.

General Precautions. For both classes of instrument certain precautions are common. These are as follows :—

(1) The point from which the gases are drawn would normally be at the outlet from the boiler or at the base of the chimney. At these points under natural draught the pressure is less than atmospheric, and unless all the brickwork, doors and damper slots between the furnace and the sampling point are in good condition, there is danger of atmospheric air being drawn into the flue and the flue gases so being diluted. All leakage between these points must therefore be made good before analysis can serve any useful purpose for the control of combustion.

(2) Owing to the reduced pressure at the sampling point, the flue gases must be sucked into the measuring instrument by some form of aspirator, which is usually incorporated with the instrument. As all flue gases at times contain smoke and dust, a filter to extract these impurities should be fitted. This usually takes the form of a porous fireclay or carborundum pot fitted to the entry end of the sampling tube. To function well this should be of sufficient size, even when partly coated on the outside with soot or dust, to allow ample gas to pass. It is desirable also to insert another filter of, say, asbestos, glass or cotton wool, at a point on the tube outside the flue which is at a lower temperature, to remove condensed tarry matter. Water will also condense in the pipe-line in the cooler regions, and the whole sampling tube and its connections down to a point at which it reaches the temperature of the boiler room should have a downward slope and a connection to a water seal pot at its lowest point. This provides for the drainage of condensed water.

(3) The temperature of the measuring instrument and of the gases entering it should be that of the room, otherwise the moisture content of the gas will be altered and the readings will be incorrect. In the instruments included under class (2) where a comparison is made with atmospheric air, it is also important that the air used for comparison should have the same moisture content as the gas and be at the same temperature. Means for ensuring this condition are provided in most of the instruments where it is necessary.

(4) The sampling tube and its connections should be kept as short as possible consistently with observing the previous requirements and should be of reasonably small bore. This ensures that there is not much dead space to be cleared of gas before the fresh gas from the flue reaches the instrument and so reduces the time lag of the reading. This is of importance in all indicating and recording types.

Other precautions peculiar to particular types of instrument are usually noted in the makers' books of instructions and need not be discussed at length here.

TYPES OF INSTRUMENTS

(1) Chemical Absorption Instruments

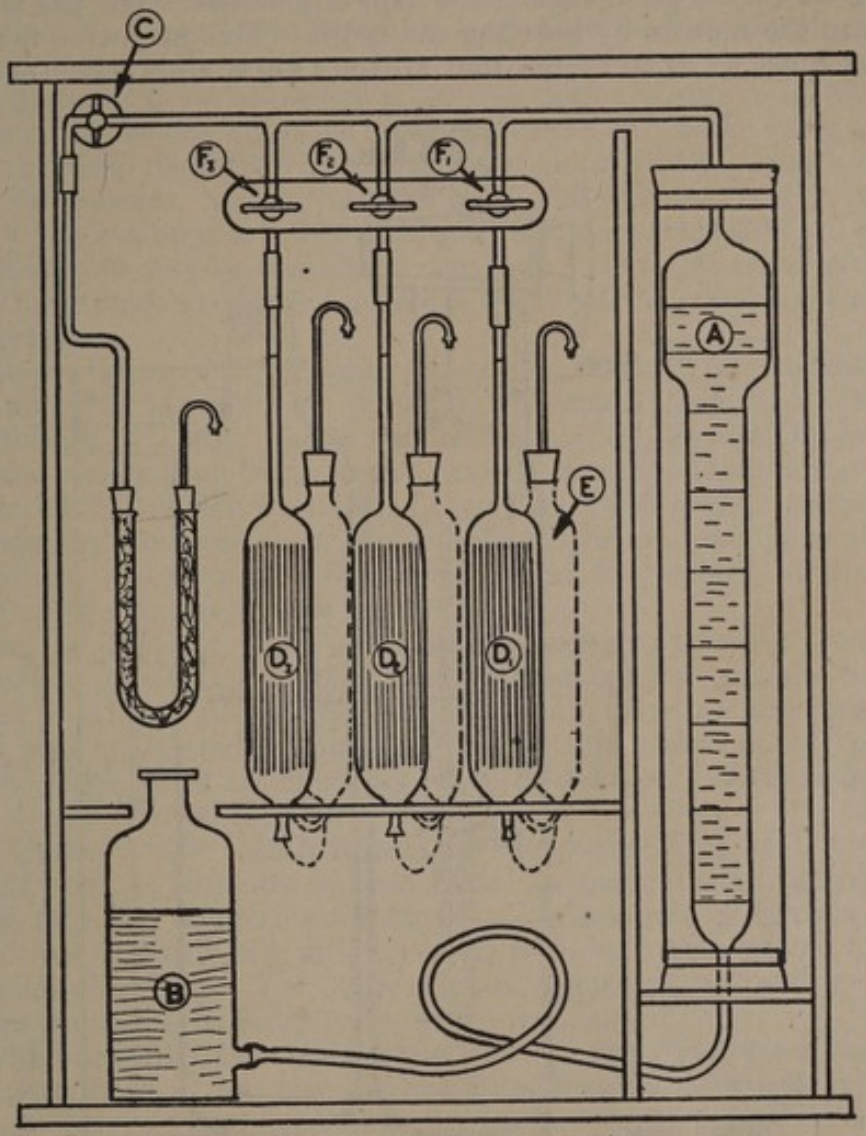
(*a*) *Liquid Absorbent type.* (i) *Spot Reading.* By far the most common type of these instruments is that known as the Orsat (Fig. 279) of which there are minor variations introduced by different makers. It is used for taking spot readings to determine CO_2, O_2 and CO.

Essentially, the instrument consists of a burette (A) graduated from the bottom upwards into 100 divisions. The burette is water-jacketted so as to maintain the gas sample with which it is filled at a uniform temperature. The sample is drawn into the burette by first raising an aspirator bottle (B) so as to fill the burette with water. The three-way cock (C) connecting the top of the burette to the gas stream is then opened and gas is drawn into the burette through a smoke filter by lowering the aspirator bottle. This operation is repeated two or three times to purge the line, the gas being blown out to atmosphere by suitably positioning the cock (C).

Before closing the cock the water level in the burette is adjusted exactly to the zero mark when the level of the water inside the burette and in the aspirator bottle is the same (which can easily be seen by eye if the bottle is held near the burette), so as to ensure that the sample is at atmospheric pressure.

A series of pipettes (D_1, D_2, D_3) contain absorbents which are connected by a capillary tube to the top of the burette, each pipette having its own stop cock (F). The pipettes are partly filled with bundles of glass tubing so as to provide a larger surface covered with a layer of absorbent when the pipette is filled with

gas. An additional limb (E) of the pipette is connected to it at the bottom to receive the absorbent liquid displaced by the gas. The top end of this limb is best closed by a rubber balloon to prevent contact with the external air when



(A)	GRADUATED BURETTE
(B)	ASPIRATOR BOTTLE
(C)	COCK FOR TAKING IN GAS SAMPLE
(D1) (D2) (D3)	ABSORBENT LIQUIDS
(E)	DISPLACEMENT RESERVOIRS
(F1) (F2) (F3)	COCKS FOR BURETTES

FIG. 279. Orsat gas analysis apparatus.

the absorbent is in the side tube. Another type of absorption tube is shown in Fig. 279A.

The first pipette contains a solution of caustic potash to absorb CO_2 (and SO_2). The second contains a freshly-prepared solution of alkaline pyrogallol (" pyro ") for absorption of oxygen. In the third is a solution of cuprous chloride in hydrocloric acid or ammonia for absorption of CO.

The solutions can be bought if a chemist is not available to make them up as required.

The sample of gas is first passed into the pipette containing potash by opening its cock F_1 and raising the aspirator bottle. The gas is then passed back into the burette by lowering the bottle. This operation is repeated two or three times so as to ensure that all the CO_2 is absorbed. The gas is then

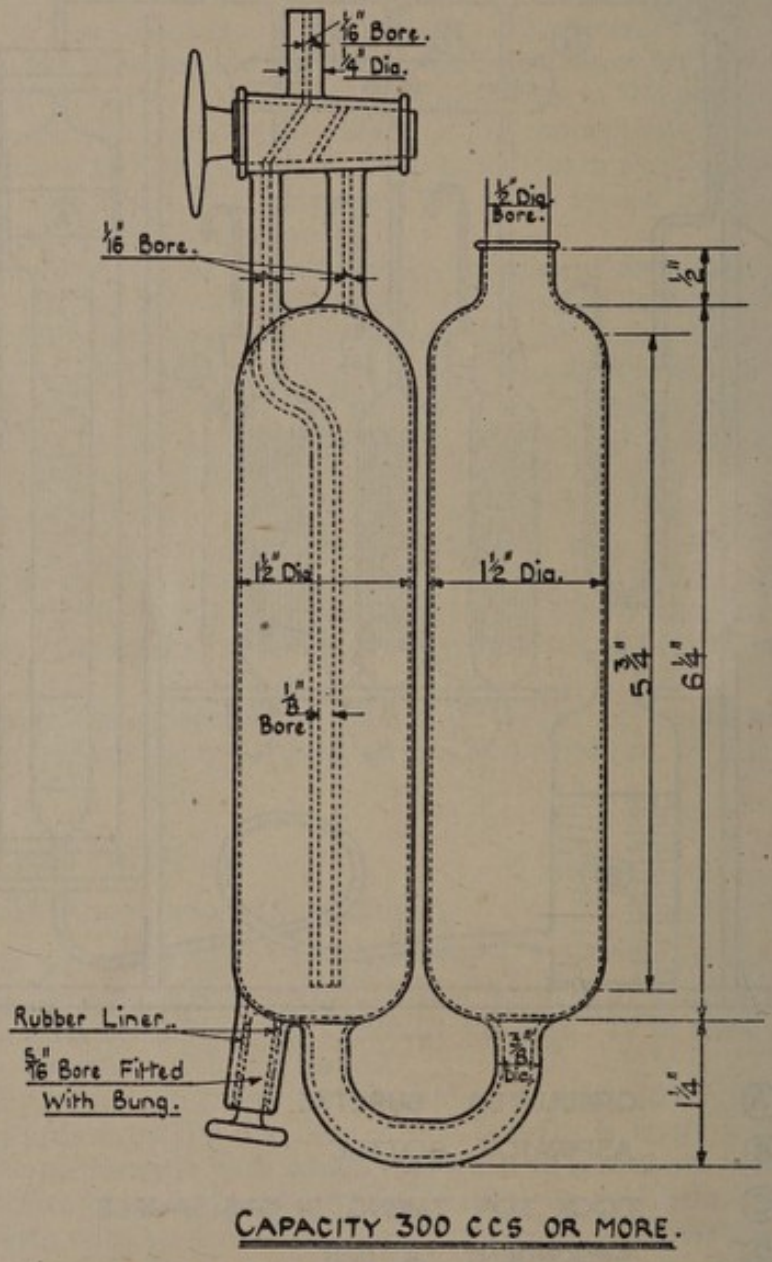


FIG. 279A. Absorption tube.

brought back into the burette and the level in the pipette brought to a fixed point in the tube. The pipette cock is then closed. The water levels in the burette and in the aspirator bottle are then made equal and the reading on the burette taken. This represents the percentage of CO_2 by volume in the gas.

Similar operations are then performed with the pipette containing pyrogallol (opening the cock F_2) to determine the oxygen present. The reading of the burette after this operation represents the sum of the percentage of $CO_2 + O_2$ present and by subtracting the percentage of CO_2 previously obtained that of the oxygen is determined.

Precisely similar operations may then be performed with the third pipette to determine CO, but except in the hands of chemists aware of the sources of error of this determination, it is not recommended that CO determination should be made with the common type of this apparatus. The operations must always be carried out in the order given.

The chief precaution to be taken is to see that absorption is complete. Incompleteness may be due either to insufficient time of contact being allowed or to weak or used-up reagents. The former can be tested by taking a reading after one or two passes, making another pass and taking another reading. When the last two readings are the same, provided the absorbents are in good condition, absorption may be considered complete. Descriptions of variations and improvements made in different makes of Orsat will be found in the makers' books of direction.

(ii) *Recording.* These utilise liquid absorbents and are of various types showing considerable ingenuity in providing automatic operation. Most of these determine the CO_2 only. Some also determine CO after the CO_2 has been determined, the residual gas being passed through a furnace containing copper oxide or other oxidant which burns the CO to CO_2 which is then measured as before and separately recorded ; or two samples of gas are taken, one of which is passed through the furnace. The difference in the records of these two samples then represents the CO burned to CO_2.

The number of different types of these instruments, however, renders it impossible here to give details of their construction and use, and the reader is referred to the booklets issued by the various makers. Most require a good deal of attention and maintenance, and, of course, replenishment of reagents, usually daily. Glass parts are specially liable to damage. When maintained in good and clean condition they are reliable.

(*b*) *Solid Absorbent type.* Instruments for measuring CO_2 by means of a solid absorbent such as Sofnolite or Soda Lime, are made for taking spot readings. In one type for determination of CO_2, a glass bulb is filled with the sample of gas and is connected with a similar bulb by a tube at the top containing the solid absorbent. The bulbs are also connected by another tube at the bottom to which a graduated water gauge is attached.

Starting with one bulb full of gas and the other of water and the water gauge filled to the zero mark at the top, the gas is then passed forwards and backwards through the absorbent by tilting the apparatus alternately in opposite directions. After some passes the water gauge is again read and will be found to be lower owing to some of the water from the gauge having been sucked back into the bulbs to replace the CO_2 absorbed. The percentage of CO_2 present is read off on the gauge. The instrument is very compact and portable.

(2) Inferential Types of Instruments

In these, some property of the gas which changes with chemical composition is measured and the chemical composition inferred from this measurement. Owing to the fact that a flue gas has more than two constituents the readings may be subject to small errors due to the variation of the proportion of the gases other than that one being measured.

On the other hand, they are mostly well adapted for recording and more rapid in response than the purely chemical types.

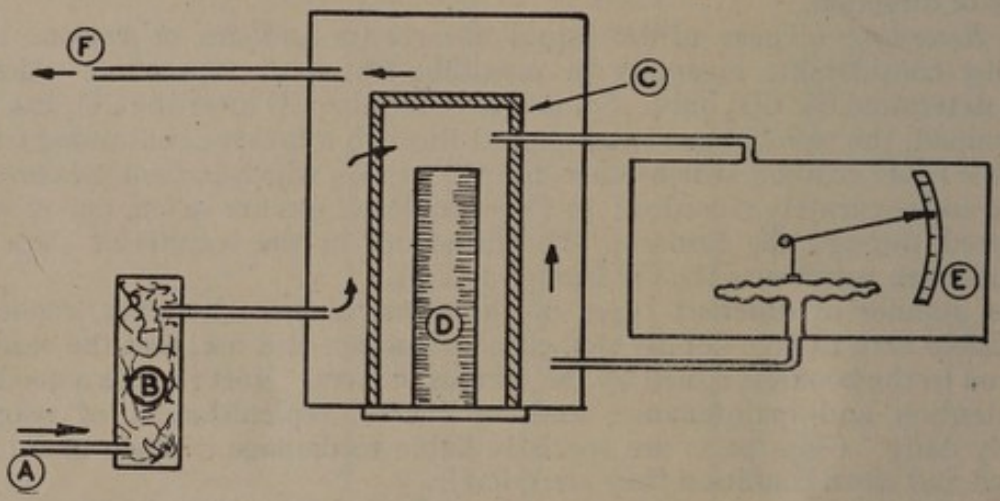
(*a*) *Instruments in which a chemical absorbent is used.* One instrument which depends on a chemical absorbent but in which a physical property is measured is based on diffusion. It measures CO_2 and is made as a portable spot reading, an indicating or a recording instrument.

A porous clay pot (C, Fig. 280) is enclosed in a chamber and in the annular space between the pot and the chamber the flue gas sample or a slow stream of

the gas delivered from (A), through filter (B), is introduced. Inside the porous pot is a cartridge of a solid absorbent for CO_2 (D). The inside and outside of the porous pot are connected to a differential pressure gauge (E). The flue gas in the annular space outside the porous pot diffuses into the pot. Of the gas thus entering the CO_2 is absorbed and the pressure inside the pot is reduced below that of the outside. This reduction of pressure is proportional to the amount of CO_2 in the gas and is read by the gauge or recorded.

These instruments require little more maintenance than the replacement of the cartridges of absorbent at intervals (usually twelve or twenty-four hours for continuous work) according to the type of instrument and the use to which it is put. The portable instrument for spot readings is convenient and compact.

(*b*) *Instruments in which no chemicals are used.* Two largely used types of



(A)	FLUE GAS INLET	(D)	ABSORBENT CARTRIDGE
(B)	FILTER	(E)	INDICATOR
(C)	DIFFUSION POT	(F)	FLUE GAS OUTLET

FIG. 280. Diffusion pot CO_2 recorder.

inferential instruments in which no chemicals are used depend (i) on the density of the flue gas and (ii) on the thermal conductivity of the flue gas.

(i) *" Density " Type.* Two streams, one of air and one of flue gas, are passed into the instrument, the air stream having previously passed through a device to bring it to complete saturation with water vapour at atmospheric temperature. The streams thus contain the same amount of water vapour.

The two streams then enter two chambers in which are placed two similar rotating fans driven by a crossed belt at the same speed, but in opposite directions.

In each of these chambers and facing the rotating fans are two other similar fans on free axes which receive the impact of the rotating gases in the chambers. These two fans are linked together by a light framework so designed that if the turning force given to one fan is greater than the opposite turning force given to the other, the fans and framework will take up a position of equilibrium which is dependent on the difference of the torques. To this frame is attached a pointer which can be used either for indicating or recording. The difference of the torques will be proportional to the difference of the densities of the gases.

As normal flue gas differs in composition from air chiefly owing to some of

the oxygen having been replaced by CO_2, and as the proportion of water vapour is the same in each stream, the reading of the instrument will be proportional to the CO_2 in the gas. Minor errors may arise when in consequence of incomplete combustion there is some hydrogen in the flue gas which, being very light, will lower the density. On the other hand, there may be some small amount of sulphur dioxide present which has an opposite effect.

As a rule, however, the errors due to such causes are not serious, and the instruments give good service and require little maintenance.

(ii) *"Thermal Conductivity" type.* The ease with which thin layers of gas transmit heat by conduction varies according to the nature of the gas and is

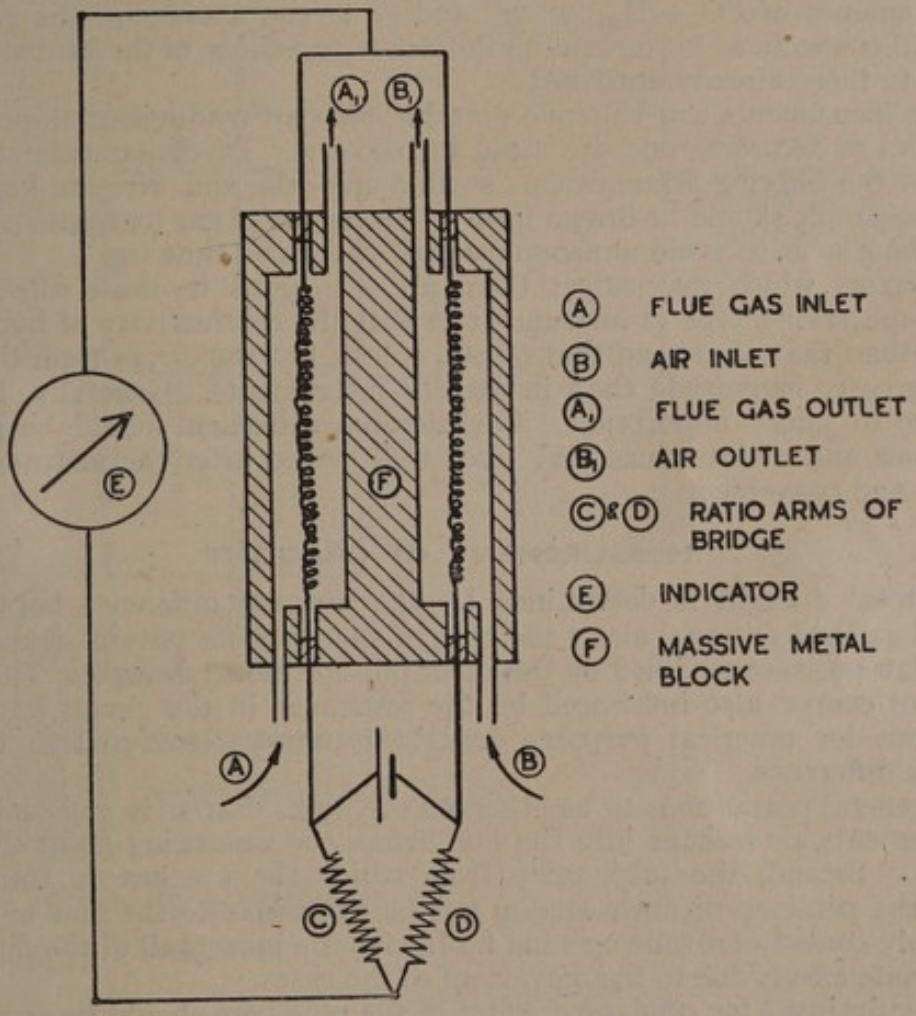


FIG. 281. Thermal conductivity CO_2 recorder.

considerably less for CO_2 than for the oxygen and nitrogen in the air. Instruments are made utilising this property. Two similar wires are slightly heated by the same current of electricity and each wire is placed along the axis of two similar cells (Fig. 281). The cells are in a block of metal to keep them at identical temperatures. Two streams of gas and air (A and B) both brought to the same degree of moisture saturation are passed through the two cells. Owing to the lower thermal conductivity of CO_2 there is less loss of heat from the wire in the cell containing flue gas than from that in the cell containing air. The temperature of the wire in the flue gas cell is therefore higher and consequently its electrical resistance is greater.

Electrical connections are made to the wires in such a way that the relative change of resistance is measured and this will be proportional to the CO_2 in

the gas. As this is an electrical measurement, its reading can easily be transmitted to an indicator or recorder at a distance.

CO present in the gas is frequently also measured by passing the gas leaving the CO_2 measuring cell to a furnace in which the CO is burned to CO_2 and then measuring against air as before in a second pair of cells. The difference in the CO_2 content as measured in the first and second pair of cells represents the amount of CO originally present.

Another method of measuring $CO + H_2$ is based on the fact that when platinum and certain other wires are heated and brought into contact with gas containing $CO + H_2$ and oxygen the combustible gases burn on the surface of the wire and so heat it further. The rise in temperature will be proportional to the amount of $CO + H_2$ burned, and so to the amount in the gas. The electrical connections for measuring the rise of resistance of the heated wire are similar to those already described.

These instruments can be made portable for spot readings or as permanent indicators or recorders and are rapid in response. Particular care should be taken of the filtering arrangements so that the cells and wire are kept clean. Also the sample should be drawn from a live stream of gas by means of narrow bore tubing so as to avoid unnecessary dead space and time lag.

The errors which may affect them are analogous to those affecting the density-measuring type of instrument in that the conductivity of hydrogen is greater than that of air and that of SO_2 is less, but the errors from this cause are of greater magnitude than in density instruments. Generally, however, they are of minor importance. Considerable attention should be given to cleanliness and maintenance, but when well looked after the instruments are reliable and convenient.

MEASUREMENT OF DRAUGHT

The total draught is determined by the pressure difference between the internal gas and external air at the base of the stack for natural draught, and by the fan pressure modified by the stack pull for forced draught. The flow of gas is, of course, also influenced by the resistance in the circuit in which it flows, but for practical purposes draught is always measured in terms of pressure difference.

The general precautions to be observed are, first, that as in gas composition measurements air leakage into the flue before the measuring point should be avoided. Second, the tube entry from which the pressure is transmitted should not point up or down stream but be transverse to the flow of gas and preferably consist of a tube opening flush with the inner wall of the flue. This will obviate effects due to the movement of the gases.

Proper drainage for condensed water in the pipe bore should be arranged as in CO_2 meters.

TYPES OF INSTRUMENTS

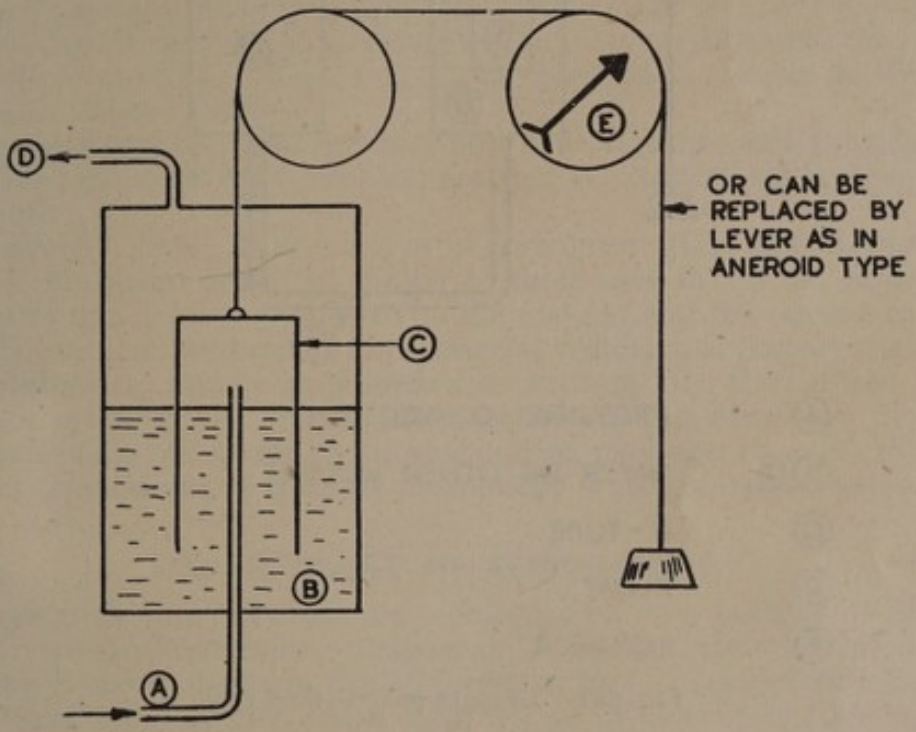
(1) Liquid Gauges

(*a*) *Water Gauge.* Of these the simplest is the ordinary water gauge. This consists of a U-tube half filled with water, one limb of which is connected to the flue gas and the other to the air. A scale is provided and the difference in vertical level of the water surfaces measures the pressure difference in " inches of water."

(*b*) *Inclined Gauge.* Where the difference in pressure is slight or where it is desired to read the figure with greater precision, the ordinary U-tube is so modified that one limb forms a reservoir of comparatively large cross-section and the other consists of a small bore tube inclined at a small angle to the

horizontal. A slight difference of pressure under these conditions causes negligible change in level of the surface of the reservoir, but a considerable movement along the inclined tube for a small difference of vertical height.

Such gauges are often filled with liquids other than water, e.g. paraffin, and must be calibrated by direct comparison with a standard, care being taken (by having a spirit level attached to the gauge) that the angle of the inclined tube to the horizontal is the same when set up as when calibrated. These gauges are simple and reliable and should only require keeping clean and making good any wastage of liquid.



(A) PRESSURE CONNECTION
(B) WATER OR OTHER LIQUID
(C) BELL
(D) TO ATMOSPHERE
(E) INDICATOR

FIG. 282. Draught gauge—bell type.

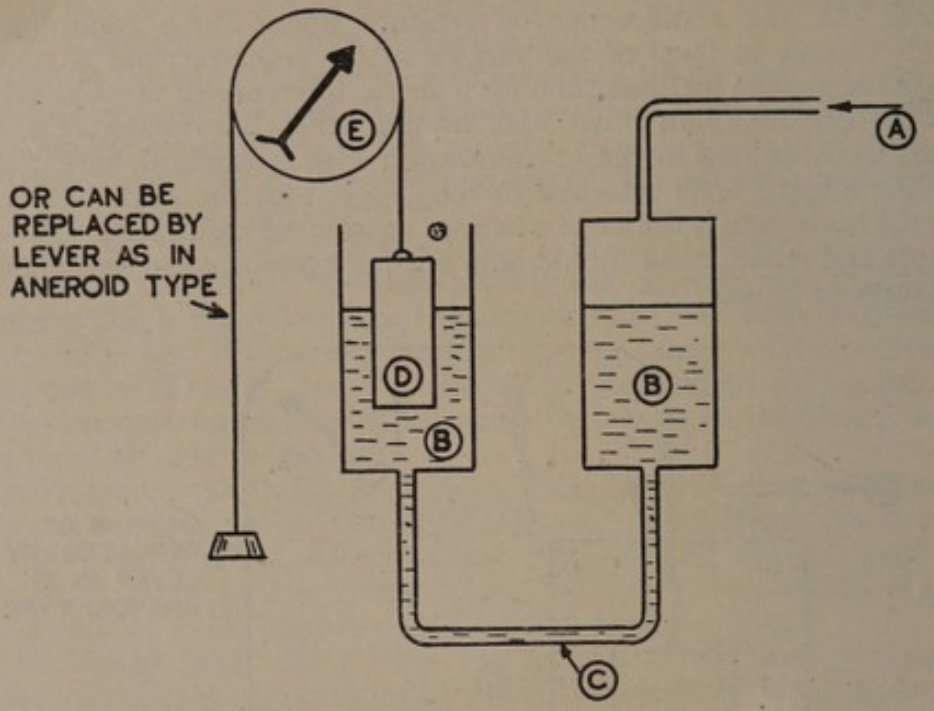
A simple type of inclined draught gauge suitable for boiler and furnace work was shown in Fig. 80.* It is constructed of light gauge sheet metal, all of which, including the inside of the water reservoir, should be protected from corrosion either by painting or stove enamelling. For a gauge to measure a total draught of 1 inch w.g. the open slot should be 9 inches long, the glass tube approximately $\frac{3}{16}$ inch internal diameter and the slope 7 degrees from the horizontal. A plumb bob is included to permit setting the gauge case horizontal.

(2) INSTRUMENTS FOR WALL MOUNTING

Of instruments of the type usually mounted on instrument boards there are three main types.

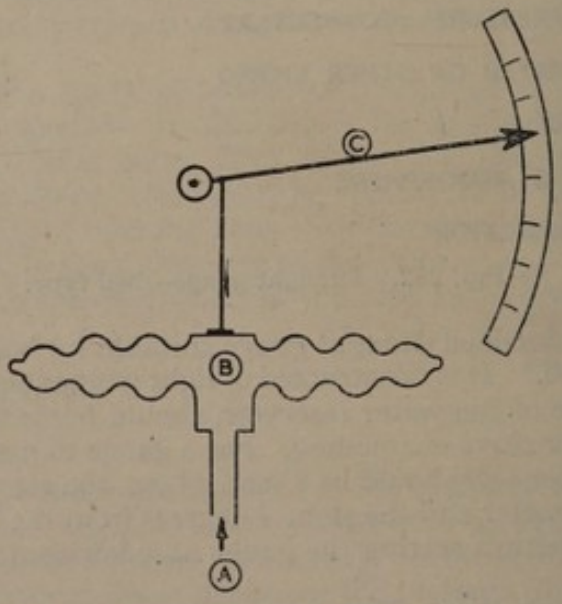
(*a*) *Bell type.* In this type (Fig. 282) the end of the air pressure tube

* Chapter XI., p. 230.



(A)	PRESSURE CONNECTION
(B), (B)	WATER OR OTHER LIQUID
(C)	U-TUBE
(D)	FLOAT
(E)	INDICATOR

FIG. 283. Draught gauge—float type.



(A)	PRESSURE CONNECTION
(B)	THIN METAL DRUM
(C)	INDICATOR

FIG. 284. Draught gauge—aneroid type.

terminates above the liquid level in a chamber in which a bell is suspended in water or other liquid. The flue gas is brought into connection with the inside of the bell by means of a standpipe coming from the bottom of the vessel. When the outside pressure is greater than that within the bell, the bell falls by an amount proportional to the pressure difference and this motion is transmitted mechanically to the dial or recorder on which the pressure is read. It is obvious that by suitable arrangements these instruments can be used for either positive or negative pressure difference.

(*b*) *Float Type.* This (Fig. 283) consists essentially of a U-tube of fairly large cross-section in one limb of which is a float (D). The float is connected by a suspension to a mechanical gearing which operates the indicator (E), so that as the float rises or falls proportionately to the pressure difference the reading is indicated or recorded. The other limb is connected to the flue in which the pressure is to be measured.

Both the float and bell instruments require little attention and, provided that the prescribed liquids of known density are used, the makers' calibration should remain correct.

(*c*) *"Aneroid" Type.* A third type of instrument (Fig. 284) consists of a corrugated thin metal drum (B) similar to those used in aneroid barometers. The inside of this drum is connected to the flue (A) and the outside to atmosphere. The motion in or out of the surface of the drum is proportional to the pressure difference, and is magnified and mechanically transmitted to the indicator or recorder (C). These instruments have the advantage that they employ no liquids, but as the metal of the drum may age, corrode or develop small leaks they should be checked occasionally with a standard water gauge.

MEASUREMENT OF TEMPERATURE

As temperature measurements may be required in a number of different situations it will be convenient to consider them together, but fuller information on this subject may be found in British Standard 1041, "Temperature Measurement." Three types of instrument only will be considered as covering most day-to-day requirements.

TYPES OF INSTRUMENTS

(*a*) *The Mercury-in-Glass Thermometer.* These are made of various types and ranges and can be used over the temperature range from atmospheric temperatures up to about 1,000° F. (or say, 550° C.). The instruments for the upper part of this range are made of special glass and are rather expensive. Their continuous use is not recommended in practice, for which 500° F. (260° C.) should be taken as the upper limit. Positions in which they are found useful are for example for air temperatures, feed water temperatures, and steam temperatures. Thermometers for use above about 250° F. or 120° C. are, or should be, filled with nitrogen under pressure above the mercury which has the effect of impeding the evaporation of the mercury and its recondensation in a cooler part of the stem and of reducing the liability to broken threads. The effect of this pressure, however, is that at higher temperatures when the resistance to distortion of the glass becomes less, the bulb is liable to expand slightly if maintained too long at a high temperature. The consequence is that the thermometer will thenceforward read low, and it is desirable to check thermometers periodically at the temperature of melting ice or steam at atmospheric pressure. Thermometers exposed to high temperatures should where practicable only be used for spot readings and not left exposed to the temperature permanently. Temperatures up to about 400° F. do not produce a very serious change as a rule. Unmounted thermometers

with engraved stem are usually calibrated with the whole of the mercury in the stem and in the bulb at the same temperature.

When the stem and exposed mercury column are at a lower temperature than the bulb a correction must be made to obtain a correct temperature reading, but for ordinary work this may usually be neglected. Other types of thermometers, e.g. some made for insertion in steam pipes, are calibrated with the immersion provided for in the pipe, and this correction is then, as a rule, unnecessary.

The method of using thermometers in a steam or water pipe requires some care if accurate results are to be obtained. The usual method is to have a steel pocket inserted into the pipe in which the thermometer is placed. This pocket should be as deep and narrow in bore as possible and be made of metal as thin as is consistent with the required strength. The pocket should if possible terminate at the outer surface of the pipe and have no external projection. The object of this is to prevent conduction of heat from the closed end of the pocket to the outside of the pipe as far as possible so that the temperature at

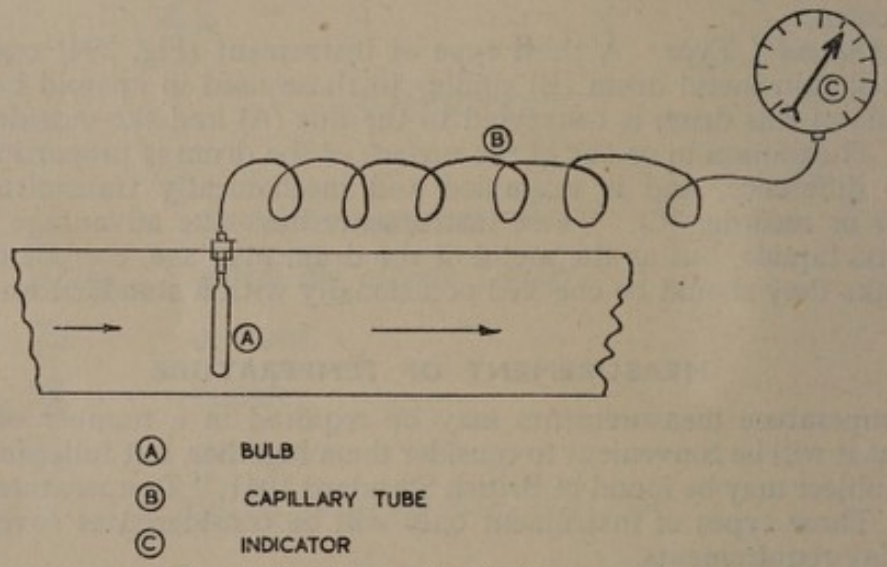


FIG. 285. Mercury-in-steel thermometer.

the bottom is not decreased by a serious amount below the temperature of the water or steam. Lateral conduction should on the other hand be as good as possible, and this is contributed to by the thinness of the walls, the narrowness of the bore, and by filling the pocket round the thermometer with liquid such as mercury or oil which will provide good thermal contact. If mercury is used the top should be covered with oil, as at high temperatures it gives off when exposed sufficient toxic vapour to endanger health in a confined situation.

(*b*) *Mercury-in-steel Thermometers.* In these thermometers mercury is confined in a steel bulb (A, Fig. 285) which is connected by an extremely fine bore tube (B) to a Bourdon pressure gauge (C) by which the pointer on the indicator or the recorder is operated. The whole system is filled with mercury so that when the mercury in the bulb expands on heating the Bourdon gauge expands to accommodate it and so gives the reading.

Owing to the fineness of bore of the connecting pipe the amount of mercury contained in it is small compared with that in the bulb and the fact that the temperature of the tubing is less than that of the bulb only introduces a very small error. The mercury content of the Bourdon gauge is also kept down as far as possible and there is usually a device incorporated with it which automatically corrects for variations in temperature of the gauge. Other automatic compensating devices are also incorporated in the connecting tubing at intervals when any very long runs of tubing are required.

Such outfits will operate satisfactorily over the whole range from atmospheric temperature to 1,100° F. and can be used either for indicating or recording. The precautions for use as far as the question of cooling of the bulb by conduction is concerned are similar to those required by the mercury thermometer, but are usually more difficult to apply owing to the larger size of the bulb and to its being made of a metal of high conductivity. However, pockets are usually unnecessary as the bulbs themselves can be made to screw into steam pipes, etc., without a pocket intervening. Where it is possible to do so, the bulb should be inserted at an elbow in a pipe-line so that a greater immersion can be obtained.

Instruments of this type are also largely used for measuring temperatures of flue gases and in economisers, etc., where the temperatures do not exceed 1,100° F. For these uses the immersion can and should be considerably greater than for use in steam pipes. Mercury-in-steel thermometers have the advantage that their scales are uniform throughout their range and they are robust. Either indicators or recorders can be used. One precaution is, however, necessary in their installation: the gauge should not be at a level greatly higher or lower than the bulb, so as to avoid an alteration in pressure due to the hydrostatic head of mercury between them. Particulars of the limits are given by the makers, but errors from this cause are generally very small and can be corrected.

The capillary tube should be carefully protected from mechanical damage, e.g. by being supported in angle iron. Great care should also be taken when the bulbs are for any reason withdrawn from their working position that no damage is done to the capillary connection.

Another instrument sometimes used operates through the pressure of the vapour of a liquid contained in the bulb operating a Bourdon pressure gauge. The bulb is not completely filled with liquid. These instruments are charac-

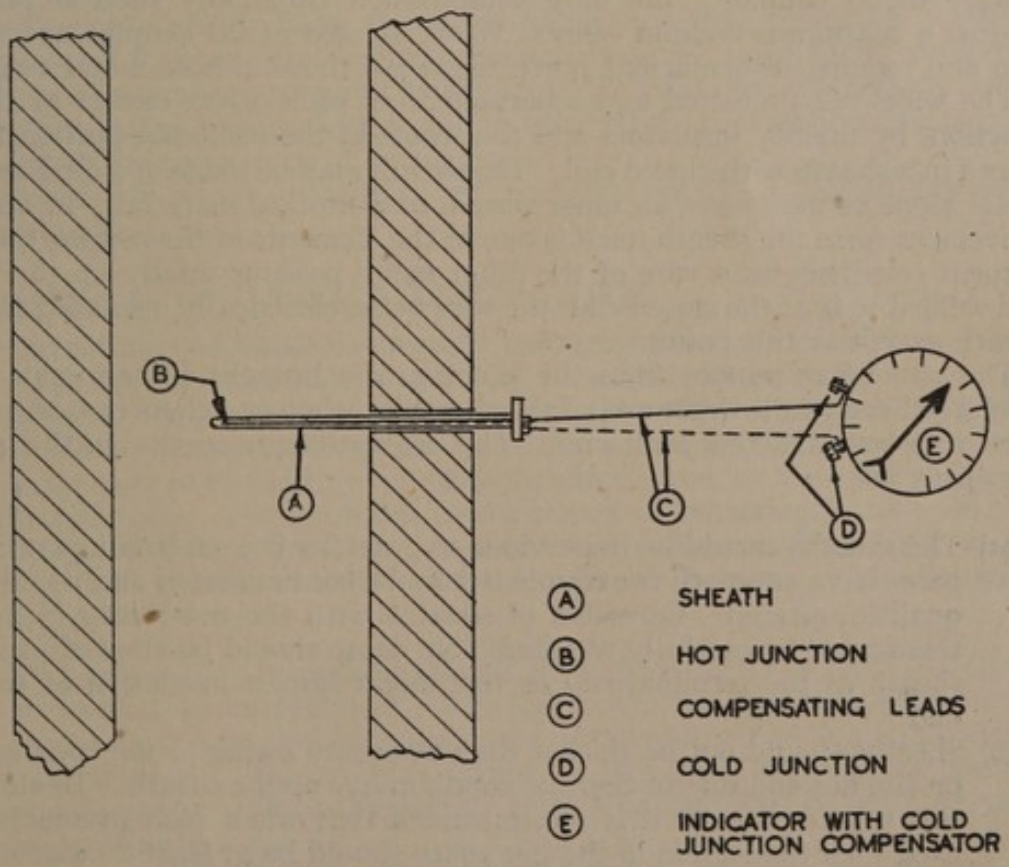


FIG. 286. Thermocouple in flue.

terised by having a scale which becomes more open the higher the temperature, and are suitable where accuracy is desired over a particular range of temperature only, the lower readings being unimportant. It is not proposed, however, to discuss them in greater detail here.

(*c*) *Thermocouple Pyrometers.* These may be employed at any temperature up to about 2,000° F. (or, say, 1,100° C.) for those made of " base " metals and to 2,500° F. (1,370° C.) for platinum metals. For all temperatures above 1,000° F. (or 540° C.) such as occur in some flue gas measurements, the types of instrument described under (*a*) and (*b*) are not available and the only common alternatives are thermocouples or resistance pyrometers. Resistance pyrometers have, however, a considerably lower upper limit of use than the thermocouple type.

The principle on which the thermocouple operates is briefly as follows. Two wires of different metals are joined together at each end forming a closed circuit. One junction, known as the hot junction, is heated and the other junction, known as the cold junction, is kept at a steady lower temperature. An electric current is then generated in the wires, the strength of which is dependent on the difference of temperature between the hot and cold junctions.

If, instead of joining the two wires at the cold junction, copper leads are attached to each at this point, and these leads taken to an instrument such as a millivoltmeter or potentiometer the electromotive force in the circuit which is of the order of a few thousandths of a volt can be measured, indicated or recorded. A description of the usual parts and of the precautions necessary can best be discussed together (Fig. 286).

The thermocouples which are most commonly used are in the following pairs :—

Base metal couples : iron-constantan, chromel-constantan and chromel-alumel. The wires used in these combinations are usually of fairly thick gauge.

Rare metal couples : the only combination commonly used is platinum against a platinum-rhodium alloy. Wires of rare metal couples are usually thin and require more efficient protection than those of base metal couples.

The wires are protected and separated from each other, except at the hot junction, by fireclay insulators and the whole of the insulated portion is then placed in a sheath with closed end. The sheath may be made of a heat-resisting metal alone or may have an inner sheath of a fireclay material. In one very convenient form the sheath itself is one of the elements of the couple, the other element consisting of a wire of the other metal passing axially up the sheath and welded to it at the closed end, the wire being electrically insulated from the sheath except at this point.

The two wires remote from the junction are brought to terminals on an insulated head at the other end of the sheath or when one element is the sheath itself to a special screw connector. The precautions necessary in this part of the outfit are :—

(*a*) The sheaths should be impervious to gases, as if even small quantities of gases have access to the couple it may be contaminated and its electrical qualities altered. Corrosion of sheaths with the possibility of holes in them must be carefully watched. No damp should be allowed inside the sheath at the terminal end as this might impair insulation at the cold end.

(*b*) Sheaths should not be thicker than necessary owing to the cooling effect on the hot end due to thermal conductivity up the sheath. To eliminate this source of error it is recommended that when such pyrometers are used their immersion in the hot space should be at least 20 diameters of the sheath.

The terminals to which the thermocouple elements are attached at the head of the sheath could normally form the cold junction of the couple. If these are in a position where the temperature is likely to alter it is necessary either (*a*) to measure their temperature with an auxiliary thermometer and make the necessary corrections to the readings of the indicating instrument or (*b*) to attach to the terminal instead of copper leads, leads of the same materials as those used in the thermocouple, or of materials of equivalent electrical properties which are termed compensating leads. Such compensating leads have the effect of transferring the cold junction to the terminals of the indicating instrument where temperatures are usually lower and less variable. Several devices have been made and incorporated in the indicating or recording instrument which automatically compensate for the variation of temperature at this point, and this is the method most generally used. (*c*) Alternately, the cold junction can be kept at constant temperature when connected to ordinary copper leads, e.g. by the use of melting ice, steam at atmospheric pressure or by burying it in the earth below the floor or by a thermostatic device.

Precautions regarding leads are that their insulation should be of high quality, that they should not pass through any regions where they are liable to damage by heat or by damp nor should they run in the vicinity of electric power or lighting mains from which there might be leakage to the couple circuit.

The instrument may be either of the indicating or recording type. Generally it is a millivoltmeter, although in some recorders a potentiometer type of instrument with automatic control is used.

Millivoltmeters should have a relatively high resistance to reduce the effect of any variations in the resistance of the couples or leads. Errors may arise from the changes of temperature of the instrument itself unless it has automatic compensation for temperature changes. The scale, when base metal couples are used, is more or less uniform, but with rare metal couples becomes wider at higher temperatures.

Thermocouple outfits have the great advantage that by using a selector switch several differently placed thermocouples of the same kind can be used with one indicating or recording instrument. In recorders the selector switch is usually operated automatically at short intervals so that records can be made of several thermocouples on the one chart.

Thermocouple outfits to operate satisfactorily require a certain amount of skilled maintenance.

MEASUREMENT OF FURNACE TEMPERATURES

It is sometimes desirable to measure the temperature of a fuel bed or other body which may be at or above a red heat. For this purpose a thermocouple is frequently not the best means either because the temperatures are too high or the position is inaccessable. Under such circumstances the radiation is often measured, either in one visible wavelength, usually red, or the total visible and invisible radiation. There are two broad classes of instrument employed :—

(1) *Optical Pyrometers.* These measure the radiation proceeding from a particular hot spot in red filtered light. There are three main designs of instrument :—

(*a*) The spot in the furnace is viewed through a telescope with a red filter interposed in the eye-piece. The light from the spot viewed is brought to a focus at a point inside the telescope at which there is a small electric lamp filament. The external circuit of this lamp contains an ammeter or milliameter and an adjustable resistance arranged in plain circuit or bridge circuit form.

In practice the telescope is sighted on the hot object and focussed.

The lamp filament will then be seen against the red hot background. The adjustable resistance is next altered until the tip of the filament just disappears into the background. The reading of the ammeter, or a direct-reading temperature scale replacing the ammeter scale, is then taken.

The instrument is standardised against a "black body" furnace, or standard instruments, by the makers or the National Physical Laboratory, and a scale of corrections is supplied. An instrument standardised against "black body" conditions can be corrected by known information regarding emissivities for use under "non-black body" conditions.

(*b*) The current in the lamp filament is kept at a constant value and illuminates a definite field of view—say half the total field of view. The light from the furnace passes through neutral tinted absorbing wedges and illuminates a second adjoining field of view. The absorbing wedges are adjustable, so that it is possible to adjust the second field of view until the illumination is equal to that on the first field. The position of a pointer, fixed to the wedges, then indicates the furnace temperature on the instrument scale. Standardisation is carried out as described for type (*a*). In some makes a standard flame from an amyl acetate lamp replaces the electric lamp.

(*c*) The light from a constant source (either an electric or amyl acetate lamp) and the light from the furnace are passed through polarising prisms which polarise the light from the two sources in planes at right angles to each other. Adjacent fields are illuminated by these beams. Both beams pass through a Nicol prism by the rotation of which the strength of one beam is increased and that of the other decreased or vice versa. By rotating the Nicol prism a position can be found in which the intensity of light in the two fields appears the same. The position of the Nicol prism then indicates on a temperature scale the temperature of the furnace. The standardisation is carried out similarly to those of types (*a*) and (*b*).

In using any of these three types of instrument the following points must be borne in mind :—

(1) The pyrometer is standardised on an object which at a given temperature is emitting the full radiation possible, i.e. a body which has an emissivity of 1. This is realised in a furnace the whole of which is at a uniform temperature and the inside of which is viewed through a small aperture. It is known as a "black body." Ordinary objects when not so enclosed at a uniform temperature have an emissivity less than 1, which varies with their nature. The optical pyrometer readings under such circumstances will therefore in general be low, though on highly reflecting bodies with an adjacent source of light at a higher temperature the reading tends to be high, owing to the light reflected from the higher temperature source. Some bodies, however, such as coal or coke, have emissivities in the neighbourhood of 0·9, and even in adverse conditions, such as lack of uniformity in the temperature of a furnace, the readings will not be far from the truth.

(2) None of the light coming from the source under measurement must be absorbed en route by smoke or gases. Smoke has a large effect ; gases (e.g. CO_2, H_2O) have a lesser effect. It is therefore desirable when practicable, to observe through pure air. Windows of any kind have an absorbing effect which must be allowed for. Glare from flames will interfere with optical pyrometer readings.

These instruments are very convenient to use, and may be of great assistance in practice. There are other types of optical pyrometers based on colour changing and colour matching, but these are, however, more difficult to use, and may also introduce personal factors.

(2) *Total Radiation Pyrometers.* These instruments operate on the total radiation, both light and heat rays, emitted by the hot object (Chapter VIII). Bodies of total emissivity less than 1 will emit less radiation than a black body at the same temperature. The same considerations apply to these instruments as were mentioned in connection with optical pyrometers under similar circumstances.

The radiation from the body is focussed by means of a concave metallic mirror or convex lens on to a small thin blackened receiving disc of metal. To the back of this disc is attached a thermojunction of very fine wire. The cold junctions of this couple are brought to a part of the pyrometer screened from the radiation.

In the radiation beam, either before it strikes the mirror or between the mirror or lens and the receiving disc, is placed a fixed aperture which limits the angular dimensions of the cone of radiation employed. This device eliminates variation of the intensity of the image thrown on the receiving disc due to variations in the distance from the object received. There are three main types of this instrument.

(*a*) A metallic concave mirror is used (Fig. 287A) ; the aperture is between the mirror and the receiving disc and at a fixed distance from the disc. This type of instrument has to be focussed to suit the distance from the furnace on which it is used, to ensure that the image of the hot body is

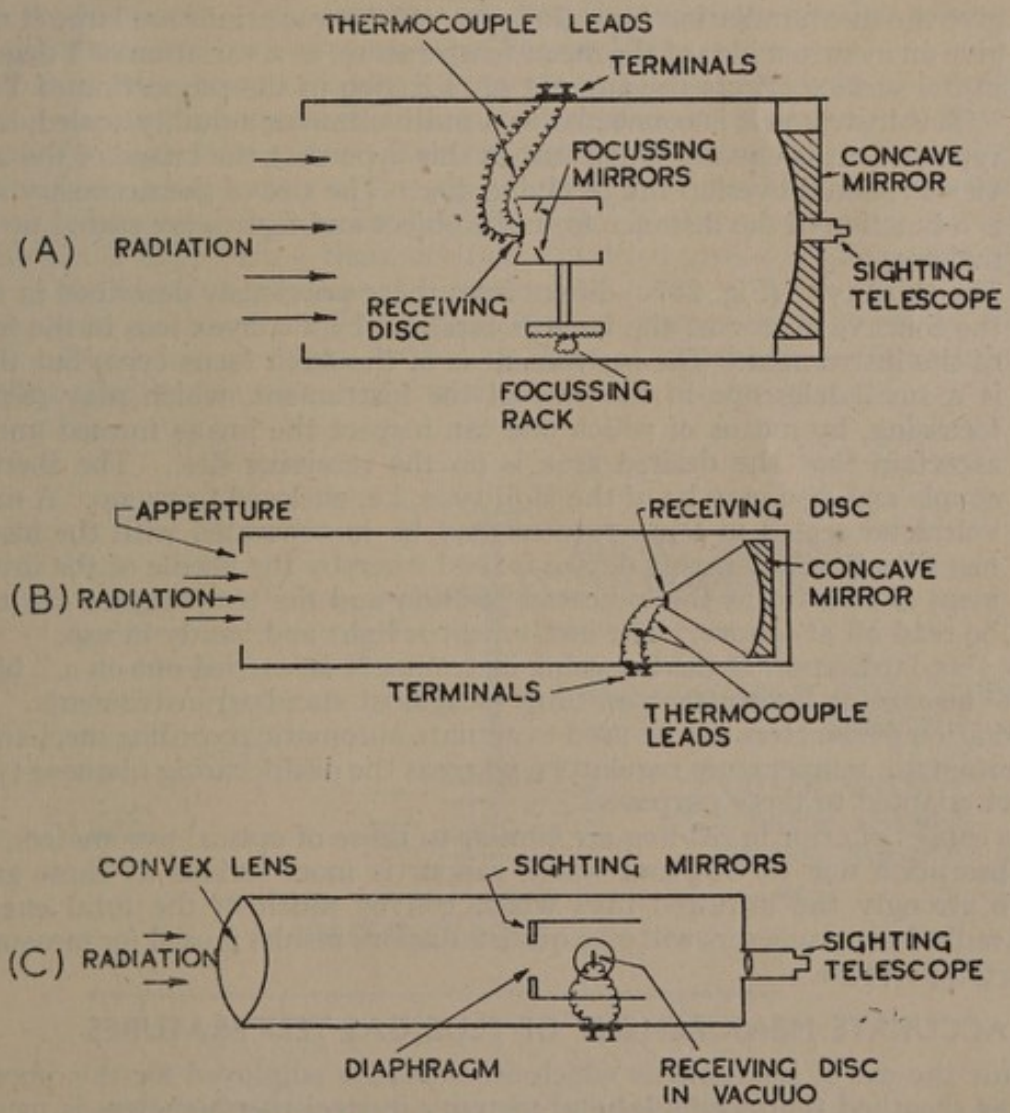


FIG. 287. Three types of radiation pyrometer.

formed on the receiving disc. This is done by moving the receiving disc and aperture assembly, to which two small mirrors are attached, set at an angle; these mirrors reflect the image formed so that it can be viewed through a small telescope, the line of sight passing through the centre of the concave mirror.

The terminals on the instrument internally connected to the cold junctions of the thermocouple are externally connected to a millivoltmeter which may be directly scaled in temperatures.

It is important to see that the image of the object viewed overlaps the receiving disc on all sides.

(*b*) In this type (Fig. 287B) there is a somewhat smaller, but more concave mirror, at the principal focus of which is the receiving disc. The instrument has a longer barrel than the preceding type, and at the one end of this barrel the aperture is placed. The aperture is much larger than in previous type. The length of the barrel is such that when the image of the aperture is formed on the receiving disc the cone of radiation striking the concave mirror has a very small angle.

This instrument requires no focussing, and can be used through a large range of distances from the furnace without error. The farther it is from the furnace, however, the larger is the area necessary for viewing. If there are variations of temperature over the large area it will, of course, give the mean radiation from the area. If the variations are large it may give an incorrect idea of the mean temperature, as a variation of T degrees in the surface affects the amount of radiation in the proportion of T^4.

This instrument is connected to a millivoltmeter suitably scaled, as in type (*a*). It is equally important in this type that the image of the area viewed should overlap the receiving disc. The size of the necessary area is a function of the distance from the object and figures are stated by the makers.

(*c*) The third type (Fig. 287C) differs from those previously described in that the concave mirror at the back is replaced by a convex lens in the front of the instrument. The instrument is of the fixed focus type, but there is a small telescope in the rear of the instrument, which may require focussing, by means of which one can inspect the image formed and so ascertain that the desired area is on the receiving disc. The thermocouple and disc may be of the Moll type, i.e. enclosed in vacuo. A millivoltmeter scaled in temperatures may be incorporated with the instrument itself. A clamping device is used whereby the needle of the instrument is arrested at the indicated position and the temperature can then be read off at leisure. The instrument is light and handy in use.

The standardisation of the foregoing instruments is carried out on a " black body " furnace at known temperatures or against standard instruments.

Radiation pyrometers can be used to actuate automatic recording mechanism and automatic temperature regulators, whereas the disappearing filament types are not adapted to these purposes.

The causes of error in reading are similar to those of optical pyrometers, but the absorption due to CO_2 and water vapour is more serious as these gases absorb strongly the infra-red rays which convey much of the total energy. Total radiation pyrometers will give quite fallacious results if used for measuring flame temperature.

THE ACCURATE MEASUREMENT OF FLUE GAS TEMPERATURES

While the use of instruments which are normally employed for this purpose such as sheathed thermocouples and mercury-in-steel thermometers is usually adequate for control purposes, there are sources of error in such measurements

which invalidate their use when it is desired to know the temperature of the gases with some degree of precision, e.g. in making out a heat balance.

It is necessary to understand the nature of these errors in order to appreciate the reason for the design of instruments which will be referred to below.

In general the temperature of the walls of a flue is less than that of the gases passing along it, as there is a heat loss outwards. A thermometer, thermocouple or other instrument placed in a stream of gas receives heat by convection and radiation from the hot gases and loses heat by radiation to the walls of the flue.

The instrument will then attain a temperature intermediate between that of the gas and of the walls such that the gain of heat from the gas is exactly equal to its loss of heat to the walls. Therefore in order to measure the temperature of the gas as nearly as possible it is necessary to increase those factors which favour the heat transfer from the gas to the instrument and to decrease those factors which favour the loss of heat to the walls.

The chief factors which increase the convection heat transfer from gas to couple are (1) a high velocity of the gas past the instrument, (2) smallness in size of the instrument. The factors which decrease the heat transfer by radiation to the walls are (1) smallness of size and (2) low emissivity for radiation of the surface of the instrument.

Practical methods of measurement based on some or all of these factors will now be discussed. The difficulty of obtaining accurate results, arising from stratification, has been discussed in Chapter IX.

(*a*) *Fine Wire Thermocouples.* Accurate results are unlikely to be obtained by the use of thermometers or sheathed thermocouples owing to their comparatively large dimensions. Bare thermocouples, however, will more closely approach the temperature of the gas, the more nearly so the finer the wire and the lower its emissivity, i.e. the brighter it is. If at the same time the velocity of the gas is high, such a thermocouple will often give a reading very close to that of the true temperature of the gas.

Fine wire platinum couples are best for this purpose as they retain a bright surface, but frequent checks of their calibration should be made owing to the liability of contamination of the wire exposed to the flue gases.

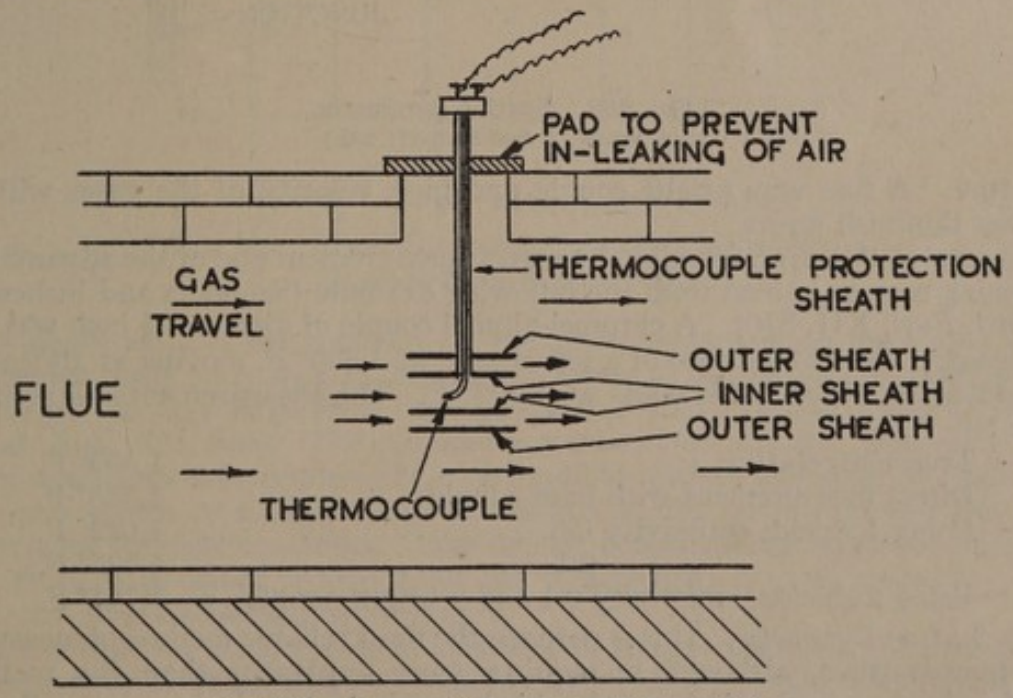


FIG. 288. Screened thermocouple in position.

To get a still more accurate measurement three couples consisting of wires of three known and different diameters are exposed together at points not far apart in the gas stream. Their readings are plotted against their diameters and the curve obtained extended to cut the axis of temperatures at zero diameter. The temperature at this point should be that of the gas, for a wire of infinitely small dimensions should measure the true gas temperature.

(*b*) *Screened Thermocouples:* In this method the object is to cut down the loss by radiation from the couple to the walls of the flue by means of screens.

Two or three brightly polished open-ended concentric tubes of different diameters are so disposed in the gas stream that gas passes through the central one and through the annular spaces between the tubes (Fig. 288). The thermocouple is placed in the axis of the central tube. These tubes all absorb heat from the gas stream and lose less heat by radiation as one progresses from the outermost to the innermost. The innermost will not be far different from the temperature of the gas and the loss by radiation of the thermocouple in the centre to it will be quite small and the reading very close to the true gas tem-

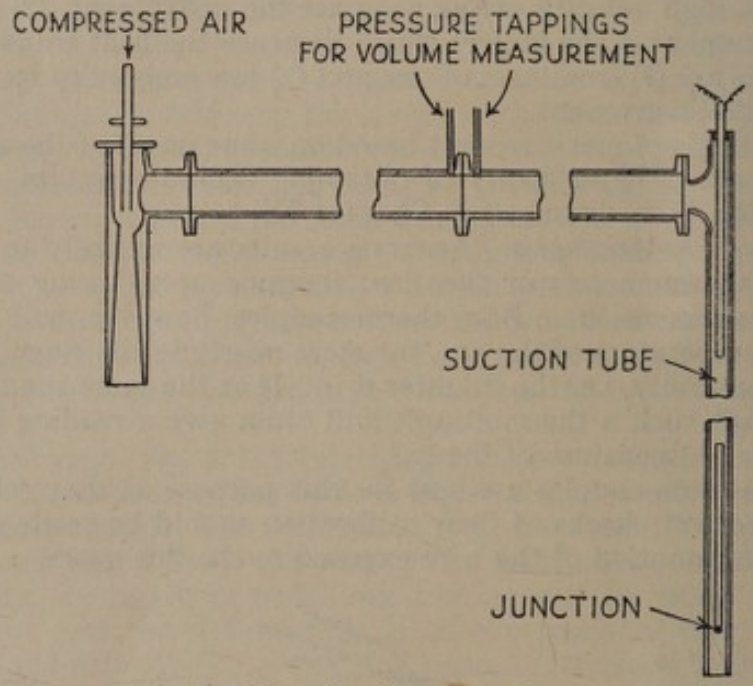


FIG. 289. Suction pyrometer.
(A. Schack, *J. Inst. Fuel*, XII, S31.)

perature. A fine wire bright couple and high velocity of the gases will still further diminish errors.

An idea of the errors possible by direct measurement and of the advantage of screening may be gained from the following example (Saunders and Fishenden, *J. Inst. Fuel*, XII, S10). A chromel-alumel couple of diameter $\frac{1}{8}$ inch was used to measure the temperature of a gas stream at 1,800° F. moving at 20 feet per second down a duct with walls at 1,400° F. The measurements as recorded were :—

True temperature	1,800° F.
Direct measurement with bare couple..	1,568° F.
Using 1 shield, emissivity 0·5	1,622° F.
,, ,, ,, 0·1	1,708° F.
Using 2 shields, emissivity 0·1 and a bright couple ..	1,781° F.

(*c*) *Suction Pyrometer.* This is perhaps the most reliable method of measuring gas temperatures, although it requires more apparatus than the methods previously described. It depends chiefly upon so largely increasing the gas

velocity past the thermocouple that the heat transfer from the gas to the couple is so great that the heat radiation losses from the couple are relatively small.

In this apparatus (Fig. 289) a fine wire bright couple is placed near the mouth of a tube and the hot flue gases are sucked into the tube at a high velocity by means of an external pump or ejector. Screens on the lines previously described are sometimes disposed round the mouth of the tube.

If measurements are taken at various gas velocities it is found that the higher the velocity the higher the reading becomes until a point is reached when a further increase of velocity does not increase the reading which when this point has been reached may be taken as the temperature of the gas.

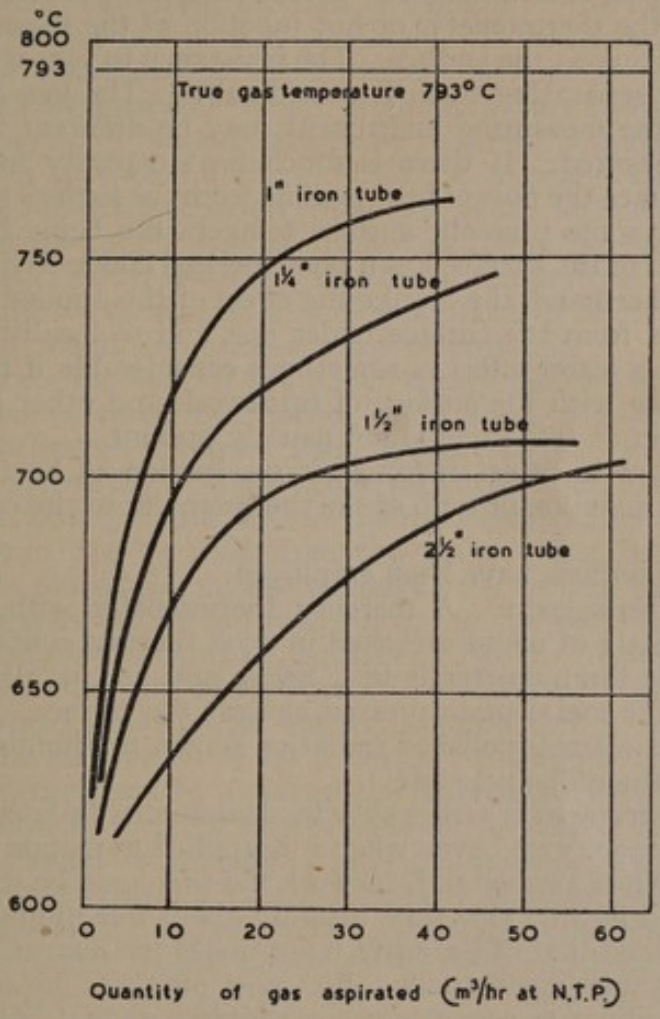


FIG. 290. Errors of readings by suction pyrometer.
(A. Schack, *J. Inst. Fuel*, XII, S36.)

The dependence of the accuracy on the gas velocity and on the diameter of the suction tube surrounding the thermocouple is shown in Fig. 290, which also shows the very large error through using a simple thermocouple (Schack, *J. Inst. Fuel*, XII, S36). The pyrometer was inserted into the flame tube of a boiler, so that the temperature of the surrounding surface was some 200°-220° C. The pyrometer here used was that shown in Fig. 289. For accuracy and particularly for difficult conditions, three or four concentric tubes should be used, through each of which a stream of gas is aspirated at high velocity. The apparatus then becomes complicated.

All these measurements depend on fine bare thermocouples preferably of the platinum metals. The chief drawbacks to the methods are the relative fragility of the couples, their liability to become covered with dust or soot and their

liability to contamination by the gases. They must, therefore, receive frequent cleaning and checking at known temperatures, and it is usually necessary to replace the wires at frequent intervals.

MEASUREMENT OF SURFACE TEMPERATURES

Surface temperatures are most commonly required in connection with the evaluation of heat losses, but may also be required for other purposes. The methods of measurement will therefore be considered separately before describing the methods of measuring heat loss.

The problem of measuring surface temperature is not an easy one. The difficulties arise from several causes. In the first place it is difficult to ensure that the bulb of the thermometer or hot junction of the thermocouple attains the same temperature as the surface. The resistance to passage of heat of even a thin film of air separating them is considerable. The loss of heat from the back surface of the measuring instrument may be different from that of the surface being measured. If there is much heat capacity in the instrument applied to the surface the flow of heat passing from the surface to the instrument may diminish for some time the surface temperature being tested, especially when the material of the surface has a poor thermal conductivity such as brickwork. On the other hand, the blanketing effect of the applied instrument may reduce heat losses from the surface under test and so locally raise its surface temperature. This latter effect is sometimes considerable if the instrument is in contact too long with the surface of brickwork and other poor conductors. With good conductors this effect is not usually present.

Another cause of incorrect readings is conduction of heat away from the thermocouple hot junction or bulb of the thermometer to the cooler parts of the stem or wire.

The following methods have been employed.

(*a*) *Mercury Thermometer.* A mercury thermometer with a small narrow bulb has a thin plate of metal attached in good thermal contact with one side of the bulb. The thermometer is in a frame and lies parallel to the surface under test with the metal plate pressed against the surface. On the opposite side of the bulb is a small polished radiation screen to diminish heat loss from the other side of the bulb and plate.

This type of instrument is rather slow in attaining its full reading owing to its fairly high heat capacity and even when it is applied to metals this time may be some minutes. When applied to brickwork the time may be somewhat greater, but when retained beyond this time (usually about five minutes) the temperature may rise beyond that of the surface not under the instrument owing to the blanketing action mentioned above. For most purposes, however, this instrument will give readings as accurate as are required and is simple and portable.

(*b*) *Thermocouple Instruments.* In these instruments it is usually possible to reduce the thermal capacity compared with the mercury thermometer and there is better thermal conductivity to get the heat to the working part, i.e. the hot junction of the couple. Both of these factors make the instrument less sluggish to use, as it attains a steady state more quickly.

In its simplest form it consists of a thin copper disc, say about $\frac{1}{2}$ inch diameter, to the centre of which are soldered two thin wires, say of copper and constantan, to form the couple. If these are brought away directly from this point there is considerable heat loss from the junction owing to conductivity along the wires. This can be reduced by bringing the wires away along the surface of the disc, insulated from it electrically by a thin film of some enamel which, however, allows fairly good thermal transmission from the disc to the wires. This form may also be provided with a radiation screen to reduce heat loss from the back of the disc. The same effects when applied to good and

bad conductors as were noted with the mercury thermometer may occur but on a reduced scale.

Another type of thermocouple instrument consists of two thin strips of different metals butt welded to each other at one end. The strips are arranged so that they are slightly bent into a hoop which, when pressed with its convex surface against the surface under test with the junction at the central point, ensures that the strips are in good thermal contact with the surface for some distance on each side of the junction. This type having small thermal capacity is rapid in response and can be used on surfaces of poor electrical conductivity.

On metallic surfaces, however, errors may arise owing to part of the return electric current short-circuiting through the metal of the surface under test. Some caution in its use is therefore necessary. It is, however, very rapid in response and can also, in certain circumstances, be applied to moving surfaces.

(*c*) *Embedded Thermocouples.* Where the temperature of a particular surface is frequently required, or when it has to be recorded, a useful and reasonably accurate method is to embed a thermocouple in a groove cut in the surface. Some length of the thermocouple on either side of the hot junction is placed in the groove so that the couple lies just flush with the surface and it is bedded in with material, e.g. fireclay, as closely similar to the material of the surface as possible. Of course for a metallic surface some form of electrical insulation on the wire on either side of the junction must be provided.

The most accurate method for use under steady thermal conditions consists in embedding two or three thermocouples in holes with their junctions at different depths below the surface. Then by plotting the temperature readings of these couples against their depths below the surface and extending the curve till it cuts the axis of zero depth, i.e. the point corresponding to the surface, the surface temperature can be read from the graph. The holes in which the thermocouples are placed should be parallel to the surface to be measured, so that the wires adjacent to the hot junction are at the same temperature.

An accurate form of instrument consists of two thermocouples placed one behind the other. They are wired directly and differentially to the indicating instrument. A small controllable electric heating coil is used to raise the junction end of the instrument to the temperature of the hot surface, the point of balance being determined by means of the differential reading. On switching over the direct reading, the true temperature of the surface is obtained.

(*d*) *Indicating Paints.* A number of paints have been devised which change colour at definite temperatures. Some of these have an irreversible change and so can only be used once. Others have a reversible change, so that in cooling they change back to the original colour at the same change point. Some have more than one change point, so that two or three temperatures can be noted by the same paint.

These may be useful in some positions when it is not desired to know the temperature exactly, but only whether it is above or below the temperature of the change point. The colour changes are also to a certain extent dependent on the duration of their exposure to any particular temperature.

MEASUREMENT OF HEAT LOSSES FROM SURFACES

The loss of heat from surfaces has been fully discussed in Chapter VIII. From what is there said it will be seen that owing to the multiplicity of factors involved the calculation of heat loss is difficult ; it is not proposed to discuss it here.

It is sometimes possible with objects of simple geometrical form and when full knowledge of emissivities, etc., has been obtained to calculate the loss in still air approximately. Calculations are also possible for such things as nests of tubes in moving air. Accuracy, however, can never be great by this method,

and it is generally more accurate to determine heat losses, e.g. from a boiler, as the residual in a heat balance in which all the other items are directly measured.

In simple cases and in still air where it is desired to calculate the loss directly, the usual method is to measure the surface temperature of the object in a number of different places, each place measured being representative of a known area over which the temperature is sensibly constant and consisting of the same surface material. The heat loss from each area is then calculated separately from a knowledge of its temperature, the temperature of surroundings, its emissivity and orientation, etc.

The addition of the losses of these areas covering the whole surface will then give approximately the loss for the whole body. When there is a wind or draught, this procedure becomes impracticable as a rule owing to the un-

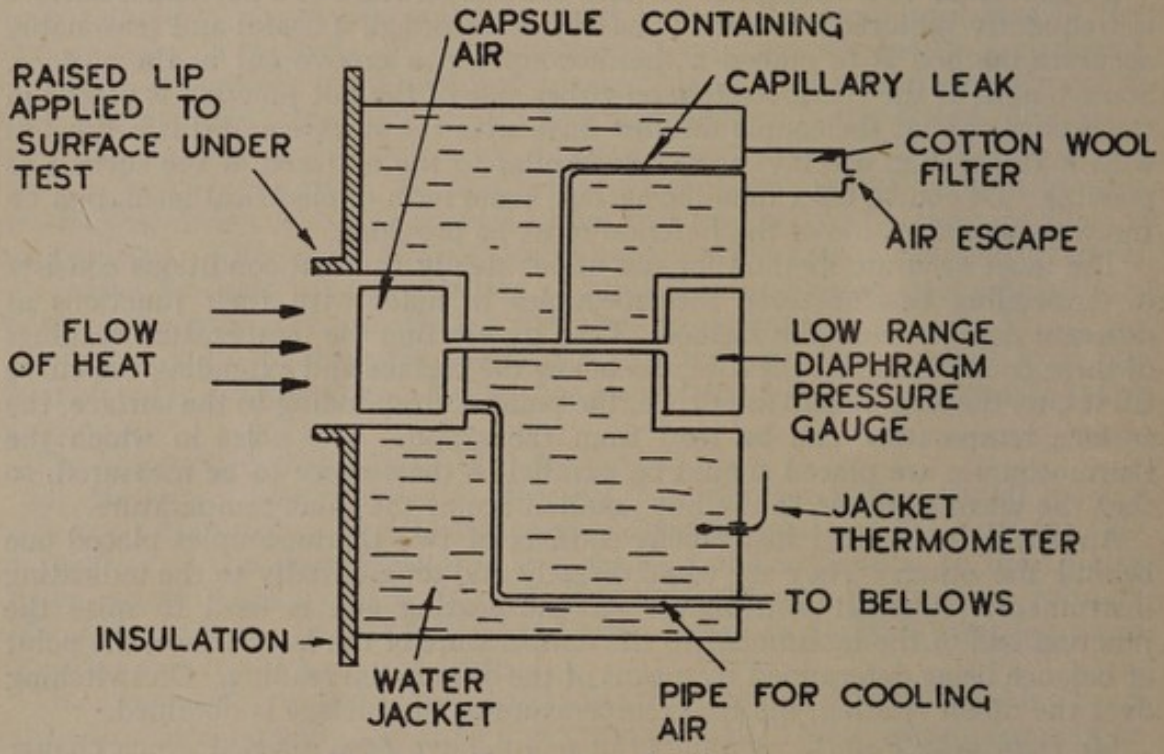


Fig. 291. Blackie heat flow gauge.

certainty of the conditions. Surface temperatures will, however, always form a useful guide as to the magnitude of heat losses.

HEAT FLOW GAUGES

Much of the uncertainty of the calculations previously referred to arises from the difficulty of knowing surface temperatures, emissivities, the appropriate convection constant, etc., which are to be used and endeavours have therefore been made to measure the heat loss directly, so cutting out the assumptions and tedious calculations which the method described above involves.

Two principal types of instrument for use in different circumstances will be briefly described.

(*a*) *A Portable Type.* This instrument (Fig. 291) consists essentially of the following parts. A flat capsule or box of thin aluminium having one surface covered with a thin layer of dull black. The box is placed in a recess in a water jacket with the blacked surface some distance below the mouth of the recess. The front of the water jacket is insulated and has a small raised lip

surrounding the recess, which lip is applied to the surface under test. The back of the capsule is connected (*a*) to a small pressure gauge of diaphragm type reading in inches of water, and (*b*) to a very fine bore glass capillary tube through which air can leak out of or into the capsule. Both of these are surrounded by the water jacket to keep them at a uniform temperature and the capillary leak is protected at the outer end by a cotton-wool filter to prevent any dust being sucked into it.

The operation of the instrument is as follows. The jacket is filled with water, the temperature of which is indicated by a thermometer fixed in the jacket. This temperature should be approximately the same as the temperature of the air and surroundings. The lip of the recess is then applied to the surface under test and heat is transmitted by radiation and convection across the space between the mouth of the recess and the blackened surface of the capsule where it is absorbed. The air in the capsule is thus heated at a rate proportional to the heat loss from the surface and consequently tries to expand at a proportional rate, developing an increasing pressure in the capsule.

Concurrently, leakage begins through the fine bore tube, the rate of leakage depending on the pressure of the air. When a certain pressure is reached the rate of leakage just balances the rate of expansion of the air and the pressure will then remain for a time sensibly constant, after which it begins to fall again owing to the heating up of the capsule diminishing the rate of heat transfer to it.

It may thus easily be seen that the temporarily constant maximum pressure developed is dependent on the rate of heat loss, and it is this pressure which is measured. The heat loss in B.Th.U. per square foot per hour is then read off from the calibration graph supplied with the instrument. The calibration graph is prepared by the makers by applying the instrument to an electrically-heated metal plate to which the power input is measured and its heat loss therefore known accurately at different inputs. The accuracy of the calibration can be checked from time to time by the user in a simple manner as follows :—

The metal box which contains the instrument can be filled with water, which is then brought to boiling point and the instrument is applied to one side of the box. The reading of the instrument should then correspond to a " box point " marked on the calibration graph determined by the makers at the time of calibration.

The use of the instrument has the following advantages : the emissivity of the radiating surface need not be known. The orientation of the surface, i.e. whether vertical, horizontal, etc., is automatically allowed for, and no factor has to be introduced to correct the readings on this account. No calculations have to be made. The radiation loss only can be measured by holding the instrument a few inches away from the surface outside the convection stream, and this loss subtracted from the total loss determined by applying the instrument to the surface will give the convection loss. Different ranges of heat losses can be measured by the substitution of capillary leaks of different bore each having its own calibration graph.

In making a determination of the heat loss from a body, the body is divided up into areas over which the material of the surface and the surface temperature is sensibly the same and the total heat loss is then the sum of the losses of the individual areas.

The chief precautions which have to be taken are as follow : As the temperature of the water in the jacket takes the place of the temperature of the surroundings it should be kept as near that of the latter as possible. As in a long series of readings the water temperature gradually rises, it is desirable to change the water from time to time. Other possible causes of error are, that a leak other than that provided for by the capillary may develop, e.g. from the

capillary not being completely screwed home on its seating; or that dust enters the capillary and so adds extra resistance to the escape of air through it. Either of these errors can be detected by taking a " box point " before a series of measurements. If there is a leak a low reading will be obtained and if there is a blockage in a capillary a high one.

(*b*) *Fixed Types.* These types work on a different principle with some slight variations in different makes. In general a thin pad or bandage of silk or other material is applied to the surface. Differential thermocouples or electrical resistance thermometers are fixed on the inner and outer surfaces of the bandage, the thermal conductivity of which is known. From the measurement of the difference of temperature between the two sides of the bandage the heat flow through it is then simply ascertained by multiplying by the factor supplied. This heat flow is obviously the same as the heat loss from the other side of the bandage.

The instruments have the great advantage that they can be used to record the heat loss continuously in the situation selected and that they are most convenient for ascertaining the heat losses from pipes for which the previous type described is not so well suited. Their drawbacks are that when first applied it may take a little time to establish a steady state of heat flow through them. The radiant emissivity of the outside of the bandage may not be the same as that of the surface being measured. This and the added effect of the bandage as an extra resistance to heat flow from the surface may make the heat loss from the bandage somewhat different from that from the free surface under measurement.

They are, however, particularly useful when a continuous record of losses from the same point under different temperature conditions has to be obtained.

MEASUREMENT OF SMOKE

The presence of smoke indicates that there is incomplete combustion and loss of fuel. This loss is not confined to the visible solid and liquid particles of the smoke, but also to the unburned gases such as CO, H_2 and hydrocarbons which generally accompany them. The loss from unburned gases may be even greater than the loss from visible unburned products.

It is evident, therefore, that some form of smoke indicator or meter may take the place of a CO and H_2 meter. To be of use to the fireman the indication must be available to him where he works.

TYPES OF INSTRUMENTS

(*a*) *Photo-electric Cell.* The best form of instrument for this purpose operates in the following manner. A beam of light of constant intensity is projected through a hole in the side of the stack and emerges through a similar hole on the opposite side, windows being provided at each hole. The windows are kept clean from smoke by a draught of pure air which is sucked into the stack behind the glass. The beam of light coming out of the stack falls on a photo-electric cell which produces an electric current dependent on the amount of the light falling on it. This current, sometimes amplified, operates an electrical indicating or recording instrument. The amount of smoke in the path of the light beam will control the intensity of the light falling on the photo-electric cell and the indication of the instrument can be taken as a measure of the smoke.

The radiation pyrometer may be used in place of a photo-electric cell in this type of instrument.

These instruments can give very accurate indications or records of the smoke, but require considerable maintenance and attention to keep them

working properly. Such points as seeing that the windows into the stack are keeping clean are of obvious importance, but it is impossible here to mention other details of precautions which will be supplied by the makers.

(*b*) *Direct Observation Type.* (i) Often, however, simpler and less expensive means can be found to inform the stoker of the production of smoke. One such plan is to have a beam of light similar to the above which on emerging from the chimney is projected on an opal glass screen. The appearance of the screen can be viewed by the stoker either directly or by means of mirrors. From its appearance he can easily judge of the intensity of the smoke being formed.

(ii) The smoke issuing at the top of the stack can frequently also be viewed in the boiler room by means of mirrors suitably disposed. Some such simple arrangement is well worth while even for small plants.

WATER AND STEAM

MEASUREMENT OF BOILER FEED WATER

The measurement of water quantities by orifice plate and V-notch has been described in Chapter X. A wide variety of other instruments and appliances is available, some of which are here described.

(1) *Reciprocating Meters.* Two pistons in two cylinders side by side move in and out alternatively, each stroke being equivalent to the volume of the cylinder swept by the piston. An integrating meter is operated by the pistons and from its readings the volume of water passing in any time interval can be ascertained. Leakage past the pistons may be one source of error in such instruments and as the quantity is based on the measurement of volume some correction for temperature may be necessary when interpreting this as weight.

(2) *Elliptical Piston Meters.* In this type the moving part consists of a balanced elliptical piston, the action being such that the water flowing through the meter rotates the piston, the speed of rotation being directly proportional to the volume passed. The rotation of the piston is transmitted to a registering counter through a train of wheels, and the counter dial can be arranged to register the total quantity passed in any desired units.

Arrangements can also be made for the readings to be recorded at equal time intervals on a graph from which the rate of flow at any time can be deduced.

Both these types of instrument may be of considerable accuracy. They are not affected by small pulsations in the flow and may be used under considerable pressures. They do not require much servicing.

(3) *Inferential Rotation Meters.* Other common types depend on the rotation of a water wheel or a revolving helix or screw whose revolutions are a measure of the amount of water flowing. Such meters are subject to errors arising from slip and frictional resistance and should be checked periodically, e.g. by direct weighing. No meters of this type will give accurate measurements when the flow of water is pulsating, e.g. from the action of a reciprocating pump, unless steps are taken to damp out the fluctuations.

MEASUREMENT OF STEAM

The measurement of steam by orifice plate is discussed in Chapter X.

MAINTENANCE OF INSTRUMENTS

The maintenance of all instruments in good and accurate working condition is a matter of primary importance. An instrument which is giving incorrect readings may mislead to such an extent that more waste occurs than if no instrument had been installed and the unsupported judgment of the operator had been relied upon.

In larger works, in which the services of properly trained engineers or chemists can be called upon, maintenance and testing of instruments can be left in their hands. In smaller works, however, where no such trained personnel is available, it will as a rule be found better to use the simpler types of instrument only, with instructions to the most suitable member of the staff to follow the directions for maintenance supplied by the makers of each instrument.

Instruments must be kept clean and a routine inspection made. Many firms of instrument makers, especially those making the more complicated kinds, have a servicing department, and where this exists, full use should be made of it. The more elaborate instruments require servicing by the makers.

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CHAPTER XXXI

SAMPLING SOLID FUELS

Principles of sampling and standards of accuracy—Sampling (*a*) large and (*b*) small coal for all purposes other than screen analysis—Preparation of laboratory sample for chemical analysis —Sampling from waggons—Sampling for screen analysis—Sampling coke for moisture, analysis and shatter test.

PRINCIPLES OF SAMPLING AND STANDARDS OF ACCURACY

THE object of sampling is to obtain a small portion that is truly representative of the entire bulk of the material. This is not difficult when the material is homogeneous, but it becomes exceedingly difficult with a heterogeneous material such as coal, which consists of pieces of coal, shale and other impurities varying in size from large lumps to fine dust, and which, moreover, is unevenly mixed.

To obtain a representative sample, the method formerly adopted was that known as " quartering," in which a considerable bulk of material was thoroughly mixed, made into a cone, flattened down and quartered. Opposite quarters were selected, broken down to a small size, mixed and the procedure repeated. This was continued with considerable labour until a sufficiently small sample was obtained. This method is still regarded as sound where mechanical devices are not available for such reduction of bulk.

The methods recommended are described in British Standard Specification 1017 of 1942. Upon this and other specifications to which reference will be made, is based the general description of the principles of sampling coal and coke given in this chapter.

The modern method of sampling involves taking a number of small portions uniformly distributed over the whole waggon or consignment sampled. The sum of these small portions builds up the total initial sample. The sample thus " grows " from the accumulation of the smaller samples, and by analogy with mathematical concepts these smaller portions are termed " increments."

The degree of accuracy of the sample is greater when a larger number of increments is taken, and for coals of greater heterogeneity more increments are required for a given degree of accuracy. The more heterogeneous the coal, the greater will be the divergence between the ash content of the " increment " samples.

It has been shown experimentally that the average error of a coal sample increases with its ash content, so that the number of increments necessary for a given degree of accuracy is greater for a high-ash than for a low-ash coal. An approximate quantitative relationship has been established experimentally between the ash content and the average error so that, knowing the ash content of a coal, it is possible to calculate, by means of the theory of errors, the number of increments required for a given degree of accuracy of the gross sample.

The standard of accuracy adopted by the British Standards Institution is such that 99 times out of 100 the ash percentage of the gross sample falls within $\pm$ 1 unit of the true ash content of the coal sampled. In figures this means, for example, that if the true ash content is 7 per cent., the degree of accuracy to be expected in 99 cases out of 100 is such that the ash content of the coal sample ultimately obtained will be between (7 + 1 =) 8 per cent. and (7 − 1 =) 6 per cent. (i.e. 7 $\pm$ 1 per cent.). The scheme of sampling here outlined corresponds to this degree of accuracy.

The minimum weight of each increment must be such that the chance inclusion or exclusion of a piece of shale or impurity will not have an undue

effect upon the ash content of that increment as representing the portion of coal from which it is drawn. Thus, large coal requires bigger increments than small coal. The principle of the size/weight ratio upon which the size of an increment is based, is discussed in connection with the subdivision of samples, when it becomes of great importance.

The theory of sampling and its applications are fully described in B.S. 403, which should be consulted for further information.

The methods adopted for sampling depend upon the purpose for which the sample is required. When it is required to make a chemical examination of the coal, e.g. to determine its ash and moisture content and its calorific value, the limits of accuracy required are different from those implied in taking a sample for screen analysis, since these latter depend only on the size of the coal and not upon its heterogeneity. A simplified method of procedure can therefore be adopted when the sample is taken for screen analysis only.

The taking of a sample for chemical analysis is governed by the distribution of the ash. Thus, the procedure must be varied according to the size of the coal and the ash content. It is clearly more difficult to sample large coal accurately than smaller coal; the greater the ash content, the more are the number of pieces of shale, and thus the greater the number of increments that must be taken to attain the necessary accuracy.

The treatment that will be adopted in this chapter will be in accordance with the following scheme :—

(A) Sampling of coal for chemical analysis and all purposes other than screen analysis. Slightly different procedure is used for—
  (1) Small coal below 3 inches.
  (2) Large coal above 3 inches.

  The treatment of this sample will then be considered. The moisture must first be determined.

  A smaller sample will then generally be required for chemical analysis other than moisture, and special precautions are necessary to obtain this sample accurately from the larger sample already secured.

(B) Sampling of coal for screen analysis only.

  The simplified methods of taking the sample will be described and information given on the method of conducting the size analysis.

(C) Sampling of coke.
  (*a*) For chemical analysis.
  (*b*) For a shatter test.

A. SAMPLING COAL FOR ALL PURPOSES EXCEPT SCREEN ANALYSIS

(1) *Sampling Small Coal up to* 3 *inches.* To comply with the limits of accuracy as defined by the B.S.I., the number and weight of the increments taken must be based upon Table 124.

This degree of accuracy will not generally be required for high-ash coals, and it is recommended that when less accuracy is sufficient, only half the quantity should be taken; the true ash content will then be within 1·4 per cent. of the true ash content in 99 times out of 100, instead of within 1 per cent.

The known or reputed ash content of the coal is given at the head of columns 3 to 6 in this table. When the average ash content is unknown, either (1), the largest sample of its size group should be taken, or (2), a preliminary sample should be taken to determine the approximate ash content.

The increments should be collected by means of a small shovel or measuring scoop holding not less than the appropriate weight as specified in the table. Where possible the sample should be taken when the coal is moving as on a

TABLE 124. SMALL COAL

Size of coal	Weight of increment	Minimum weight in lb. of gross sample and minimum number of increments			
		Ash content—per cent.			
		0–6	6–10	10–15	15–20
3 inches	6 lb.	66	144	252	390
2 ,,	4 lb.	44	96	168	260
1½ ,,	3 lb.	33	72	126	195
1 ,,	2 lb.	22	48	84	130
½ ,,	1 lb.	11	24	42	65
Minimum number of increments ..		11	24	42	65

chute or conveyor, the shovel * being plunged into the flowing stream of coal at regular intervals. Errors arising from segregation should be avoided by taking alternate increments from the sides and middle of the conveyor or chute. It is even better to sample at a point where the coal is falling and to collect it in a bucket or box.

Every care should be taken to prevent exposure of the sample to the atmosphere, which would lead to loss of moisture. This is particularly important when the coals are obviously wet superficially; when extreme accuracy is desired each increment as taken should be placed in an air-tight container. If the sampling of wet coals occupies a considerable time—an hour or so—it may be necessary to do this in any event.

Where the coal cannot be sampled from a flowing stream, increments may be taken at regular intervals during hand discharge. Sampling from the tops of waggons can never be recommended, because it is impossible, in this way, to obtain a representative sample of the coal unless it is more or less homogeneous, as graded or washed coal may be. If, however, no other method is possible, holes at least 18 inches deep should be dug into the coal and increments taken by scraping the sample shovel from the bottom to the top of the steepest side of the hole, or, alternatively, from the bottom of the hole.

(2) *Sampling Large Coal over 3 inches including uncrushed run-of-mine.* Large coal should be sampled as the waggon is being discharged. To obtain the B.S.I. standard of accuracy, the procedure as regards number and size of increments should be based upon Table 125.

TABLE 125. LARGE COAL

Ash percentage	Minimum weight of gross sample—lb.				Minimum number of increments			
	Large coal screened over			Unscreened run-of-mine	Large coal screened over			Unscreened run-of-mine
	3 in.	1½ in.	½ in.		3 in.	1½ in.	½ in.	
Up to 4	100	100	100	—	10	10	10	—
Between 4 and 5 ..	150	180	180	220	15	18	18	22
,, 5 ,, 6 ..	240	300	300	360	24	30	30	36
,, 6 ,, 7 ..	330	410	410	500	33	41	41	50
,, 7 ,, 8 ..	430	530	530	640	43	53	53	64
Over 8	500	630	630	750	50	63	63	75

* Errors may arise through nutty coal tending to run off the edges; a specially constructed scoop is preferable, the bottom conforming to the curvature of the belt, and having sides built to prevent the larger material flowing away.

When the coal contains lumps exceeding 10 lb. in weight, the increment should contain a section, or sections, cut evenly across the bedding planes of the large pieces, and any attached or inter-stratified shale should be included. When sampling a mixture of large and small coal, the increments of the large and small coal may, if desired, be placed in separate containers, and the final sample prepared by taking proportionate amounts of the lumps and the small coal on the basis of an assessment, made by the sampler, of the relative proportions of each in the waggon.

TREATMENT OF THE GROSS SAMPLE

From the gross sample which, up to this point, should have been protected from exposure and loss of moisture, a separate sample must be taken for the moisture determination, after which the remainder is prepared for ash and other determinations. The correct determination of moisture is important in order to obtain the calorific value of the coal as received.

(1) *The Preparation of the Sample for Moisture Determination.* If the coal is over 3 inches in size it may be crushed to below this size as a first step; otherwise it is not treated in any way until the moisture sample has been taken.

To take the sample for moisture determination, the whole gross sample is poured without any previous mixing on to a plate to form a cone, and flattened out. Increments are then taken from points evenly distributed over the flattened heap, as follows :—

			Total
(*a*)	From coal up to 1 inch,	10 increments of 1 lb. each ..	10 lb.
(*b*)	,, ,, ,, 2 inches,	10 ,, ,, 2 ,, ,, ..	20 ,,
(*c*)	,, ,, ,, 3 ,,	10 ,, ,, 3 ,, ,, ..	30 ,,

The amount of the gross sample, as shown in the foregoing tables (124 and 125) should be increased by an amount equal to the moisture sample as given above, except when the gross samples are large relative to the moisture sample.

Unless the coal is already smaller than ½ inch, the sample must be crushed to ½ inch before further reduction. If the coal is visibly dry this may be done by crushing directly, either mechanically or by hand, but if the coal is wet it must first be air-dried.

To carry out this procedure the whole sample of 10–30 lb. must first be weighed. It is then spread out in a thin layer and exposed in a dry room at atmospheric temperature for six to ten hours, or overnight; alternatively, it may be put in a drying oven at 50° C. for 1½–3 hours. The dried coal is then re-weighed to ascertain the weight of moisture that has evaporated.

From the determinations of loss of moisture on air-drying and of the moisture content found in the laboratory sample of the air-dried coal, the total moisture content of the moisture sample may be calculated as follows :—

If X = percentage loss of weight in air-drying,
Y = percentage moisture in laboratory sample ground to pass a 6 B.S. test sieve.

Then total moisture content of moisture sample $= X + Y\left(1 - \frac{X}{100}\right)$.

If the original coal was above ½ inch in size it is now broken. The dried coal, now below ½ inch in size, comprising a sample of 10–30 lb., is then reduced to 2 lb. by taking not less than 10 increments, evenly distributed, by means of a spoon or small scoop. This sample is placed in an air-tight container and sent to the laboratory.

(2) *Preparation of the Laboratory Sample for Moisture Determination.* There are three methods available for the determination of moisture in coal (pp. 10–13,

B.S. 1016), two of which use the 2-lb. sample of ½-inch coal, as prepared above. Where these methods are used, therefore, no further treatment of the moisture sample is necessary.

The third method involves crushing the coal to pass a 6 B.S. sieve, after which a small portion is used for the determination of moisture.

If the original coal was under ½ inch, the 2-lb. moisture sample will have been taken without air-drying and may be visibly wet. If so, it must be air-dried as previously described before proceeding to use the third method. Otherwise the moisture sample will have been already air-dried.

In order to avoid appreciable loss of moisture during reduction in size, the air-dried coal should be sieved rapidly through a 6 B.S. sieve, and the undersize immediately replaced in its tin whilst the oversize is rapidly crushed to pass a 6 B.S. sieve in a flat mortar using a flat pestle. This is also returned to the tin and mixed by shaking. This 6 B.S. coal is then reduced to 8 oz. by taking not less than 10 increments evenly distributed, and used for the moisture determination.

PREPARATION OF THE LABORATORY SAMPLE FOR CHEMICAL ANALYSIS

Now that the moisture sample has been taken, there is no need to avoid exposure to the air, and if the coal is too wet to grind readily it may be air-dried without determining the loss of weight.

It must be ground, either directly or in stages, to pass a 36 B.S. sieve for determinations of ash and moisture or to pass a 72 B.S. sieve for other analyses.

At this point a distinction may be made between crushing and grinding. Crushing, which is used for moisture samples, implies that the coal is broken up by impact, either in a jaw crusher, or by hand in a mortar. This causes the minimum of exposure to air, but may occupy more time than grinding, which breaks up the coal between moving surfaces and causes considerable exposure to air. Either grinding or crushing may be used for the sample for chemical analysis.

It is strongly recommended that, where possible, the gross sample should be reduced to pass a 36 B.S. sieve in one stage. This involves a minimum expenditure of time and labour, and can be carried out in mills of the hammer type which are now available. An 8-oz. sample can then be taken directly by 20 increments, evenly distributed, and used for analysis.

If, however, this method cannot be adopted, the gross sample may be ground to 36 B.S. sieve in stages, the sample being reduced in weight at each stage, as determined by the size/weight ratio.

The theory of the size/weight ratio in relation to the sampling of coal is fully dealt with in B.S. 763, which should be consulted for further information. It is very important in connection with the subdivision of coal, because the errors which occur at each stage of the subdivision and may be cumulative, can only be controlled by means of the size/weight ratio.

GRINDING AND SUBDIVISION

The subdivision or reduction in weight of a sample was formerly carried out by " coning and quartering," as previously described. It is strongly recommended that this process should be discarded in favour of the more accurate and labour-saving automatic sample dividers now available. The simplest of these is the so-called riffle, which is shown in Fig. 292 (B.S. 1017, Fig. 6).

The coal is poured evenly across a number of parallel slots, so arranged that alternate slots deliver into different containers. Provided that the width of the slots is reasonably proportional to the size of the coal, the two samples so obtained have the same ash content within small limits of error. The process can be repeated until a sufficiently small sample is obtained.

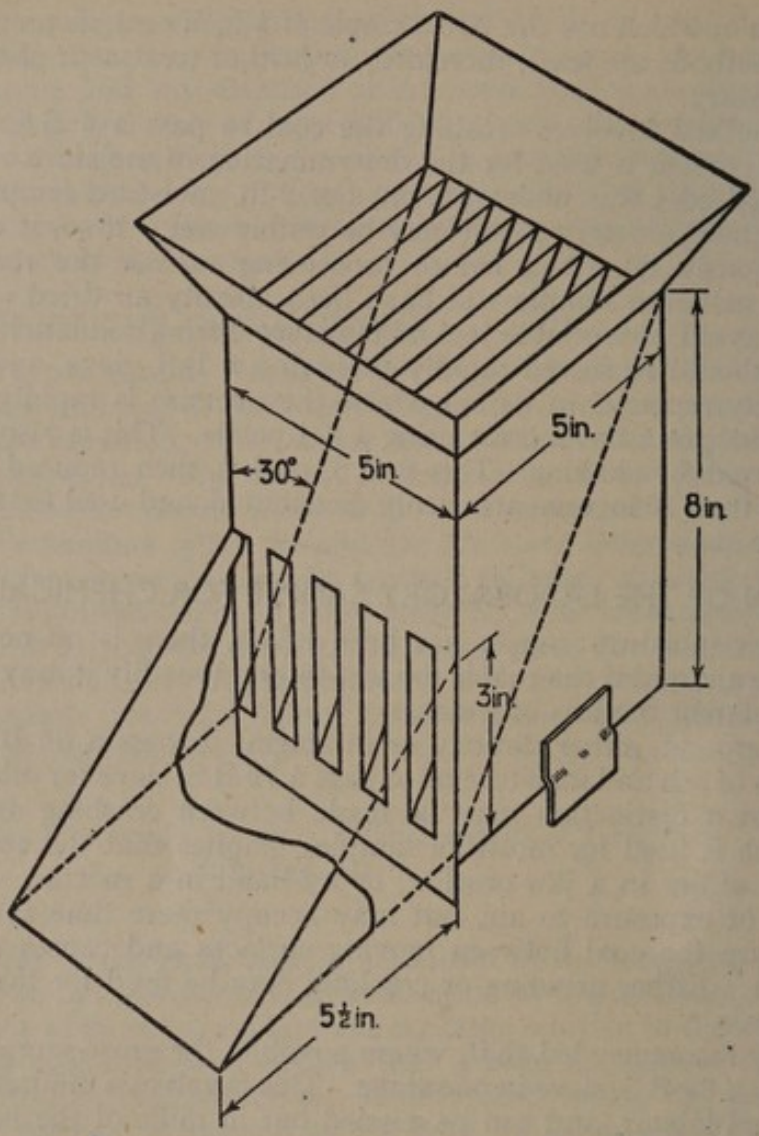

FIG. 292. Riffle sample divider (British Standard Specification 1017, Fig. 6).

Method 1. In the grinding and subdivision of a gross sample by stages it is strongly recommended that, whenever possible, the gross sample should be crushed to $\frac{1}{4}$ inch, or preferably $\frac{3}{16}$ inch before reduction in weight by means of subdivision. In order to keep the errors at each stage below $\pm$ 0·45 per cent., the gross sample, crushed either to pass $\frac{1}{4}$ inch or $\frac{3}{16}$ inch should be reduced according to Table 126.

The selected sub-sample should then be ground to pass 6 B.S. sieve, reduced

TABLE 126

Size of coal after crushing	Ash per cent.	Weight to which sample can be reduced. lb.	Maximum error 99/100
$\frac{1}{4}$ inch.	15	20	0·41
	10	10	0·39
	6	5	0·30
$\frac{3}{16}$ inch.	15	6	0·42
	10	3	0·42
	6	1½	0·31

to 2 lb. by means of sample dividers, ground to 36 B.S. sieve and used for ash determination. When other analyses are required, the 36 B.S. sample should be reduced to 2 ounces and crushed to pass a 72 B.S. test sieve.

If the coal contains less than 6 per cent. of ash, the 6 B.S. material may be reduced to 1 lb. before grinding to 36 B.S. sieve.

Method 2. Where there are no facilities for grinding the whole of the gross sample to $\frac{1}{4}$ inch or $\frac{3}{16}$ inch, it must be ground in stages and reduced at each stage by means of sample dividers. In such circumstances the following procedure may be adopted, although the errors of subdivision will exceed those of the procedure given above :—

The sample is ground to pass $\frac{1}{2}$ inch screen, either directly or by screening over $\frac{1}{2}$ inch screen, grinding the oversize and mixing back.

↓

Sample is then reduced to about 25 lb. by dividers.
(Maximum error 99 times out of 100 : with 10 per cent. ash $\pm$ 0·6,
with 15 per cent. ash $\pm$ 0·85).

↓

25 lb.-sample from above is ground to $\frac{3}{16}$ inch, either directly or by screening out oversize and crushing.

↓

25 lb.-sample is reduced to 3 lb. by divider.
(Maximum error 99 times out of 100 : with 10 per cent. ash $\pm$ 0·42,
with 15 per cent. ash $\pm$ 0·56).

↓

3 lb.-sample is ground to pass 6 B.S. sieve and reduced to 8 oz.
(Maximum error 99 times out of 100 : with 10 per cent. ash $\pm$ 0·44,
with 15 per cent. ash $\pm$ 0·60).

↓

Sample is further ground to pass 36 B.S. test sieve, either by hand or in a coffee mill.

SAMPLING WAGGONS

The ash content of industrial coal not only varies from place to place inside a waggon, but the average ash content of individual waggons also varies. In order, therefore, to obtain a true average figure for consignments delivered over a period, a minimum number of waggons must be sampled.

These are approximately as follows :—

Ash content	No. of waggons to be sampled
0–6	10
6–10	25
10–15	40
15–20	65

CORRELATION OF SAMPLING WITH THE PURPOSE OF ANALYSIS

Routine analyses are most often required to give :—

(i) The variation of individual fuels, and to provide an accurate average analysis over a period of, say, one year.

(ii) An average analysis of all coals burned at a boiler or other plant over a period of, say, a month, in order to calculate the efficiency.

For the former purpose the number of waggons given in the above table should be sampled for each fuel. Over a yearly period this is equivalent to :—

Ash content	Waggons sampled per month
0–6	1
6–10	2
10–15	3
15–20	5

For the latter purpose, if several different fuels are burned at a plant, each individual fuel need not be sampled more frequently than as given above, because the average for all the fuels used will be made up of a sufficient number of samples to give the required degree of accuracy. It must be pointed out, however, that if a fuel shows signs of abnormal ash content, further samples should be taken at once in order to obtain confirmation of deterioration or otherwise.

ACCURACY OF ANALYSIS

Just as the accuracy of a gross sample increases with the number of increments of which it is composed, so will the accuracy of an average sample increase with the number of individual tests averaged.

The average error varies inversely as the square root of the number of tests averaged. Thus it is halved where four tests are averaged, and reduced to one-third with 9 tests, one-quarter with 16, and so on. Beyond a certain point, therefore, little is gained in accuracy without a very large increase in the number of tests.

B. SAMPLING FOR SCREEN ANALYSIS

The following method of sampling for screen analysis is based on a British Standard Specification which will be issued shortly (June, 1944).

The object is to obtain from the examination of the sample, results which, within defined limits of accuracy, shall be representative of the whole of the consignment.

These limits are dependent on :—

(*a*) The error involved in the collection of the gross sample.

(*b*) The error involved in making the screen analysis.

SAMPLING

As with the sample for purposes other than screen analysis, the degree of accuracy is determined by the weight and number of increments by which the sample is taken. For screen analysis, however, these depend upon the size of the coal only and *not* upon the ash content. Table 127 should be used.

TABLE 127

Size of coal *	3 ins.	2½ ins.	2 ins.	1½ in.	1 in.	½ in.
Minimum weight of gross sample—lb. ..	500	300	160	75	50	50
Minimum number of increments	50	50	40	25	25	25
Minimum weight of increments—lb. ..	10	6	4	3	2	2

* The size of coal is defined by the upper limit and is such that not more than 5 per cent. shall remain on a square mesh of the size stated in the table.

As in all sampling, the increments should be uniformly distributed over the consignment, and, if possible, taken from a conveyor or flowing stream of coal, and never from the tops of waggons or heaps when this can be avoided.

SCREENING

When the coal is wet, small particles may cling to larger particles and so give false results. With graded coals, where the percentage of small particles ($\frac{1}{16}$ inch) is small, there is no need to dry the coal unless it is exceptionally wet. With coal containing fines, e.g. slack, the sample should be in an air-dried condition when screened.

The method of carrying out the screening may have an appreciable effect on the result as it has been shown that the movement of the coal on the screen causes breakage and so alters the particle distribution. The following method must, therefore, be followed rigidly :—

For coal less than 4 inches, British Standard screens, with square openings, should be used, and with coal over 4 inches, gauge rings or plates with round holes are employed.

The sample and fractions should be weighed so that no error greater than 0·5 per cent. is incurred, and the sample should be screened in such quantities as will allow the pieces to be in direct contact with the openings at the completion of the screenings. When the undersize material has to be re-screened, the coal should not be allowed to drop more than 6 inches.

METHOD OF CONDUCTING THE SCREEN ANALYSIS

(A) *Coal containing material larger than* $1\frac{1}{2}$ *inches.* The coal should be placed at the end of a stationary, horizontal or inclined screen, having $1\frac{1}{2}$ inch square openings, and should be moved by hand or brush across the screen until no more passes through, after which each piece should be tried by hand to see if it will pass through in any position. The pieces must on no account be forced through the openings, and doubtful pieces should be left on the screen.

The oversize should then be re-screened by the above method, starting with the largest screen or gauge ring. The undersize should be treated as described in B below.

(B) *Coal smaller than* $1\frac{1}{2}$ *inch.* (i) *Hand Screening.* The coal should be placed on screens which should be shaken as follows :—

The screen should be moved horizontally at a rate just sufficient to cause the pieces of coal to tumble or roll on the screen. The motion should not be stopped by impact and should be continued for eight movements (four in each direction). The screen should then be given two upward jerks, and the process repeated four times, making forty movements in all.

The screening should be started with the largest screen.

(ii) *Mechanical Screening.* Coal less than $1\frac{1}{2}$ inch can be screened on mechanically shaken screens. Screening should be started with the smallest size. The screen is inclined at an angle of 13 degrees and receives 190 upward movements of $\frac{1}{2}$ inch per minute. Travel of the coal over progressively larger screen openings is assisted by hand brushing.

DEGREE OF ACCURACY OF SCREEN ANALYSIS

It is somewhat difficult to specify the degree of accuracy of screen analysis because it has been shown theoretically, and proved practically, that the degree of accuracy varies with the percentage of material retained between two screens. That means that when the percentage is relatively small or large, the sampling error is relatively small, and when the percentage is 50, the possible sampling error reaches a maximum. The same theory is true of the error arising in the screening process itself. For the method given above, the errors, which should not be exceeded more than once in ten times, are as in Table 128.

It is, therefore, advisable to select screen sizes such that the percentage between two screens does not lie within the range of approximately 30–70 per cent. for

TABLE 128. ERRORS IN SCREEN ANALYSIS

On a fraction	Sampling error	Screening error	Total error
Per cent.			
2 or 98	1·0	0·5	1·1
5 ,, 95	1·5	1·0	1·8
10 ,, 90	2·0	1·0	2·25
20 ,, 80	2·0	2·0	2·8
25 ,, 75	2·5	2·5	3·5
30 ,, 70	3·0	2·5	3·9
40 ,, 60	4·0	2·75	4·9
50	5·0	3·0	5·8

coal under 4 inches. Above this size, rings or plates should be used with openings of 12, 8, 6, 5 or 4 inches diameter.

When frequent routine screen analyses are being made on any particular coal, the same degree of accuracy of single determinations may not be necessary, and some departure from the specified number of increments may be permissible. It is then possible to combine sampling for screen and other analyses by putting regularly spaced increments into each of two containers so as to obtain two samples, one for each purpose. If a larger screen analysis sample is required, two increments may be taken for every one taken for the other sample.

The screen sizes may also be standardised, the most useful being 2, $1\frac{1}{2}$, 1, $\frac{3}{4}$, $\frac{1}{2}$, $\frac{1}{4}$ and $\frac{1}{8}$ inches.

When testing coal over 4 inches, the somewhat laborious method given above may be modified so as to use movable screens which should be moved horizontally at a rate just sufficient to cause the pieces of coal to tumble or roll on the screen. The motion should not be stopped by impact and should be continued for ten movements (five each way). The coal should then be examined to see if any pieces remaining on the screen will pass through by hand.

C. THE SAMPLING OF COKE

The principles involved in the sampling of coke are exactly the same as for coal. The weight of the gross sample necessary for any degree of accuracy is independent of the quantity sampled and is determined by the number of increments by which it is taken. This number increases with the variability or heterogeneity of the coke sampled. The variability of coals is measured by the average error of the ash content, the moisture content being relatively constant. In coke, however, the ash is the more constant factor, whilst the moisture content, which depends to a very large extent upon the quenching procedure at the point of production, and also, to a lesser degree, upon the subsequent exposure to the atmosphere, varies over much wider limits, not only in bulk but from lump to lump. The variability of coke is, therefore, measured by the average error of the moisture content, for the determination of which it has been shown experimentally that a minimum increment of $2\frac{1}{2}$ lb. must be taken.

Coke emerges from the carbonisation process red hot, and is generally quenched with water either outside or inside the retort. A large piece of coke will be wetted on the outside, but will retain sufficient heat inside to evaporate the surface water. Unless quenching has been injudicious, large pieces should be dry, but if wet may be expected to have more water on the surface than inside. Small pieces, on the other hand, which are necessarily treated with the same weight of quenching water per unit weight of coke, may be cooled completely. Consequently the smaller the size of the coke derived directly

from the manufacturing process, the wetter it will tend to be. This rule will not necessarily hold when the smaller sizes of coke are derived from coke breakers.

With coal the average error increases with ash content within the limits usually encountered, but with coke it has been found that, using 2½ lb. increments, the average error increases with moisture content up to 8 or 9 per cent., above which it remains constant up to 15 per cent. moisture. As a consequence the number of increments necessary for any required degree of accuracy also increases with moisture up to 8 or 9 per cent.

SIZE OF SAMPLE AND NUMBER OF INCREMENTS

For the purposes of sampling, the B.S. Specification No. 1017 divides coke into two classes :—

Class 1

(*a*) Screened or unscreened coke containing not more than 7½ per cent. below 1 inch when quenched, and not more than 5 per cent. moisture.

(*b*) Graded coke (e.g. coke nuts).

Class 2. Other cokes, breeze.

The moisture in Class 2 is more variable than in Class 1, and consequently more increments and larger gross samples are required, as shown in Table 129, which should be used for an accuracy within ± 1 unit of the true moisture content in 99 cases out of 100. The minimum weight of increment is 2½ lb.

TABLE 129. MINIMUM WEIGHTS OF GROSS SAMPLE

Moisture content	Class 1		Class 2	
	No. of increments	Minimum weight of gross sample lb.	No. of increments	Minimum weight of gross sample lb.
Up to 2	20	50	35	88
Over 2 and up to 3	44	110	70	175
„ 3 „ „ „ 4	74	185	120	300
„ 4 „ „ „ 5	95	238	170	425
„ 5 „ „ „ 6	112	280*	235	588
„ 6 „ „ „ 7	118	295*	280	700
„ 7 „ „ „ 8	126	315*	300	750
„ 8	134	335*	305	763

* Graded coke only.

If the size when quenched or the approximate moisture content is not known, the coke should be considered as falling into Class 2 unless a lower degree of accuracy is acceptable.

B.S. 1017 also gives data for sampling with higher degrees of accuracy such as may be required for special performance or efficiency tests. Much larger samples, however, are required, and reference should be made to the specification for further details.

THE COLLECTION OF THE GROSS SAMPLE

Because coke readily takes up or loses moisture according to atmospheric conditions, waggons should never be sampled from the top except for coke freshly loaded at the producer's works, when it is known that the loaded waggons have not been subjected to rain, snow, or prolonged air-drying. Otherwise, as with coal sampling, the necessary increments should be spread over the consignment to be sampled by taking uniform increments at regular intervals from chutes or conveyors whilst the coke is in motion, or from a

falling stream of the coke. The ladle or shovel should be regularly plunged into different parts of the stream so as to avoid the effects of segregation.

A single lump of coke weighing 2½ lb. or over, may be regarded as an increment, but every care should be taken to obtain the correct proportions of small and large coke. Every precaution should also be taken to avoid free ventilation of the coke until the moisture sample has been taken.

THE REDUCTION OF COKE

Owing to its abrasive action coke should never be ground in steel mills. This practice causes contamination with iron, and increases the ash content appreciably. A manganese steel, agate, or similar mortar may be used for grinding in the final stage of reduction, but otherwise the whole process must be carried out by crushing or impact. A mechanically driven jaw crusher may be used, or, alternatively, the coke may be crushed by hand using a manganese steel plate and punner or maul.

PRELIMINARY REDUCTION AND COLLECTION OF MOISTURE SAMPLE

Where a mechanically driven jaw crusher is available, it is recommended that the whole of the gross sample should be crushed to pass ½-inch B.S. sieve. The crushed sample should then be mixed quickly on the floor, with a shovel, avoiding undue ventilation of the coke, scraped into a heap and flattened. If the gross sample much exceeds 50 lb. in weight, it should be reduced to that amount by taking fifty increments of approximately 1 lb. each, such increments being spread uniformly over the whole mass.

This 50 lb. should then be quickly mixed again, scraped into a heap and flattened, and a sample of not less than 2 lb. (4 lb. when sampling breeze) taken by not less than twenty equal increments, uniformly distributed. This sample should be placed at once in an air-tight container and sent to the laboratory for moisture determination.

Where hand crushing has to be adopted, the gross sample should be rapidly broken to approximately 1 inch, and reduced to 50 lb. by means of fifty increments, uniformly distributed. This 50 lb. should then be crushed rapidly to ¼ inch and, as before, 2 lb. taken by twenty increments, placed in an air-tight tin, and sent to the laboratory.

The gross moisture sample must not be reduced below ½ inch or the loss of moisture incurred will be appreciable.

REDUCTION OF SAMPLE FOR GENERAL ANALYSIS

The ½-inch sample (2 lb.) taken as just described will be dried in the laboratory for moisture determination without further reduction and may then be used for preparation of the general sample. Exposure no longer matters.

It should first be crushed on a manganese steel plate, using a manganese steel maul, to pass ⅛ inch, scraped into a heap, and flattened.

A sample of not less than 4 ounces should then be taken by not less than ten equal increments, uniformly distributed over the heap and then crushed or ground to pass 72 B.S. test sieve, using a manganese steel, agate, or wedgwood mortar. The final sample should then be bottled for further chemical analysis.

SAMPLING FOR THE SHATTER TEST

The sample should be taken from the waggons leaving the maker's works or on arrival at the user's works.

The gross weight of the sample should be 250 lb. collected by increments distributed evenly over the whole number of waggons in the consignment.

Each increment should consist of a single piece of coke, over 2 inches in size, and the pieces taken should be representative of the sizes of the lumps in the consignment.

CHAPTER XXXII

THE SAMPLING OF GASES

The problems and precautions in sampling gases—Selection of position for sampling—Suspended matter—Temperature effects and their control—Other factors vitiating the true composition of the sample ; effect of materials and air leaks—Rate of sampling—Transport—Contamination of samples ; solutions for use in sampling.

THE accurate sampling of gases requires considerable care both in the choice of the position for sampling and in the method of sampling adopted. Frequently the problem of collecting a true sample presents greater difficulty than the analysis of the sample. To obtain a true sample two conditions must be satisfied :—

(1) The sample must be representative of the gas flowing at the time of sampling.
(2) No change in composition either by chemical action or solution must take place during the taking or storage of the sample.

Of these the first is by far the more difficult to satisfy and a considerable proportion of this chapter is therefore devoted to the problem of how to obtain a representative gas sample under the conditions commonly found in practice. The importance of obtaining a truly representative sample cannot be stressed too strongly, as analytical data obtained from an incorrect sample are useless for plant control or for any other purpose.

From the many gas sampling apparatus which have been devised, a selection has been made of the types which may be purchased or which can be constructed fairly easily in a laboratory. The general principles of gas sampling have been given rather than detailed instructions of the methods by which the sampling may be carried out. For the reader who requires more information a bibliography is appended to facilitate reference to the original papers.

GASES CONSIDERED

The principles set out below apply to fuel gases such as coal gas, water gas and producer gas, and to the waste gases from the combustion of any fuel.

Certain special problems have been omitted, such as the sampling of petroleum hydrocarbons at low pressures (1),* or gases from high pressure apparatus (2), or high pressure natural gas (3).

PROBLEMS IN GAS SAMPLING

The rate of flow and pressure of the gas to be sampled may vary with time in an unknown manner. The chemical composition and physical properties may vary not only with time, but from one point to another in the same main. The gas may carry suspended matter such as fly ash or soot, tar fog or other liquid particles, the deposition of which would block the gas passages of the sampling apparatus. Any vapours present will condense if the gas is cooled below its dew point ; the liquid precipitated may dissolve some of the gaseous constituents and also corrode the apparatus.

PRECAUTIONS

The requirements necessary to obtain a representative sample fall into three classes :—

(1) Care and experience are needed in choosing the correct position from which to withdraw the sample.

* The figures refer to the bibliography at the end of the chapter.

(2) The rate of collection of the sample should be strictly regulated to maintain correct proportionality between the rate of sampling and the flow of gas.
(3) No change should occur in the composition of the sample between the time when it is collected and the time when it is analysed.

SELECTION OF SAMPLING POSITION

Dead Spaces. The sampling point must be so chosen that the main gas stream is flowing past the end of the sampling tube. Branch lines which contain stagnant gas are obviously unsuitable. Similarly the volume of the sample line must be kept small and the velocity of the gas in it must be great enough to ensure that the lag is reduced to the minimum.

Eddies. Immediately following obstructions, changes of direction and changes of cross-section, eddies form in which the gas, although it may be in violent motion, is circulating in vortices without interchange with the main stream. In order to avoid such pockets, it is recommended that a sampling tube should not be inserted in a main nearer than 15 pipe diameters away from such disturbing features.

Mixing. Where the gas stream to be sampled is made up of two or more component streams it is obviously necessary to ensure that any sample repre-

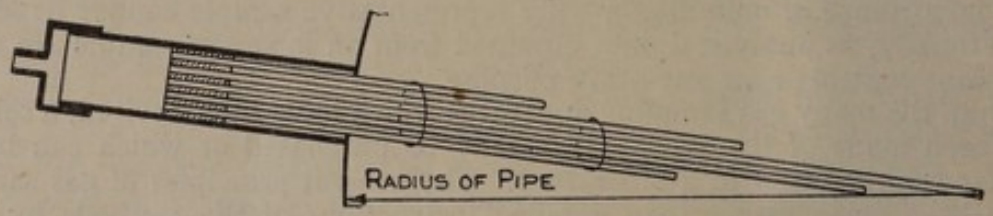

FIG. 293. Arrangement for sampling over the cross-section of a gas stream.

sents all the component streams in their true proportions. It is surprisingly difficult to mix two streams of gases thoroughly, and the difficulty is increased when it is necessary to mix gases of widely differing density such as the hydrogen and carbon dioxide in producer gas and water gas.

Sampling from the Cross-section of a Main. If thorough mixing cannot be assured, it may be necessary to divide a cross-section of the pipe into a number of imaginary areas and to take samples simultaneously from the centres of these areas (4).

There are many possible ways of automatically carrying out this principle. One simple arrangement, consisting of a bundle of tubes of different lengths, is shown in Fig. 293.

A perforated tube may be inserted along a diameter of the pipe-line, the perforations being dimensioned in the proportions determined by a pressure exploration of the cross-section of the pipe so that each hole contributes its correct share to the sample.

An arrangement of this type which is suitable for permanent sampling positions in flues makes use of two tubes at right angles perforated along their length and placed across the full span of the flue. The holes should not face up or down stream, but across the direction of flow, and their diameters should be graduated, the largest being at the middle of the span. Sizes M30 and M40 (approximately $\frac{1}{8}$ and $\frac{1}{10}$ inch diameter) are usually adequate. The sample should be withdrawn from both ends of both tubes by a system of interconnections, as shown in Fig. 294.

The sampling device should not be so big that it impedes the flow of gas.

The presence of suspended matter may cause blockages and provision must be made for frequent inspection and clearing.

It will often be sufficient, when there are no disturbing factors, to draw the sample from the position of average velocity of the stream. To do this the sampling tube should be inserted to a depth of 0·3 of the radius of the main or duct, or to an equivalent depth in a duct of rectangular cross-section.

Sampling the components of a mixture. If thorough mixing cannot be assured or if the sample cannot be made to represent each element of the cross-section, the only remaining possibility is to sample the components separately and calculate the analysis of the final mixture from the analyses and proportions of the components.

Final Selection of Sampling Position. The soundest practice in choosing a sampling position is first to explore all possible sampling positions thoroughly

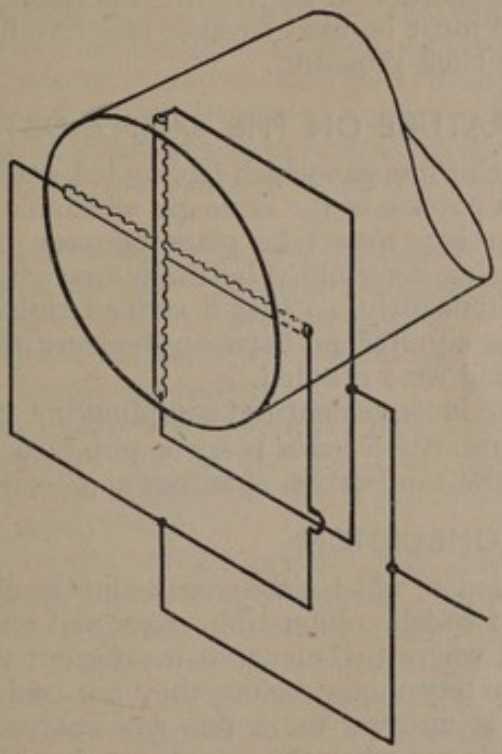

FIG. 294. Permanent arrangement for sampling across a gas stream.

in accordance with these principles, and then to choose the most suitable position from which a true sample of the gas will be obtained.

With a new plant, or when beginning new or special work on an existing plant, it is usual to sample at a number of points at first and subsequently to discontinue sampling at some of these positions, as experience indicates. Although it is laborious, a number of samples should be taken at various distances across a pipe or duct at the position chosen for sampling and the analyses compared with a composite sample taken by means of a perforated tube, or bundle of tubes, as described below. This checking is a very necessary precaution.

Recording Instruments. In addition to the considerations and precautions outlined in the section dealing with the selection of the sampling point, the special requirements of the instrument must be taken into account when choosing the site for a recorder. Draughts, high temperatures, or wide variations of temperature are always to be avoided. It may be desirable to place the recorder or indicator in a position where it can be read easily by the operator of the plant, and this may necessitate enclosing it in a dust-proof case.

SUSPENDED MATTER

Where the gas to be sampled carries dust or liquid droplets in suspension, steps must be taken to free it from the suspended matter at an early stage in the sampling line.

Filters for Dust. Dust may be removed by means of a chamber loosely packed with slag wool or asbestos, or granular material such as coarse sand or crushed and graded pumice, or by a porous partition of carborundum or sintered glass. The dust filter should be placed where the temperature never falls below the dew point for the gas, as condensation of moisture in any filter causes serious resistance to the gas flow.

Tar Fog Separator. An apparatus for the removal of suspended matter from a stream of gas has been described (5) in which a jet of the gas at high velocity is directed against an obstructing surface. The sudden change of velocity and direction causes the deposition of matter in suspension.

Some type of pump must be used because effective filtering must always be accompanied by high back pressures.

EFFECTS OF TEMPERATURE ON THE SAMPLE OBTAINED

In general, a sample of flue gas which has cooled gradually will not have the same composition as a portion of the same gas which has been suddenly chilled. Certain reversible reactions may take place between the constituents of flue gases, the equilibria being determined by the temperature. These reactions do not take place instantaneously, so that a sudden chilling of the flue gas will prevent the shift in the equilibrium between the constituent gases which would take place if the cooling were gradual.

Thus, in spite of the inconvenience of sampling hot gases, it is essential if a true sample of the combustion gases is to be obtained to sample the gases as near to the outlet of the combustion chamber as possible.

COMPLETION OF COMBUSTION

A different phenomenon which produces a similar effect is met with when sampling flue gases in which combustible gases and excess oxygen both exist at the same time and where turbulence is insufficient to bring them together. If the gases are intimately mixed before they are cooled further burning will take place. It is thus unusual for a flue gas analysis to show both excess oxygen and combustible gas when the sample is taken through a hot tube.

WATER COOLED SAMPLING TUBE

The change in the equilibrium between the constituents of a flue gas and the completion of combustion can be prevented by using a water cooled sampling

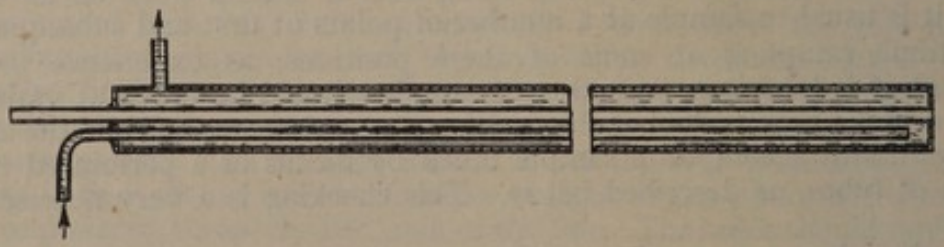

FIG. 295. Water-cooled sampling tube.

tube, Fig. 295. An additional desirable result of using such a tube is that the gas sampling apparatus is protected against damage by heat.

One form of water cooled sampling tube is described by L. H. F. Nichols (6). The gas tube was only $\frac{1}{16}$ inch internal diameter in order to secure high gas velocities and thus increase the rate of heat transfer. Arrangements were made

to clear blockages, when necessary, by means of a steel wire. The gas was aspirated by means of an exhauster which would deliver 3 cubic feet per minute and which would produce a vacuum of 600 millimetres mercury. The sample was passed through a filter and a flow gauge and the rate of flow was regulated by a by-pass on the exhauster. The apparatus was tested for leaks under a vacuum of 24 inches of mercury which it held for half an hour.

With this apparatus Nichols was able to withdraw samples from very near the fuel surface of a producer bed which contained about equal quantities of carbon dioxide, carbon monoxide and oxygen.

INTERACTION BETWEEN THE SAMPLE AND THE MATERIALS OF THE APPARATUS

Most fuel gases and flue gases contain constituents which react to some extent with the materials normally used in the construction of sampling apparatus, but the amount of the action does not usually invalidate the sample.

The permeability of rubber to carbon dioxide is easily demonstrated, but it is doubtful whether a reasonable length of rubber tubing would reduce the carbon dioxide content of a flue gas sample flowing through it by an amount sufficient to be detected by the methods of analysis used for furnace control purposes.

Sulphur compounds all probably react with rubber to some extent ; those likely to be encountered include hydrogen sulphide, carbon disulphide and oxysulphide, and the oxides and oxy-acids of sulphur. Oxygen and the oxides of nitrogen are probably not without action on rubber. Hydrocarbon gases and liquids such as tar fog dissolve in rubber.

Glass, fused silica and glazed porcelain are not acted upon by any of the constituents of fuel gases or flue gases. Silica at high temperatures is permeable to hydrogen and other gases, but the practice is now to sample hot flue gases through a water cooled tube.

The metals likely to be used in the construction of gas sampling apparatus are iron, possibly oxidised, copper and brass and " compo " tubing.

All these metals are acted upon by hydrogen sulphide, by oxides of sulphur and oxides of nitrogen. Under suitable temperature conditions iron and its oxide scale will react with oxides of carbon, hydrogen, oxygen and water vapour. Copper and brass react with ammonia, hydrocyanic acid and cyanogen. Aluminium or tin tubing is generally used where hydrogen sulphide, being present, must not be removed before analysis.

Protection of Rubber Connections. Rubber connections are very widely used in gas sampling. They should be kept as short as possible and should be protected where necessary. Coating the exposed rubber with cellulose lacquer or anti-sulphuric paint should be sufficient to prevent contact with the gas, if a continuous coating can be assured.

An alternative which more positively avoids the exposure of the rubber to

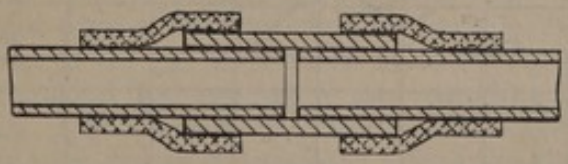

FIG. 296. Glass sleeve to avoid contact of rubber connections with gas.

the gas is shown in Fig. 296. A glass sleeve is fitted over the joint, short lengths of rubber tubing being used to seal the ends of the sleeve. It is particularly important to prevent hot gases from coming into contact with the rubber connections, and where high temperature precludes the use of rubber the

sealing of the sleeve shown in Fig. 296 can be carried out by means of a quick-setting cement, such as zinc oxychloride or one of the dental cements.

Zinc oxychloride is readily made by taking a small quantity of zinc chloride which has deliquesced and making it into a paste with zinc oxide. It sets rapidly and therefore should be made up only in small quantities as required. It has the advantage that it can easily be dissolved away, when needed, by a little concentrated hydrochloric acid.

AIR LEAKS

Air may leak into flue gases in several ways. Flues are usually working below atmospheric pressure and air leaks in continuously through badly-fitting sight holes and doors, and through cracks in the brickwork of the setting. The amount of dilution which waste gases suffer during passage from the combustion chamber to the base of the stack may be considerable and flue gases should therefore be sampled as near to the combustion chamber as possible, in order that a true indication may be obtained of the efficiency of combustion. The amount of air leaking into the setting is normally determined from the analyses of samples of flue gas taken from different points in the flue system.

The joint where the sampling tube enters the setting must be tight to prevent inleakage of air at this position.

Again, as the flue is below atmospheric pressure it is necessary to aspirate the sample with a suction ball or bellows or by some other means. The sampling line must therefore be free from leaks.

Leakage of air into samples being taken from mains under positive pressures can only occur through negligence.

RATE OF SAMPLING

If the composition of a stream of gas is uniform, any snap sample taken at any time is a representative sample.

If the composition of the gas varies but the rate of flow is constant, a series of snap samples taken at equal intervals will give a truly representative sample, provided that the number of snap samples is sufficiently large in relation to the rate at which the composition of the gas is changing. For small variations, one sample per hour may be sufficient, while in some conditions one sample every thirty seconds may be required. Any apparatus, therefore, which takes the sample at a constant rate will ensure the collection of a representative sample of the gas, provided that the rate of flow is constant.

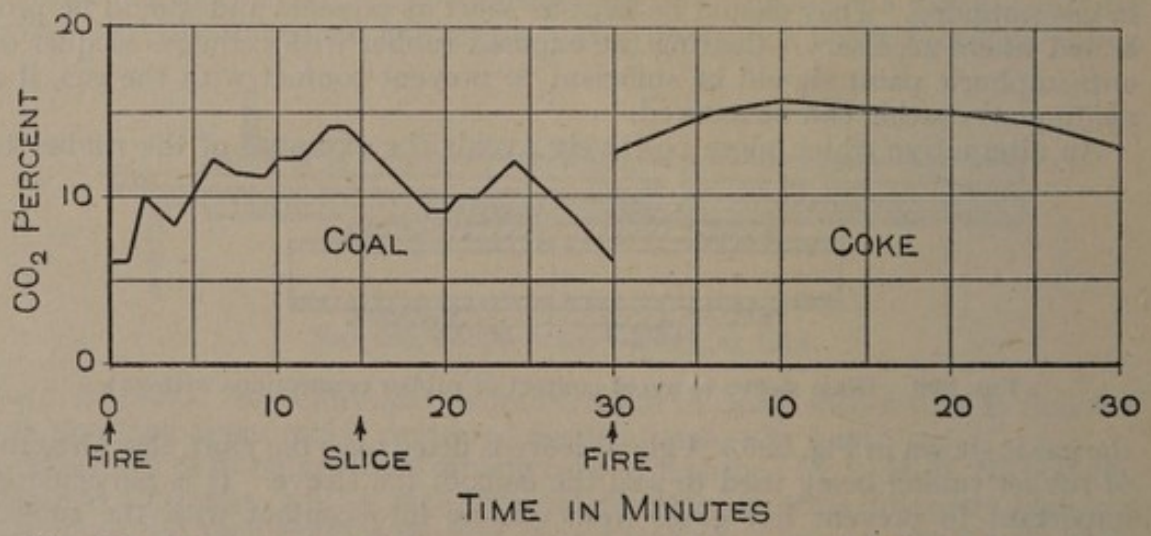

FIG. 297. Variations of CO_2 content of flue gases from a hand-fired marine boiler.

In general, however, where the composition, rate of flow, and pressure of the gas are all varying with time, special methods must be adopted to ensure that the rate of collecting the sample is a constant fraction of the rate of flow of the gas stream.

Example of Variation of Flue Gas Composition. An example of the manner in which the carbon dioxide content of flue gases can vary in practice is shown in Fig. 297 for a hand fired Scotch Marine boiler, operating with natural draught. Snap samples taken at frequent intervals were analysed for CO_2 and the results show a cyclic variation over the thirty-minute periods from one firing to the next. These results are for good practice ; even more variable CO_2 contents are met with in normal operation with boilers of this type.

Obviously, to obtain a representative sample of the flue gases under such conditions, especially when burning coal, would entail taking an average sample over many cycles or a very large number of snap samples.

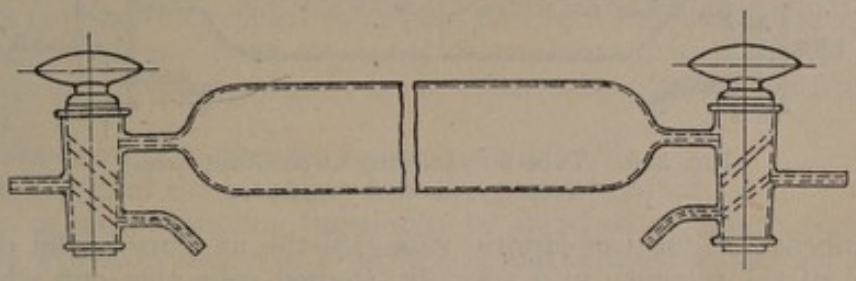

FIG. 298. Gas collection tube.

Snap Samples. When the gas analysis apparatus is portable, snap samples are usually taken directly into the measuring burette of the apparatus.

If, however, the analysis cannot be done at the plant, or if the gas analysis apparatus is not portable, snap samples may be taken in a plain tube, Fig. 298, by allowing the confining liquid to run out rapidly.

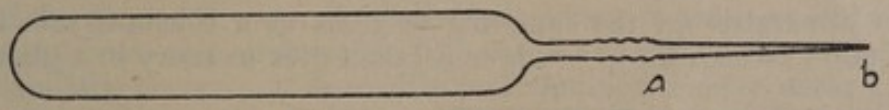

FIG. 299. Sealed gas collection tube.

It is sometimes convenient, as for instance, in taking snap samples of vitiated air from inaccessible places, to use a tube which has been evacuated and sealed off, Fig. 299. The sealed tip of the tube may be broken off by remote control ; one method is to use a " break back " mousetrap (7).

Other means of taking snap samples are the Gooderham holder (8) and forms of apparatus such as that described by L. Silverman (9).

Samples are frequently taken by allowing the gas to blow through a plain sample tube until it is thought to have been purged and then to close the cocks. This practice does not give a very satisfactory sample, but may be tolerated where the composition of the gas is constant or changing only slowly.

A metal plunger pump for use as a gas sampling apparatus has been described (10), which, it is claimed, has been used successfully in analyses of air samples to a very high degree of precision. Samples stored in it for twenty-four hours have shown no change.

The piston is made gas-tight by a leather ring gripped between two plates which can be pressed together by turning a long screw passing down the tubular piston rod, thus forcing the leather ring radially outwards into gas-tight contact with the wall of the cylinder.

Sampling at Constant Rate. If the gas sample is aspirated into a simple sample tube, Fig. 298, by allowing the mercury (or other liquid) to drip from the lower cock, the rate at which the sample collects is proportional to the square root of the head of mercury above the regulating cock. The rate of flow of the sample diminishes progressively and an error of 50 per cent. may be introduced in the rate from the start to the finish of sampling.

With a Huntly tube (11), Fig. 300, it has been stated (12) that the rate of efflux of the mercury is uniform and the volume of the sample increases uniformly, but since the gas is under a reduced pressure which varies with time, the rate of collection of the gas increases towards the end, and an error of 50 per cent. may be introduced in the rate.

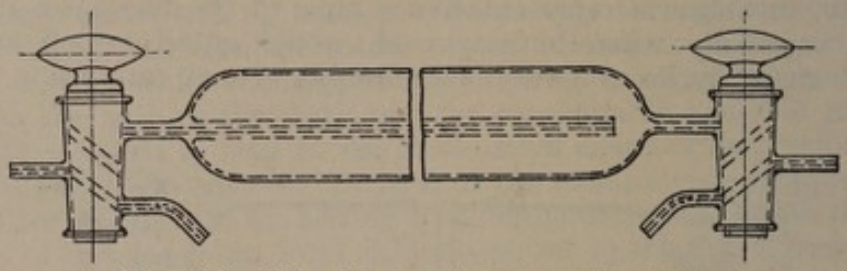

FIG. 300. Tube for sampling at constant rate.
(G. N. Huntley, *J. Soc. Chem. Ind.*, **29**, 312.)

In a compensated form of Huntly tube (13) the mercury outlet dips below the surface of the mercury in a specially shaped collecting cup which is the frustum of a cone with slightly curved sides, flared towards the top. A constancy of rate within 5 per cent. is obtained.

It is usually considered that tubes of the Huntly type should only be used with mercury because the gas, as it collects, is scrubbed by the confining liquid. An apparatus has been described (14) in which the inlet tube is enclosed in another tube and the gas bubbles through only a small fraction of the confining liquid. An oil film keeps the main bulk of the liquid out of contact with the gas.

Another apparatus for the sampling of gases at a constant rate has been described (15) in which the gas sample is taken over mercury in a plain sample

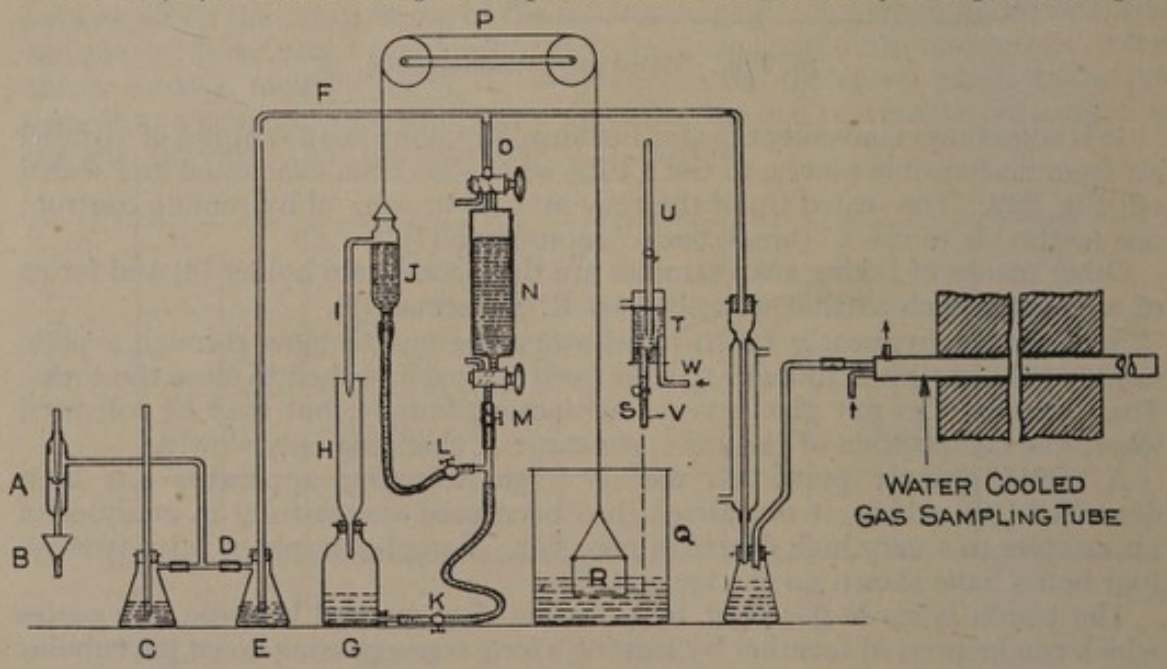

FIG. 301. Apparatus for sampling at constant rate.
(T. Gray, *J. Soc. Chem. Ind.*, **32**, 1093.)

tube. The mercury displaced passes through an orifice (the diameter of which determines the rate of sampling) to a reservoir suspended on a spring, the characteristics of which are chosen so that the increasing weight of the displaced mercury extends the spring by the correct amount to maintain a constant head.

An apparatus devised by T. Gray (16) is shown in Fig. 301. Gas is drawn through the apparatus by the filter pump A. The sample is collected in the sample tube N and the confining liquid flows through the counterpoised reservoir J. This reservoir is lowered at a constant rate which can be varied according to the rate at which water from the constant head vessel T flows through the orifice V into the tank Q, thus raising the float counterpoise R. The connection O is preferably made of capillary tubing from 3–4 inches long to prevent diffusion of gas backwards from N into F.

The period of sampling may be varied in a number of different ways :—

(1) Jets may be constructed to deliver the requisite quantity of water in the specified times ; these are easily drawn from glass tubing in a blowpipe flame, the final adjustment being effected by altering the head of water in T.
(2) Tanks of various diameters may be employed.
(3) The height of the sampling tube, N, may be varied.
(4) Two or more sampling tubes may be attached at different points to the tube, F, and by means of T-tubes, to J and G, these being used in sequence.

A portable apparatus has been devised by S. Pexton and W. K. Hutchison (17) for collecting a gas sample over mercury at a rate which is sensibly uniform. Mercury, in flowing from the sampling tube, displaces water through an Edwards regulator in which the water is driven through a capillary and through a mercury seal. Any change in pressure in the displacement vessel (due, for instance, to the decreasing head of mercury in the sampling tube), brings about a corresponding change in the depth of the mercury seal in the regulator. A constant differential pressure is thus maintained across the capillary and hence the rate of flow of the displaced water through the capillary is constant and the volume of the sample increases uniformly. The rate of collecting the sample depends on the capillary chosen ; it is readily interchanged and by using a length of thermometer tubing a rate as low as 200 cu. cm. in forty-eight hours has been attained.

An apparatus of similar type has been developed by J. R. C. Duke (18), and is shown in Fig. 302. The gas sample is taken into an interchangeable, standard gas sampling tube, A. The mercury displaced by the gas in its turn displaces water from the vessel, B, to the regulator, C, where it flows through the capillary tube, D, and through the partition, E, which consists of a Jena sintered glass diaphragm of porosity G4, which is impervious to mercury at pressures below about one atmosphere. The displaced water enters at a pressure sufficient to keep the mercury pressed against the underside of E. The rate of flow of the water through the capillary (and consequently the rate of collection of the gas sample) is determined by the height, h, of the jet above the outer mercury surface in the regulator ; this is constant during the taking of a sample. The sampling rate can be changed by adjusting the quantity of mercury in the regulator. A sampling rate as low as 150 cu. cm. in a day has been attained. The overflow tube, F, must not be submerged or the back pressure may vary and cause unsatisfactory regulation. The bent-over discharge tube of D should nearly touch the surrounding tube, so that the water, instead of issuing as detached drops, passes in a continuous stream between the mercury and the glass. All connections must be as rigid as possible and rubber joints must be of strong tubing bound with tape. Rubber stoppers cannot be used in the

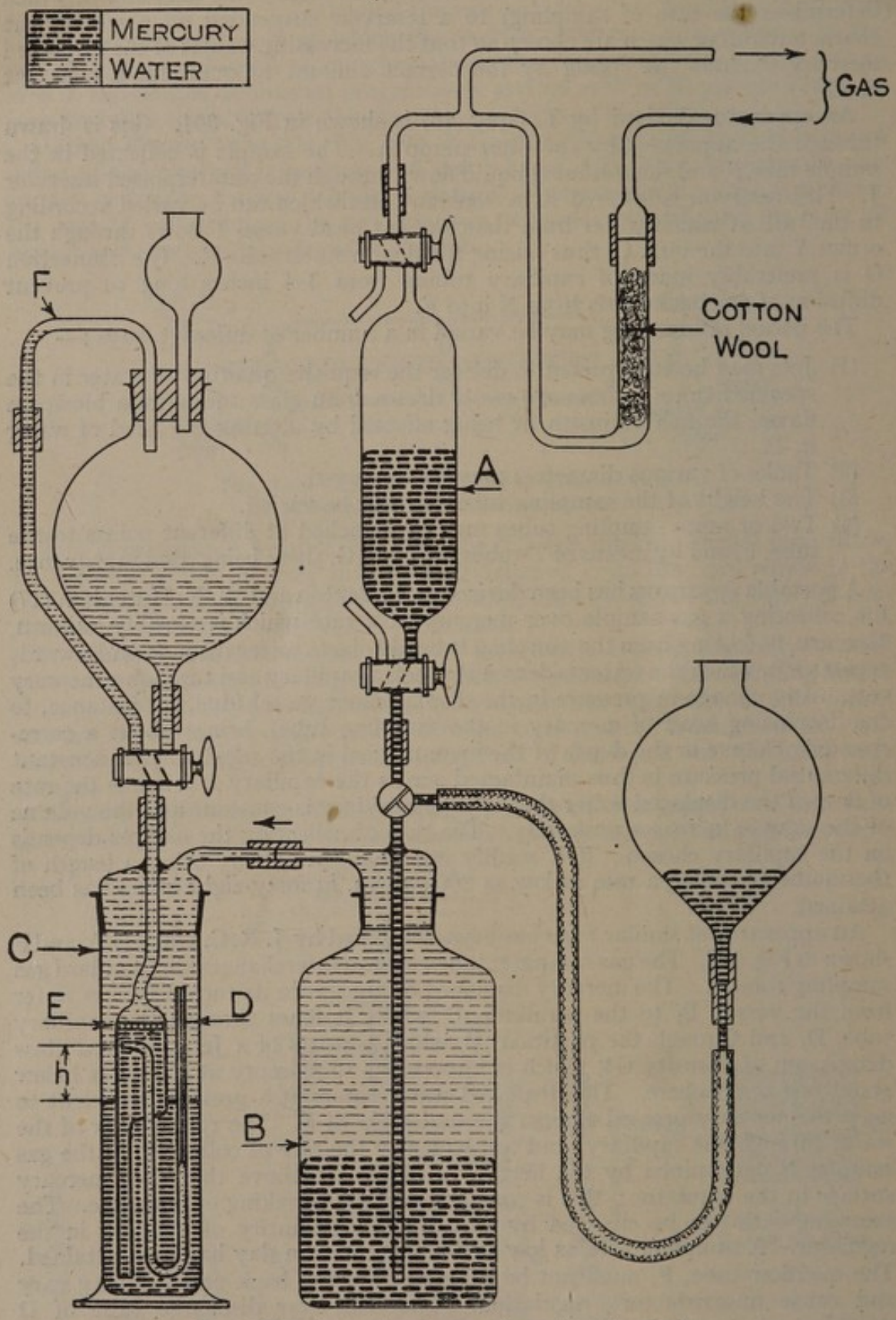

FIG. 302. Apparatus for sampling at constant rate.
(J. R. C. Duke, *J. Soc. Chem. Ind.*, **58**, 231T.)

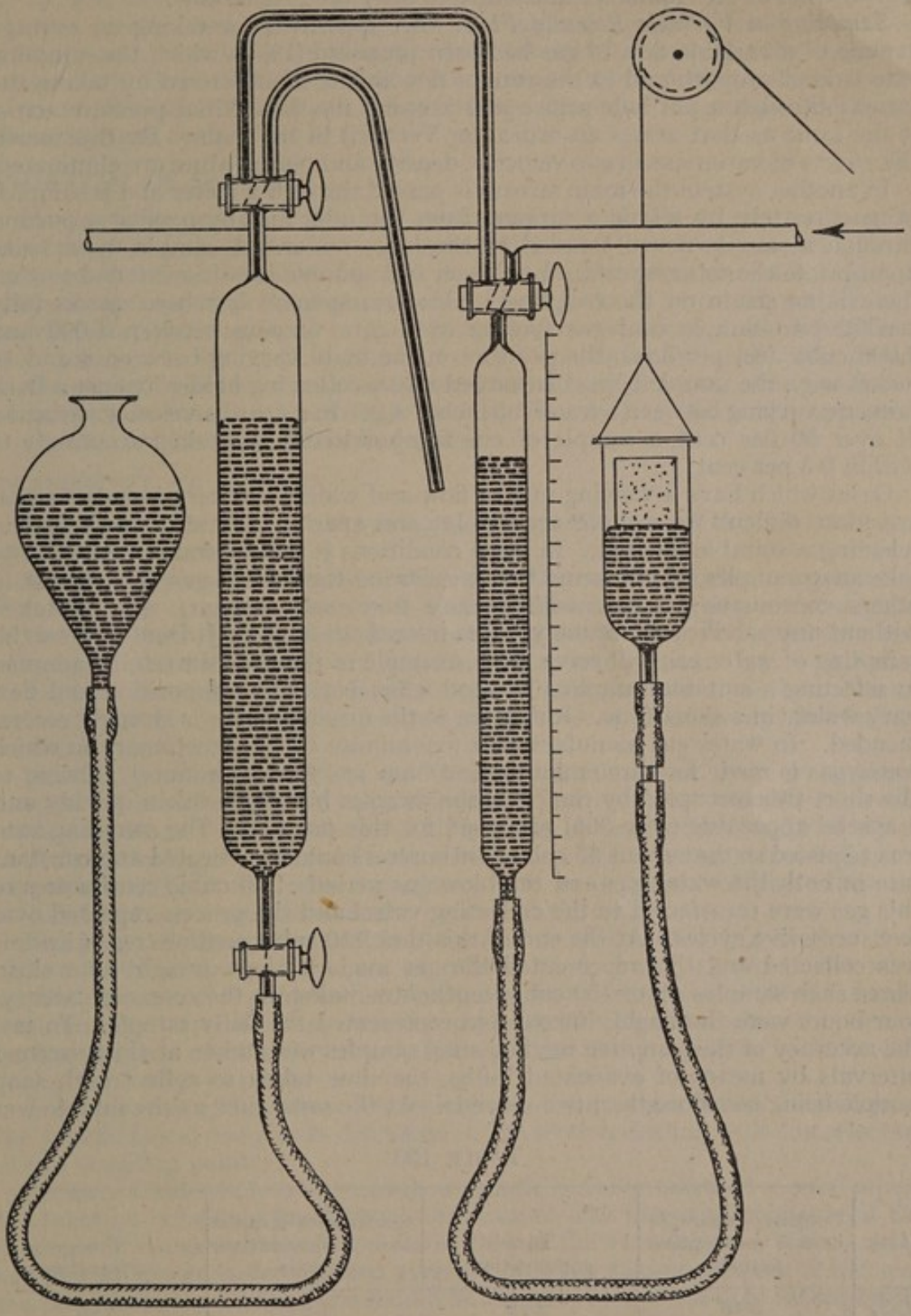

FIG. 303. Apparatus for sampling variable rates of flow.
(F. J. Dent, *Trans. Inst. Gas Eng.*, 82, 212.)

regulator as they are easily deformed by pressure to an extent which affects the position of the capillary sufficiently to alter the rate of flow.

Sampling at Variable Rates of Flow. An apparatus for taking an average sample of a variable flow of gas has been proposed (19) in which the sampling rate is kept proportional to the rate of flow in the main stream by taking the sample through a pin hole orifice and keeping the differential pressure across it the same as that across an orifice (or Venturi) in the main. By this means the effects of variations in gas velocity, density and temperature are eliminated.

In another system the main stream is passed through a meter and is sampled proportionately by taking a tapping from the inlet and by-passing a portion through a sample meter coupled to the large meter. A sample meter with appropriate characteristics can be chosen and suitable conditions found so that there is no strain on the coupling. This arrangement has been successfully used (20) to sample coal gas flowing at a rate varying between 4,000 and 7,000 cubic feet per hour, the pressure in the main varying between 6 and 10 inches w.g., the sample being discharged into a collecting holder against a back pressure varying between 2·5 and 4·5 inches w.g. In spite of a velocity variation of over 50 per cent. a sample of one five-hundredth was taken accurately to within 0·3 per cent.

Gases which have a varying rate of flow and which also vary in composition are most difficult to sample accurately, and special care must be taken in selecting a suitable method. In some conditions it will be found necessary to take snap samples, at the same time measuring the rate of gas flow, whilst in others continuous samples without any flow measurements can be taken without any sacrifice of accuracy. The investigation of F. J. Dent (21) on the sampling of water gas will serve as an example of the procedure to be adopted in selecting a suitable sampling method when both gas composition and flow vary widely in a short time. Reference to the original paper is strongly recommended. In water gas manufacture a five-minute cycle is customary in which water gas is made for three minutes and blow gas for two minutes. Owing to the short time occupied by the cycle the samples had to be taken quickly and a special apparatus (Fig. 303) was used for this purpose. The sampling rate was adjusted so that about 35 cubic centimetres could be collected at a constant rate in both the water gas and the blow gas periods. 30 cubic centimetres of this gas were transferred to the collecting vessel and the process repeated over five successive cycles. At the end of this time 150 cubic centimetres of sample was collected and this represented the gas made during an eight-hour shift. Three such samples all of 150 cubic centimetres taken in the course of twenty-four hours were thoroughly mixed and represented the daily sample. To test the accuracy of the sampling method snap samples were taken at thirty-second intervals by means of evacuated bulbs, the time taken to collect each snap sample being not more than two seconds. At the same time as the sample was

TABLE 130

Gas	Continuous sample over 5 consecutive periods	True composition	Continuous sample over 5 consecutive periods	True composition
CO_2	3·45	3·20	7·27	7·01
H_2	51·28	51·02	55·09	55·25
CO	44·19	44·70	36·52	36·62
CH_4	0·3	0·3	0·4	0·4
N_2	0·78	0·78	0·72	0·72
	100·0	100·0	100·0	100·0

taken, the gas volume was measured and from the composition and the volume of the gas the true average composition could be calculated. For the blow gas the gas composition obtained by the two methods varied by not more than 0·02 per cent. on any one constituent whilst for the blue water gas the variations were small. Table 130 shows that the continuous method of sampling adopted gives for this set of conditions a true average sample of the gases made during the water gas cycle.

VITIATION OF SAMPLES

Having collected an accurately representative sample of gas, precautions must be taken to ensure that the composition of the sample is not altered between the times of collecting and analysing. The ways in which a sample may change include contamination due to air leakage or to the residue of a previous sample remaining in the apparatus, release of a soluble constituent from solution in the confining liquid, and preferential solution of a constituent in the confining liquid or in a liquid condensing from the gas. The possibility of interaction between the sample and the material of which the apparatus is constructed must be guarded against.

TRANSPORT OF SAMPLES

It is often necessary to send gas samples from the place of origin to the place where they are examined.

A sample for analysis is preferably collected in a tube of the form shown in Fig. 298, and since the time which elapses between the taking and analysing of the sample is lengthened by the duration of the journey, and since extremes of temperature, and consequently pressure, may be experienced, it is necessary to examine the tubes closely for defects, and to make sure that the cocks are well lubricated. It is possible to store gas samples without change indefinitely in sound tubes.

Usually, when gas samples are transported from one place to another, batches of samples will be sent together. A convenient wooden case can readily be made to take a dozen sample tubes in two padded racks to which the tubes are clipped with wooden turn-buttons. The tubes must be adequately supported without straining the glass; the weak points of the tubes are obviously at the necks between the cocks and the bulb.

The plugs of the cocks must be wired in or tied with string so that they cannot be shaken out. The use of rubber bands for this purpose should be avoided because of the marked tendency for elastic bands to turn the cocks on. Samples taken over mercury should completely fill the tubes when they are to be transported. A sample tube should never be sent by post or by rail when partly filled with mercury.

Contamination of Samples. The leakage of air into a gas before the taking of the sample has already been discussed in the section dealing with the selection of the sampling point.

Air may accidentally leak in while a sample is being taken at a point where the pressure is below atmospheric. Air cannot leak into a sample tube if the pressure of the sample is always above that of the atmosphere. When taking samples at places under reduced pressure or when taking samples of hot gas, the pressure in the sample tube should be raised so that at no time subsequently does it fall below atmospheric. If a cock on a sample tube leaks slightly, it is better to lose a little of the sample than to suffer contamination with air.

Contamination with Residual Gas. Often the required volume of the sample is such that a small gas holder is used for its collection. The crown of the holder inevitably contains the residue of the previous sample. This can be reduced to a volume of the order of 0·1 per cent. by raising the water level to

cover the crown. The stand pipes within the holder must be lengthened to project above the new water level, small domes being provided to accommodate the ends of the pipes.

Contamination with Gas Released from Solution. Water or aqueous salt solutions which have been saturated with a gas will give up some of that gas if the temperature increases or if the partial pressure of that gas over the surface of the solution is reduced below that partial pressure with which the solution was in equilibrium. It is not sufficiently realised that the taking of a sample over a liquid "saturated" with the gas being sampled will nearly always result either in some solution of the constituents or the liberation from the liquid of some dissolved gas.

Solvent Action of Confining Liquids. The liquid confining a sample should have no solvent action. This ideal is most nearly attained by mercury although mercury is attacked by some gases. Apparatus making use of mercury is, however, both heavy and expensive.

For some purposes, e.g. the sampling of flue gases, water used as the confining liquid may be kept out of contact with the gas by means of a film of oil on the water surface.

With most types of gases, however, solutions of certain substances in water can be employed as confining liquids without dissolving the gases to an extent which will seriously affect the subsequent analysis of the sample.

The solubilities of gases in the solutions vary with the temperature and the pressure, but it may be assumed that the sample of gas will usually be collected under atmospheric pressure and temperature.

Solubilities of Gases in Water. When mixed gases are collected in contact with water the volume of each gas dissolved is directly proportional to its volume (partial pressure) and its solubility.

If the gases being sampled over water are flue gases of the analysis given, the composition of the gases dissolved in the water will be as shown in Table 131 when equilibrium has been established.

TABLE 131

Composition of flue gases		Solubility. Volume of gases in 100 vol. of water	Composition of gas dissolved by water	Composition of gas after standing over water until equilibrium is attained
CO_2	10	10·70	85·0	5·2
O_2	10	0·36	2·9	10·4
N_2	80	1·52	12·1	84·4

The large proportion of CO_2 which can be dissolved in the water may therefore result in serious errors in the sample of gas as analysed.

If an extreme example is taken and it is assumed that a sample vessel is half filled with water and half filled with a sample of flue gases of the above composition, and that the gas is agitated with the water or left in contact with the water until equilibrium is attained, then the gases analysed would have the composition shown in column 4 of Table 131. This is extreme, but considerable errors can occur, especially when the gas has a high CO_2 content.

The water wetting the walls of a sample tube of the usual capacity (about 250 cu. cm.) after allowing the tube to empty, is of the order of 0·25 cu. cm. The loss of carbon dioxide due to solution in this volume of water is negligible and would only reduce the carbon dioxide content of a gas containing 20 per cent. CO_2 to 19·98 per cent. CO_2.

The solubilities of gases in aqueous solutions of suitable chemicals are lower than in water alone, and many of these solutions make satisfactory confining liquids. Thus T. Gray (16) has shown that a solution of magnesium chloride ($MgCl_2$) can be utilised for gases containing much CO_2 provided the gases do not remain in contact with the solution for prolonged periods and the gas is not agitated with the solution. His figures are given in Table 132.

TABLE 132

	Percentage CO_2
Solution A { 1 part $MgCl_2.6H_2O$ / 1 part water	
Solution B { 2 parts $MgCl_2.6H_2O$ / 1 part water	
Composition of original gas	11·42
½ hour over Solution A	11·29
½ ,, ,, ,, B	11·36
1 ,, ,, ,, A	11·21
1 ,, ,, ,, B	11·32
24 hours over Solution A	9·32
24 ,, ,, ,, B	11·13
After 5 minutes agitation with equal volume of gas and Solution B ..	9·73

These results show that either of the solutions is satisfactory if the period of contact does not exceed one hour. When the period of contact is twenty-four hours the weaker solution is unsatisfactory. The results also indicate the pronounced effect of agitation of the gas with this confining liquid.

If a gas, such as town gas, water gas, etc., containing relatively little carbon dioxide is being sampled, solution B would be suitable as a confining liquid, but the period of contact should not exceed twenty-four hours and agitation of the gas and solution should be avoided.

More recent work is reported from America by K. A. Kobe (22), (23), and his co-workers, on the solubilities of carbon dioxide in several solutions (Table 133).

The results indicate that aqueous calcium chloride dissolves the least amount of carbon dioxide. It has, however, the disadvantage that the solution is very viscous and corrosive ; it does not dry when spilled, but remains as a sticky liquid. Magnesium chloride solution is less viscous and shows about the same solubility for carbon dioxide. It suffers, however, from the defect that if alkaline reagents come into contact with it, even though it may be slightly acidic, magnesium hydroxide is formed which readily blocks connecting tubes. In the absence of alkalies, however, it is quite satisfactory.

For general use a solution containing 20 per cent. by weight sodium sulphate plus 5 per cent. by volume sulphuric acid is recommended by Kobe as the best for use in technical gas analysis equipment.

Subsequently, further investigation of the solubilities of various gases and mixtures of gases in the recommended solution was made. The solution was made by dissolving 200 grammes anhydrous sodium sulphate (A.R.) in 800 grammes distilled water and adding 40 cu. cm. concentrated sulphuric acid (36N). For the experimental work precautions were taken to avoid contamination of the solution by air, etc. It was found that the solution crystallised at 14·6° C., and it is suggested that it could not be used below 16° C. without danger of precipitation. This temperature restriction would seem to be of minor importance, except that any spilled solution may crystallise out by evaporation.

Results obtained both for single gases and mixed gases are given in Table 134, confirming the suitability of the proposed solution for the purpose in question. Mixtures of air and carbon dioxide have about the same solubility as flue gas of the same carbon dioxide content. If the figures for the solubility of carbon

dioxide—air mixtures are plotted against the carbon dioxide content in the acidified sodium sulphate solution recommended by Kobe and his co-workers, the curve will be a guide to the solubility of a flue gas of the same CO_2 content.

TABLE 133. SOLUBILITY OF CO_2 IN VARIOUS SOLUTIONS
Temperature 25° C.; Total Pressure 783·5 millimetres

Salt	Concentration of salt per cent by weight.	CO_2 dissolved (*a*) vol./vol. solution	Bunsen coefficient (*b*)
None	—	0·823	0·754
H_2SO_4	5	0·746	0·683
H_2SO_4	10	0·717	0·657
NaCl	10	0·551	0·505
NaCl	20	0·334	0·306
NaCl 20 / H_2SO_4 5		0·328	0·302
NaCl	25	0·257	0·236
NaCl 20 / HCl 2		0·352	0·323
Na_2SO_4	20	0·260	0·237
Na_2SO_4 20 / H_2SO_4 5 vol.		0·263	0·242
$CaCl_2$	40	0·158	0·144
$MgCl_2$	30	0·164	0·150

(*a*) i.e. vol. of CO_2 at 25° C., 760 mm., dissolved in 1 volume of solution at 25° C.
(*b*) Vol. of gas corr. to 0° C., partial pressure of gas 760 mm., dissolved in 1 volume of solution at 25° C.

TABLE 134. SOLUBILITY OF GASES IN KOBE'S PROPOSED SOLUTION OF SODIUM SULPHATE AND SULPHURIC ACID

Gas		Gas dissolved vol./vol. solution	Bunsen coefficient
SO_2		13·5	12·5
CO_2		0·270	0·247
C_2H_2		0·343	0·324
C_2H_4		0·024	0·022
CH_4		0·0093	0·0085
C_2H_6		0·0108	0·0099
H_2		0·0073	0·0067
CO		0·0039	0·0036
O_2		0·0089	0·0081
N_2		0·0049	0·0045
Mixed gases	Comp.		
Gas	Per cent.		
Air		0·0050	0·0049
CO_2 5 / Air 95		0·0135	0·0124
CO_2 10 / Air 90		0·0235	0·0215
CO_2 14·5 / O_2 6·1 / N_2 79·4		0·0310	0·0284
CO_2 20 / Air 80		0·0447	0·0410
CH_4 40·3 / C_2H_4 39·9 / C_2H_2 19·8		0·056	0·0513

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CHAPTER XXXIII

ANALYSIS OF SOLID FUEL

MOST users of industrial coal on a small or moderate scale do not ascertain by analysis the composition and properties of the fuel supplied to them. When washed coal is derived from the same source year after year, its composition is assumed with some justification to remain reasonably constant and the staff and workmen have discovered the best methods of using it through practical—and often costly—experience. It is, however, impossible to know with what thermal efficiency a plant is operated unless the number of heat units charged to the boiler or furnace is ascertained. It is likewise impossible to know whether successive deliveries of coal are of equal value unless a check is kept upon the incoming coal by analysis. Apparently inexplicable variations in performance may frequently be explained by variations in composition. In view of these considerations some examination of the coal, even if limited to simple analysis, is recommended.

When the same class of coal is not supplied with each delivery a knowledge of the properties of the coal is of especial value to the combustion engineer. With too clean coal mechanical stokers may suffer by overheating ; dirty coal is undesirable for obvious reasons.

Any swelling or coking propensities that a coal may possess will materially affect its behaviour in gas producers and in many forms of stoker. A coking stoker, for example, requires a coking coal and will not work well on a non-coking coal. If the coal is sufficiently agglutinating to form coke on the fire, the upper surface of the fuel bed must be disturbed more frequently with the poker. A coal of this character becomes pasty and causes difficulty in forcing air through the fuel bed of a boiler furnace.

Excessive moisture content in the coal will lower the flame temperature of a furnace and may even be the cause of smoke through chilling of flames. Although moisture does not, like ash, leave a solid residue for disposal, it detracts equally from the effective value of a purchased coal. There are thus many reasons why the aid of chemical and physical tests should be sought by the combustion engineer.

The tests conducted on coal or coke to ascertain their general quality may be far-reaching and detailed or they may be of quite a simple character. The more detailed the tests, the more may be learnt about the coal. But elaborate tests require skilled personnel not only to perform the tests, but to interpret the results.

For a detailed method of examination and testing of coal and coke British Standard 1016 of 1942 should be consulted. It is only possible to give here a very brief summary of the general methods used in order to indicate their nature to the practical engineer who may not be conversant with them.

The first necessity in making an analysis of coal is accurate sampling ; accurate analysis is wasted if the sample does not properly represent the bulk. Sampling has been discussed in Chapter XXXI.

I. SIMPLE ANALYSIS

Even if it is only possible to carry out the simplest tests on coal, from the information given in Chapter I a good indication can be obtained of the general characteristics of unknown coals.

A simple analysis of coal will involve :—

(*a*) The taking of an accurate sample for (1) moisture and (2) analysis.

(*b*) Determination of moisture. For wet coal the sample of 10–30 lb. is treated as described in Chapter XXXI, section A, " Treatment of the Gross Sample," to determine moisture removable by air-drying (pp. 9, 10, B.S. 1016). The remainder is reduced to a 2-lb. sample and its inherent water content ascertained either by drying in an oven at 110° C. or by distillation as described on pp. 10–12 of B.S. 1016.

(*c*) Determination of ash. The laboratory sample is ground to pass 72-mesh B.S. sieve and a weighed quantity, 1–2 grammes, is burnt to ash in a muffle furnace (B.S. 1016, p. 30).

(*d*) Determination of volatile matter and appearance of the coke produced in this test. From the powdered coal (72 B.S. mesh) 1 gramme is weighed into a crucible for determination of volatile matter according to B.S. 1016, pp. 25–28. This sample is, in effect, carbonised in a closed crucible at 925° C. and the weight of solid non-volatile residue ascertained. The appearance of the solid residue is also noted.

The quantity of inherent moisture present in a coal affords considerable information as to its general character. Coals of higher rank, i.e. coal of high carbon content and low volatile matter content (anthracitic coals) have little inherent moisture (1–2 per cent. and even lower). For higher-volatile coals the inherent moisture content is closely connected with the coking properties of the coal. A good coking coal, such as would make metallurgical coke, will generally contain from 1–3 per cent. of inherent moisture, and the gas coals slightly more, say 2–4 per cent. ; as the coking powers diminish the inherent moisture content increases. At about 5–7 per cent. the coal may produce a flat, coherent " coke " when carbonised in a closed crucible in the laboratory, but would not make a saleable coke in carbonisation practice. Non-coking coals, such as those from the Leicester and Warwick coalfield contain some 11–14 per cent. of inherent water and form a very friable and barely coherent " coke " in the crucible.

From the appearance of the ash obtained in (*c*) some information as to its character can be deduced. It can be assumed generally with a good degree of probability that a red ash will be easily fusible and a white ash will fuse at a relatively high temperature. The information contained in Chapter VI under " Clinker Formation " should, however, be noted. From the total moisture content and the total ash content the proportion of incombustible inorganic matter contained in the coal or coke can be ascertained. The calorific value of coke can be deduced from these figures by the formula given in Chapter II under " Coke."

From the volatile content of the coal (*d*) calculated to the dry, ash-free basis coupled with the appearance of the solid residue (e.g. powdered, slightly coherent, coked, swollen, etc.), the general character of the coal can be ascertained and its calorific value approximately deduced from the information given in Chapter I and from Tables 1 and 2 in that chapter.

The general characteristics of unknown coals can thus be deduced from simple tests.

II. MORE COMPLETE ANALYSIS

The more complete analysis of coal will comprise the determination of the following :—

Impurities :
(i) Moisture removable by air drying.
(ii) Total moisture.
(iii) Ash content.
(iv) Sulphur content.

Characteristics : (i) Volatile matter content calculated on dry coal.
(ii) Agglutinating and swelling properties.
(iii) Type of coal.

Quality : (i) Size grading of coal.
(ii) Calorific value.
(iii) Elementary or " ultimate " analysis.

GENERAL METHODS OF ANALYSIS

The determination of moisture, ash and volatile matter in accordance with B.S. 1016 have been briefly described under " Simple Analysis." Sulphur is ascertained by purely chemical methods.

The swelling characteristics of coal are determined by the Woodall-Duckham method, which is also to be found in B.S. 1016, or by the other methods mentioned in Chapter I. The general method of determination of the type of the coal will be evident from what has been said in Chapter I, particularly Tables 1 and 2, and previously in this chapter.

In terms of their volatile matter and agglutinating power, coals can be put into the classes there described under " Classification of Coals." When a coal has been put into its class the combustion engineer will know its general characteristics.

The melting point of the ash is determined by observing by means of an optical pyrometer the temperatures at which moulded triangular pyramids of ash, when heated in a mildly reducing heating atmosphere, first show signs of softening and at which they have completely fused into a blob or button (B.S. 1016, p. 89).

The " elementary " analysis of a coal involves the determination of carbon, hydrogen, nitrogen, sulphur and ash, oxygen being taken by difference as there is no accurate method of determining this element in coal (B.S. 1016, pp. 31–45). Special chemical methods are required for nitrogen and sulphur. The carbon and hydrogen are determined by burning a small quantity of coal in a current of oxygen under carefully controlled conditions, so that no carbon monoxide is formed, and weighing the quantity of carbon dioxide and water vapour produced. This test is difficult in that it requires a good deal of experience on the part of the operator and in general accurate results are not obtained on a newly set up apparatus until several determinations have been made.

The direct determination of the calorific value of coal (B.S. 1016, p. 55) is effected by burning a weighed quantity of the coal (generally about 1 gramme) in oxygen under such conditions that the whole of the heat generated can be measured. The combustion chamber is immersed in water. From the rise in temperature of the water, corrected for loss by radiation, etc., and knowing the water equivalent of the apparatus, the total heat evolved can be measured.

It will be appreciated that the simpler tests such as those for moisture, ash content and size grading (cf. Chapter XXXI) can be performed by an intelligent semi-skilled operator with a little practice, but that the services of a chemist are required for conducting most of the tests satisfactorily.

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CHAPTER XXXIV

THE SELECTION OF FUELS FOR INDUSTRIAL PURPOSES

Cost and operational factors on which selection must be based—The characteristics of the fuels available.

THE selection of fuels for industrial purposes involves taking account of many considerations. In this chapter some account is given of the principles of fuel selection, though it must be regarded as nothing more than a guide. Local conditions and the availability of certain fuels may be governing factors which will cause a fuel to be selected for a particular purpose in one place to the exclusion of another fuel which may have advantages under other circumstances for the same purpose. Thus no hard-and-fast rules can be laid down, but the choice must be a matter for individual selection having regard to the special local conditions.

The fuel bill as an item in total works costs may often appear very small, so small as not to merit any deep consideration. Nevertheless, the fuel and the equipment for its use may make a very great difference to the output of the works, both in rate of working, e.g. in the time of heating-up and in the number of wasters. Some fuels may be easier to handle by the available labour than others.

A 20 per cent. increased output on the same plant and with the same labour costs is obviously a very big item and may easily be achieved—or lost—by the intelligent selection of fuel, equipment and instruments, thus greatly affecting the cost per unit of output although the cost of fuel *per se* may remain negligible as an apparent item in the process costs.

PRICE

The obvious starting point, though not invariably the best one, is price. To compare different fuels, the first step is to reduce the price to a uniform basis of price per therm. Thus, for example, a table of possible fuels might be made out, somewhat as in Table 135, which is purely illustrative and based on pre-war figures (P.E.P. Report on the Gas Industry in Great Britain, p. 90).

TABLE 135

Fuel	Price	Price per therm pence
Coal	26s. per ton	1·0
Coke	25s. per ton	1·1
Coke breeze	7s. 6d. per ton	0·5
Producer gas	2d. a therm	2·0
Oil	72s. a ton	2·0
Town gas	4d. a therm	4·0
Electricity	0·79d. a unit	23·0

Having ascertained the price per therm, the next step is to determine as closely as may be possible in advance, the cost per *useful* therm. This involves knowing or estimating the thermal efficiency with which the various fuels will be used. The thermal efficiency varies considerably from one process to another, and there may or may not be a similar variation between the efficiency attainable with various fuels for the same process.

Details cannot therefore be given of this calculation in terms of general industrial efficiency of the several fuels. For any particular set of conditions

the required figures are obtained by calculating the cost in pence per useful therm in the following manner, which is illustrated by taking two hypothetical fuels A and B :—

Fuel	Initial cost per therm pence	Efficiency in use	Therms utilised per initial therm	Cost per therm used pence
A	1·0	40 per cent.	0·4	(1/0·4 =) 2·5
B	2·5	70 ,, ,,	0·7	(2·5/0·7 =) 3·6

Another example may be quoted of three coals for use in a boiler plant. All these coals were suitable for this particular plant, and they were priced differently. Preliminary tests showed marked differences in evaporative power, and the final results were as follows :—

Coal	A	B	C
Price per ton delivered	22s. 9d.	22s. 4d.	18s. 9d.
Equivalent evaporation from and at 212° F., lb. water per lb. coal	8·62	7·7	6·5
Cost of coal per 1,000 lb. steam generated (pence)	14·14	15·53	15·45

This calculation indicates that coal A was the cheapest for this particular installation.

The comparison is thus made not on price per unit of heat as received, but upon the cost per unit of heat utilised.

In many processes it is not possible to give an " efficiency " figure, since the term " thermal efficiency " has no clear meaning. The calculation is the same, however, if the heat units required per unit of output are taken as the basis. The fuel cost per unit of product is a fundamental figure of interest to the user. The significance of this figure, as will be explained, may be reduced by other considerations.

RECOVERY OF WASTE HEAT

A fuel that otherwise appears too high in cost may be capable of showing an improved efficiency if adequate methods of heat recovery are practised.

HANDLING CHARGES

The bare cost of heat units supplied to a process is only one of many factors which are involved in the cost of heating. A solid fuel, for example, has to be brought into the works, unloaded and stored. This will probably involve labour, elevators and storage bunkers ; capital expenditure, power and maintenance charges are incurred thereby. In addition to bunkers or hoppers, solid fuel is generally stored on the ground in bulk against possible non-delivery of supplies to time, etc. Handling costs are involved in taking the material from the store or from the place where it is unloaded to the various parts of the works ; for this there will be needed conveyors or railway lines with haulage engines, etc.

When the fuel has arrived at the place where it is to be used it may be fed mechanically into the furnace or it may still require further handling, e.g. the hand firing of boilers. Solid fuels involve further handling charges for the removal of the ashes and clinkers.

Gas producers also require similar handling charges for the solid fuel and ashes, and this must be taken into account when assessing the net cost of producer gas. Electricity involves no more handling costs than the installation and upkeep of the necessary cables and the wages of a switchboard attendant, though there may be transforming costs in addition. High tension current is

measured at the inlet of transformers and the consumer bears transformer losses. Town gas, if used at the pressure at which it is supplied by the gas undertaking, involves no handling charges other than the installation and maintenance of the gas equipment throughout the works ; it is often convenient to boost the supply up to a higher pressure and to use high pressure air ; the costs involved in power, machinery and attendance must be taken into account. Oil involves storage tanks, pipe-lines, pumping machinery, and often heating the oil before pumping, the capital, operating and maintenance costs of which must be added to the bare cost of oil.

On the other hand, electricity cannot be stored and gas can only be stored in a limited quantity. Coal for the production of gas and electricity is, of course, stored at the works in quantity, the consumer being thus safeguarded against failure of supplies.

The labour and operating costs involved in handling fuels and their waste products must, therefore, be added to the initial cost of the fuel per useful therm.

COST AND AVAILABILITY OF GROUND SPACE

All fuels require a certain amount of space in the works. In some works this is easily arranged, in others ground space may be limited. The cost involved in storage has been mentioned, but the ground space must also be taken into consideration. If continuous operation is essential, the possibility or impossibility of affording the necessary stocking space may govern the selection of fuel. Gas and electricity require virtually no space except for mains, cables, meters, switchboard, transformers, etc. Coal and coke involve weighing, bunkering, conveyors and possibly railway facilities. Producer plant requires considerable ground space. Oil tanks can be placed overhead or underground (at some additional expense) so long as the quantity stored is not too great.

CONVEYANCE OF FUELS

In addition to the handling charges, the arrangement of the conveying plant must receive attention. There may be obstacles in the layout of the works which will cause difficulties, or at least additional expense, in arranging the necessary railway lines, conveyors, pipes and mains or cables.

Producer gas requires large mains to convey the gas about the works. Town gas or coke oven gas also requires mains. It may be that the layout of the works will involve placing these mains underground, and if so, the cost of excavation and making good the ground must be taken into account and balanced against the cost of rail or conveyor transport for solid fuel.

QUALITY OF PRODUCT

In some branches of industry temperature control has a very important bearing on the quality of the product. A dust-free gaseous fuel, oil fuel, or electricity will in general enable the furnace temperatures to be more exactly controlled than will a solid fuel.

In Chapter XVIII on furnaces, particular attention was directed to the importance of controlling the composition of the furnace atmosphere in certain industrial operations. The promotion of scaling by oxidising agents and sulphur dioxide was there discussed. The need for controlling continuously and with certainty the oxidising character of the atmosphere in internally-heated furnaces may govern the choice of fuel.

RANGE AND CONTROL

The range of temperatures that can be reached with competitive fuels will often be of supreme importance when high temperatures are involved. The degree to which the gas and air may be preheated, for example, may control the maximum temperature attainable.

Closely bound up with this point is the applicability of automatic control apparatus, including thermostatic control, and pressure control. If control is manual, care must be taken that the nature of the process lends itself to reasonably effective operation with the fuel selected.

Automatic controls require careful handling to ensure that their efficiency is maintained. Generally the makers will undertake to service their instruments, but in addition to this it is often necessary to consider whether the personnel available are capable of operating the controls between servicing periods.

OUTPUT

The properties of the fuels used may determine the output from a given operation. Oil fuel, for example, with its highly radiating flame, enables a furnace to make a quicker start from the cold or from a lower temperature than a fuel such as coal, which is slower to attain its full temperature. In intermittent furnaces this may be an important consideration.

Wherever possible the performance of similar equipment on work elsewhere should be ascertained as a guide.

FURNACE MAINTENANCE

The importance of keeping the furnace structure in first-class repair has been emphasised in other chapters in this book. Large fluctuations of temperature are detrimental to brickwork, and, conversely, regular and uniform temperatures will help to preserve the brickwork.

The slagging action of the ash of solid combustibles is a source of damage to brickwork which cannot be neglected. This damage may be so serious as to become a major problem, or it may, as in gas-making retorts, be spread over so long a period of time that it is relatively unimportant.

With fuels not containing ash that may be carried into the furnace, insulating refractories can be used which enable considerable savings to be made in initial cost, size and thermal capacity of furnace.

Spare parts will always be required and the availability of these spare parts is important.

TECHNICAL ASSISTANCE

All firms have not highly skilled staffs accustomed to furnace operation. One quite important consideration is the extent to which help can be obtained from without, either from the suppliers of the fuel or from the makers of the equipment in which it is used. The Industrial Gas Centres of the gas industry, the Coal Utilisation Joint Council of the coal industry, the Electrical Development Association and the Coke Associations are examples of technical bodies connected with fuel industries from which outside help can be obtained without charge when required. Advice may also be sought from consultants of repute, whose names are obtainable from the professional institutions. The Fuel Research Station and the local Coal Survey Laboratories of the D.S.I.R. are always ready to give advice and assistance.

Certain guarantees of performance are often given by the makers of equipment. The selection of a fuel on the basis of performance of the plant will naturally be based to some extent upon these guarantees, but it is necessary

to consider how far the working results when the plant is new and expertly handled will be maintained in day-to-day operation.

MAINTENANCE OF SUPPLIES, PRICE AND QUALITY

Before finally selecting a fuel, steps should be taken, especially under war conditions, to ascertain whether the desired fuel can be obtained. This may seem elementary, but omission of this precaution has several times led to grave difficulties.

Having obtained satisfactory assurances on availability, the maintenance of supplies of suitable fuel must be considered. There should be no difficulty on this score in normal times unless the appliance selected is highly sensitive to small changes in quality. Quality changes cannot always be avoided to-day, however, and due weight must be given to this factor in fuel selection.

Price stability cannot be guaranteed in war conditions, but since all fuels except oil are closely tied up with coal, any increase in the price of coal brings about a corresponding increase in the price of coke or coke breeze, and to a lesser extent in the price of gas and electricity. It is important to the consumer who is considering the type of plant to be installed and the relative cost of his fuel, that coal prices should be stabilised, so far as is possible, on a rational assessment of heat values, and should not be subject to the substantial variations in the relative prices of different coals which have been experienced over recent years.

WORKING CONDITIONS FOR OPERATIVES

The avoidance of heavy work, a reduction of temperature around furnaces, an improved atmosphere freed from smoke and dust may materially improve the health and capacity for good and sustained work of those who operate the plant. The cleanliness of some fuels compared with others is often an important consideration.

Fire and accident risks come under the same general heading. This may also exert an influence on the insurance premium demanded against these risks.

THE CHARACTERISTICS OF FUELS

The foregoing discussion has been devoted to the general considerations that must be taken into account when selecting a fuel for a particular purpose. In that discussion, and in other parts of the book, the characteristics of the fuels have been repeatedly mentioned. It will thus be of value to discuss these characteristics insofar as they affect selection.

I. SOLID FUEL

COAL

Ash troubles, the clinkering of the fuel bed choking the air flow, and imperfect combustion involving the production of smoke may be associated with the use of solid raw coal as a furnace fuel though these difficulties are greatly reduced if the fuel is properly selected and if supplies of the selected coal are maintained. The mechanical stoker offers pronounced advantages in many respects. In small units (up to 5 cwt. of coal per hour) the main difficulties are economic, e.g. the provision of the necessary mechanical gear to feed the fuel bed regularly with raw coal and to discharge the ashes, at a cost in keeping with the maintenance and capital charges which can be borne by a unit with so small a rate of fuel consumption. In large units troubles arise from the high temperature of the surface of the fuel bed, and the wear of tuyeres and refractories utilised in side walls and crowns may become excessive if high-grade materials are not used. Modern heat-resisting alloys meet the first requirement, and sillimanite and high aluminous bricks the second.

The mechanical stoker, with primary air preheated up to some 400° F. and water vapour admission with coals yielding the more fusible ashes, offers the possibility of a high measure of economy in solid fuel firing.

PULVERISED COAL

In industrial furnace installations some preference is shown for unit systems in which one pulverising plant supplies one furnace unit, or a compromise between the unit and central systems in which the pulveriser is operated in conjunction with a storage bunker from which the various pipe-lines are fed by means of separate conveyors and primary air fans.

Another method in operation is to provide the coal already pulverised at central stations and convey it in covered waggons to bunker storage at the furnace plant. Problems of wear and tear of refractories, flue dust, the design of burners and their correct disposition in the furnace chamber to give uniform heating, and the cost of pulverisation, must also be taken into account.

The adoption of pulverised coal on a considerable scale generally involves some nuisance through the emission of dust. Electrostatic precipitators or flue gas washing plants are frequently installed to avoid this trouble, but the capital and operating cost of these plants must be taken into account.

COKE

Coke is applicable to certain classes of heating, notably drying, steam-raising and metallurgical operations. Its manner of combustion is related to its physical properties, the harder cokes burning with a higher fuel bed temperature than the softer cokes. Cokes of low reactivity are thus the more suitable for use in high temperature operations, such as melting in reverberatory hearth furnaces, crucibles, and cupolas. In general, for heating purposes, where mechanical or adequate natural draught is available, " hot " cokes, the general character of which is best revealed by the shatter test, are the most suitable.

The high radiant efficiency of a coke fire, due to carbon having an emissivity approaching unity, gives to coke a special value for many metallurgical operations, such as the blacksmith's forge, ladle heating and mould drying. The admixture of coke with coal provides a means of lighting up furnaces from cold without excessive production of black smoke.

2. THE SELECTION OF COALS FOR COMBUSTION

Apart from the general principle of selecting a fuel of a particular character for a particular purpose, there is a fundamental difference between coal and other fuels. There is no appreciable difference between one town gas and another, and if town gas is selected for a heating job, there is no choice (and no necessity for choice) as to the source of supply. Similarly, only the voltage and other characteristics of electricity need be considered if electricity has been selected on general grounds. Producer gas may be cleaned or crude, it may be cooled or hot, but the differences between one producer gas and another are inconsiderable except when there is appreciable divergence in the hydrogen content.

Coal, however, is very far from being a standardised product, and may vary from high-volatile bituminous coal to anthracite, from low ash content to high ash content, and in other ways. If coal has been chosen as the heating agent, whether for direct heating, or for conversion into producer gas, it is further necessary to select a particular coal for the purpose or to choose between a number of coals offered.

It may be known through long experience that certain types of coal as designated by their trade descriptions are suitable for definite purposes under

well-known conditions. If so, selection may not be difficult, though it is always worth considering if the coals used are the best that can be found for the purpose.

Among the important characteristics that must be taken into consideration are :—

(1) Suitability for conditions under which the fuel is to be used.
(2) Calorific value.
(3) Volatile content.
(4) Analysis, particularly moisture, ash, sulphur, hydrogen, oxygen, and fixed carbon content.
(5) Ash fusion temperature.
(6) Agglutinating, or coking, properties.
(7) Size and uniformity of grading.
(8) Rate of burning.
(9) Weathering and storage characteristics, including friability.
(10) Grindability, if for use in pulverised form.

Solid fuels for steam-raising purposes may be grouped broadly as indicated in Chapter I.

All the simple methods of classification which have from time to time been proposed, e.g. " chemical," " volatile yield," " use," " trade," and so on, are, on account of the diversity of the information sought and the complexity of the raw material, too indefinite to indicate with any certainty the rapidity of burning, mechanical properties, and coking qualities which may be expected.

(1) *Suitability of Coals.* Laboratory tests give much useful information as to the value of coals and the way in which they will behave in practice. Unless the analyst is skilled in interpreting his results and, moreover, uses tests that will give the vital information required, the user will be compelled to make practical trial to obtain the necessary information. This is, in fact, the only final proof of the suitability of coals for a particular purpose.

Sized high-carbon solid fuels can be burned successfully in any ordinary furnace ; but with the same draught they burn at a lower rate of combustion than high-volatile coals. Consequently the thermal output will be reduced.

Some conditions call for slow-burning fuels, while others, particularly at peak periods, require quick-burning flashy, high-volatile coals.

(2) *Calorific Value.* If the coal is otherwise suitable, its value will depend partly on its calorific value, this being a measure of the potential heat content of the coal as purchased.

(3) *Volatiles and Analysis.* The smoke-producing properties of coals may generally be assumed to vary directly as their volatile constituents except for coals containing upward of 40 per cent. V.M. on the dry ash-free basis.

Moisture may be advantageous or detrimental, according to the nature of the coal used. Some fine coals cannot be burned satisfactorily without being wetted.

The washing of a coal with 10 per cent. ash content may reduce the ash to 6 per cent. ; but the moisture may be thereby increased by 4 per cent. There is, therefore, no gain in calorific value, and someone must pay for the washing. Nor is there any real saving in freight, since moisture is being carried instead of ash. The inference has been drawn that if it be necessary to wet coal before firing, it is more profitable to do so on the site than to pay for water and its transport at the price of coal. On the other hand, some authorities hold the view that while too much moisture is undesirable, washed, well-drained coal, homogeneous in size and slightly damp throughout, gives the most efficient results with chain-grate stokers. They emphasise that the coal must be *uniformly* moist, and that it is no use buying dry coal and then spraying it just before putting it into the stoker, since uniformity of moisture content cannot be secured in this way.

Excessive ash is undesirable, but grate maintenance charges increase rapidly with low-ash coals especially when the fuel bed temperature is high.

(4) The coking powers of the fuel considerably affect the behaviour of the coal on combustion as has been indicated in previous chapters.

3. PRODUCER GAS

Producer gas, as a heating agent, suffered a set-back in the years before the war owing to a growing tendency for furnace operations to be confined to the day shift, since separate producer units carry a heavy stand-by charge for lighting up. There is also a gasification loss of 15–25 per cent. of the potential heat of the fuel according to the type of plant used. Nevertheless, it still remains an economical and easily controllable means of heating large furnace batteries in continuous operation under conditions permitting adequate recuperative or regenerative practice.

The mechanical gas producer constitutes the most economical type of unit for large batteries. There must be suitable provision of dust catchers and tar drip wells for the avoidance of choking. The tarring-up of valves and the necessity for burning out the flues at a " shut down " are inevitably associated with the utilisation of uncleaned coal producer gas.

For small units the built-in producer has the advantage of conserving appreciable amounts of heat otherwise lost in gas mains, and offers means of simplifying construction. Coke is the fuel usually used in built-in producers.

4. COAL GAS

Town gas and coke oven gas, except in special circumstances, involve a higher cost per therm of heat delivered at the furnace than solid fuel or producer gas. Nevertheless their greater " adaptability " value and higher efficiency of application often offsets their higher price, and their distribution and utilisation require a lower capital outlay than producer gas. Modern burner design permits of good combustion conditions being readily obtained in a simple manner. With the correct disposition of burners uniformity of heating is attainable, and the applicability of this form of gaseous fuel to methods of surface combustion admits of high radiation effectiveness.

Town gas is used for industrial heating in over 3,000 distinct trades, in each of which it is employed on the average for seven or more processes. Coke oven gas has such advantages in the reheating operations of steel works practice that the tendency is to heat the ovens by a lower grade gas in order to conserve a larger quantity of the higher grade gas for other purposes. The development of gas grids in industrial districts to take surplus coke oven gas is an outcome of the proved value of the fuel for many industrial purposes.

For the smaller types of industrial heating furnaces covering a wide range of service in repetition work, in which the relation of the number of articles to their individual heat capacity is high, gaseous fuel has an enhanced value in convenience of control, flexibility, economy of floor space required by the plant, fuel storage, and suitability for intermittent operation brought about by the possibility of using insulating refractories.

5. OIL FUEL

Oil fuel is convenient in handling and readily controlled, but its main advantage lies in its high flame intensity.

The high intensity of heating practicable with oil makes it a desirable fuel for operations at high temperatures, notably in the melting of non-ferrous metals and for " in and out " furnaces for small forgings, drop stamping and rivet heating. Where oil has replaced coke its continued use has been justified on the ground that an increased rate of output became possible. Oil also shares with town gas, and to a lesser extent with producer gas, the advantages over

solid fuel of ease of lighting or shutting down, capacity for accurate regulation including automatic temperature control, cleanliness, reduction in floor space, and so forth. The reduction of time in operating biscuit ovens on oil in the ceramic industry has proved one dominant factor in its application. In biscuit ovens, glost ovens and continuous tunnel kilns in the ceramic industry, gas and oil both give facility to the operator for controlling the rate of firing, the furnace atmosphere and the temperature with resulting benefits to the quality of the ware.

Oil fuel has a high flame intensity, and this necessitates special care in furnace design, especially where the temperature must be very uniform. Oil has been used, for example, to heat many heat treatment furnaces where the most accurate control is required to give uniformity. To secure satisfactory results there must be proper furnace design, the correct choice of oil burner equipment and skilled calculation of heat distribution.

Similarly good furnace design is needed to secure the refractories against erosion from oil flames. Very few oil-fired furnaces require special refractories, though in earlier installations this erosion provided a problem which was solved in several ways.

Carborundum may be used, though high in first cost ; sillimanite is suitable, its quality having been considerably improved in recent years. Other practical methods include the use of grog firebricks, high in alumina, patchings of silica-fireclay millings, plumbago bricks, and even chrome-magnesia refractories, provided conditions as regards spalling are favourable.

To keep refractory face temperatures substantially within the limits allowable with good refractories of reasonable commercial grades, furnaces must be so designed that both the formation of flames and the radiation from them are under control. Much depends, of course, on the rapidity with which heat can be removed from the zones of highest temperature.

In extensive oil-burning equipment adequate control of the viscosity by suitable provision for heating in cold weather is needed (cf. Chapter XXVIII). Coal tar oils and pitch, as indicated in Chapter II, may replace petroleum oils.

6. ELECTRICITY

Electricity, though on a thermal basis the most expensive of the available fuels, has compensating advantages which lead to its use for industrial purposes. Its efficiency in use approaches 100 per cent. For furnace work, the primary characteristics are exactitude of control and flexibility. It lends itself, like town gas and purified producer gas, to automatic control. The furnace atmosphere can be adapted to suit the process ; there is no risk of sulphur contamination. The cost of chimneys is saved.

CONCLUSION

Whilst no complete rules can be given for the selection of fuel, it will have been evident from this necessarily brief survey that far more must be considered than the bare cost of the fuel. Some of the factors here listed are sometimes overlooked until it is too late with consequent disappointment, waste and expense. There is often a tendency to give inadequate consideration to fuel selection on the ground that the fuel cost is a small proportion of the total cost of the finished goods, and need not therefore receive the same attention as other aspects of manufacture. It is sometimes in these very circumstances that the choice of fuel needs the most experienced and searching enquiry. By achieving a greater throughput or reduced wastage (which comes to the same thing) production costs per article may be so reduced as more than to offset a higher fuel or furnace cost. This is clearly of equal benefit in war and peace.

APPENDIX I

THE SURVEY OF BRITISH COALS

THE geological data relative to the various coalfields were believed to be fairly complete for many years, but, until recently, information about the nature of the coal seams and their commercial value was hard to obtain and lacked correlation.

The Lancashire and Cheshire Coal Research Association commenced a systematic survey of their coalfields in 1918, and in 1922 the work of this Association was combined with the Government-sponsored "Physical and Chemical Survey of the National Coal Resources." The Coal Survey Laboratories form part of the fuel research organisation of the Department of Scientific and Industrial Research, and are situated in, or near, the principal coalfields of the country. The Coal Survey staff have been engaged for many years upon the detailed study of the characteristics of the coal seams, and of the various local coals as marketed, and have amassed a fund of information, much of which is available to *bona fide* enquirers. The headquarters of the Coal Survey is at the Fuel Research Station, and enquiries addressed to the Director of Fuel Research (Department of Scientific and Industrial Research) at the Fuel Research Station or to the Officers-in-Charge of the local Coal Survey Laboratories are always welcome.

The addresses of the Coal Survey Laboratories are given below :—

Coalfield	Address of Coal Survey Laboratories
Lancashire & Cheshire *	7, Park Street, Cheetham, Manchester, 3.
South Yorkshire	Portobello Street, Sheffield, 1.
West Yorkshire	The University, Leeds.
Nottinghamshire & Derbyshire, Leicestershire & South Derbyshire	16–18 Shakespeare Street, Nottingham.
North Staffordshire, North Wales & North Shropshire	Old Government House, Dee Hills Park, Chester.
South Staffordshire, Warwickshire, Cannock Chase & Forest of Dean & South Shropshire	St. Martin's Street, Edgbaston, Birmingham.
Northumberland & Durham, Cumberland	King's College, Newcastle-on-Tyne.
South Wales, Bristol & Somerset..	128, Newport Road, Cardiff.
Scotland	Royal Technical College, Glasgow.
Kent	Fuel Research Station, River Way, E. Greenwich, S.E.10.

Considerable progress has been made in the examination of the country's coals and the results have been published in the following Fuel Research Survey Papers :—

Survey Paper No.	Title	Year of Publication
3	The Lancashire Coalfield. The Arley Seam	1924
4	The Lancashire Coalfield. The Ravine Seam. Part I ..	1925
5	The Lancashire Coalfield. The Smith Seam	1925
6	The Lancashire Coalfield. The King Seam	1926
9	The Lancashire Coalfield. The Ravine Seam. Part II. Carbonisation in Continuous Vertical Retorts	1927
10	The Lancashire Coalfield. The Wigan Four Feet Seam ..	1927
13	The Yorkshire, Nottinghamshire and Derbyshire Coalfield. South Yorkshire Area. The Parkgate Seam	1929

* Work in the Lancashire & Cheshire coalfield is undertaken for the Fuel Research Board by the Lancashire & Cheshire Coal Research Association.

Survey Paper No.	Title	Year of Publication
14	Description of the Coalfields of North Staffordshire	1929
18	The Yorkshire, Nottinghamshire and Derbyshire Coalfield. South Yorkshire Area. The Barnsley Seam	1931
19	The Lancashire Coalfield. The Burnley, Accrington, Darwen and Bacup Area. The Lower Mountain Mine	1931
20	The Yorkshire, Nottinghamshire and Derbyshire Coalfield. South Yorkshire Area. Analysis of Commercial Grades of Coal. Part I	1931
21	The Northumberland and Durham Coalfield. Northumberland Area. The Yard Seam	1931
22	The Northumberland and Durham Coalfield. Northumberland Area. The Main Seam	1932
24	The Northumberland and Durham Coalfield. Durham Area. The Brockwell Seam..	1932
25	The Yorkshire, Nottinghamshire and Derbyshire Coalfield. South Yorkshire Area. The Silkstone Seam	1932
27	The Northumberland and Durham Coalfield. Durham Area. The Hutton Seam	1933
30	The Yorkshire, Nottinghamshire and Derbyshire Coalfield. Nottinghamshire and Derbyshire Area. The Deep Hard Seam..	1933
31	The Yorkshire, Nottinghamshire and Derbyshire Coalfield. South Yorkshire Area. Analysis of Commercial Grades of Coal. Part II	1934
32	The Lancashire Coalfield. Miscellaneous Seams of the Lower Coal Measures	1934
33	The Lancashire Coalfield. Analysis of Commercial Grades of Coal. Part I	1934
34	The Northumberland and Durham Coalfield. Northumberland Area. The Plessey Seam	1934
35	The Yorkshire, Nottinghamshire and Derbyshire Coalfield. West Yorkshire Area. The Beeston Group of Coals. Upper Beeston Seam. Part I	1935
36	The Northumberland and Durham Coalfield. The Beaumont Seam. Part I. Northumberland Area	1936
37	The Yorkshire, Nottinghamshire and Derbyshire Coalfield. Nottinghamshire and Derbyshire Area. Analysis of Commercial Grades of Coal. Part I	1937
38	The Yorkshire, Nottinghamshire and Derbyshire Coalfield. West Yorkshire Area. The Wheatley Lime Seam	1937
39	The Coal Seams of North Staffordshire	1937
40	The Lancashire Coalfield. The Trencherbone Seam. Part I ..	1937
41	The Northumberland and Durham Coalfield. The Busty Seam..	1937
42	The Cumberland Coalfield. The Little Main Seam	1938
43	The Leicestershire and South Derbyshire Coalfield. South Derbyshire Area. The Stockings Seam	1938
45	The North Wales Coalfield. The Main Seam	1938
46	The Yorkshire, Nottinghamshire and Derbyshire Coalfield. West Yorkshire Area. The Barnsley-Warren House Seam ..	1939
47	The Northumberland and Durham Coalfield. Northumberland Area. The Bensham Seam..	1939
48	The Yorkshire, Nottinghamshire and Derbyshire Coalfield. Nottinghamshire and Derbyshire Area. Analysis of Commercial Grades of Coal. Part II	1939
49	The Cumberland Coalfield. The Six-Quarters Seam	1940
51	The Yorkshire, Nottinghamshire and Derbyshire Coalfield. West Yorkshire Area. The Beeston Group of Coals. Upper Beeston Seam. Part II	1941
52	The Leicestershire and South Derbyshire Coalfield. South Derbyshire Area. The Kilburn Seam	1940
53	The Yorkshire, Nottinghamshire and Derbyshire Coalfield. Nottinghamshire and Derbyshire Area. The Top Hard Seam..	1942
54	The Leicestershire and South Derbyshire Coalfield. South Derbyshire Area. The Eureka Seam	1942
55	The Coals of South Wales	1942
56	Description of an Isovol Map of the South Wales Coalfield ..	1944
	An Isovol Map of the South Wales Coalfield	1944

APPENDIX II

TABLE 1. PROPERTIES OF SATURATED STEAM*

Pressure		Temperature	Heat B.Th.U. per pound			Volume Cu. ft. per pound		Entropy B.Th.U. per °F. per pound		
Pounds/ sq. in. absolute	Vacuum inch Hg.	°F.	Sensible liquid	Latent	Total vapour	Liquid	Vapour	Liquid	Evaporation	Total vapour
0·2	29·51	53·1	21·2	1063·8	1085·0	·0160	1526	·042	2·074	2·116
0·3	29·31	64·5	32·5	1057·4	1090·0	·0161	1040	·064	2·017	2·081
0·4	29·11	72·9	40·9	1052·7	1093·6	·0161	791·9	·080	1·976	2·056
0·5	28·90	79·6	47·6	1048·8	1096·4	·0161	641·4	·092	1·945	2·037
0·6	28·70	85·2	53·2	1045·7	1098·9	·0161	540·0	·103	1·919	2·022
0·7	28·50	90·1	58·1	1042·9	1101·0	·0161	466·9	·112	1·897	2·009
0·8	28·29	94·4	62·4	1040·4	1102·8	·0161	411·7	·119	1·878	1·997
0·9	28·09	98·2	66·2	1038·3	1104·5	·0161	368·4	·126	1·861	1·987
1·0	27·89	101·7	69·7	1036·3	1106·0	·0161	333·6	·133	1·845	1·978
1·2	27·48	107·9	75·9	1032·7	1108·6	·0162	280·9	·144	1·819	1·963
1·4	27·07	113·3	81·2	1029·6	1110·8	·0162	243·0	·153	1·797	1·950
1·6	26·66	118·0	85·9	1026·9	1112·8	·0162	214·3	·161	1·778	1·939
1·8	26·26	122·2	90·1	1024·5	1114·6	·0162	191·8	·168	1·761	1·929
2·0	25·85	126·1	94·0	1022·2	1116·2	·0162	173·7	·175	1·745	1·920
2·2	25·44	129·6	97·5	1020·2	1117·7	·0162	158·9	·181	1·731	1·912
2·4	25·03	132·9	100·8	1018·3	1119·1	·0163	146·4	·186	1·718	1·905
2·6	24·63	135·9	103·8	1016·5	1120·3	·0163	135·8	·192	1·706	1·898
2·8	24·22	138·8	106·7	1014·8	1121·5	·0163	126·7	·196	1·696	1·892
3·0	23·81	141·5	109·4	1013·2	1122·6	·0163	118·7	·201	1·685	1·886
3·5	22·80	147·6	115·5	1009·6	1125·1	·0163	102·7	·211	1·663	1·874
4·0	21·78	153·0	120·9	1006·4	1127·3	·0164	90·63	·220	1·643	1·863
4·5	20·76	157·8	125·7	1003·6	1129·3	·0164	81·16	·228	1·625	1·853
5·0	19·74	162·2	130·1	1001·0	1131·1	·0164	73·52	·235	1·609	1·844
5·5	18·72	166·3	134·2	998·5	1132·7	·0164	67·24	·241	1·595	1·836
6·0	17·71	170·1	138·0	996·2	1134·2	·0165	61·98	·247	1·582	1·829
6·5	16·69	173·6	141·5	994·1	1135·6	·0165	57·50	·253	1·570	1·823
7·0	15·67	176·9	144·8	992·1	1136·9	·0165	53·64	·258	1·559	1·817
7·5	14·65	179·9	147·9	990·2	1138·1	·0165	50·29	·263	1·548	1·811
8·0	13·63	182·9	150·8	988·5	1139·3	·0165	47·34	·267	1·539	1·806
8·5	12·62	185·6	153·6	986·8	1140·4	·0165	44·73	·272	1·529	1·801
9·0	11·60	188·3	156·2	985·2	1141·4	·0166	42·40	·276	1·520	1·796
9·5	10·58	190·8	158·8	983·6	1142·3	·0166	40·31	·280	1·512	1·792
10·0	9·56	193·2	161·2	982·1	1143·3	·0166	38·42	·284	1·504	1·788
11·0	7·53	197·8	165·7	979·3	1145·0	·0166	35·14	·290	1·490	1·780
12·0	5·49	202·0	170·0	976·6	1146·6	·0167	32·40	·297	1·476	1·773
13·0	3·45	205·9	173·9	974·2	1148·1	·0167	30·06	·303	1·464	1·767
14·0	1·42	209·6	177·6	971·9	1149·5	·0167	28·04	·308	1·452	1·761
14·696	0	212·0	180·1	970·3	1150·4	·0167	26·80	·312	1·445	1·757

* Abridged from "*Thermodynamic Properties of Steam*," Keenan & Keyes, 1936.

TABLE 1. PROPERTIES OF SATURATED STEAM—*continued*

Pressure		Temperature	Heat B.Th.U. per pound			Volume Cu. ft. per pound		Entropy B.Th.U. per °F. per pound		
Pounds/ sq. in. absolute	Pounds/ sq. in. gauge	°F.	Sensible liquid	Latent	Total vapour	Liquid	Vapour	Liquid	Evaporation	Total vapour
15	0·3	213·0	181·1	969·7	1150·8	·0167	26·29	·314	1·442	1·755
16	1·3	216·3	184·4	967·6	1152·0	·0167	24·75	·318	1·431	1·750
17	2·3	219·4	187·6	965·5	1153·1	·0168	23·39	·323	1·422	1·745
18	3·3	222·4	190·6	963·6	1154·2	·0168	22·17	·328	1·413	1·740
19	4·3	225·2	193·4	961·9	1155·3	·0168	21·08	·332	1·404	1·736
20	5·3	228·0	196·2	960·1	1156·3	·0168	20·09	·336	1·396	1·732
21	6·3	230·6	198·8	958·4	1157·2	·0169	19·19	·340	1·389	1·728
22	7·3	233·1	201·3	956·8	1158·1	·0169	18·38	·343	1·381	1·724
23	8·3	235·5	203·8	955·2	1159·0	·0169	17·63	·347	1·374	1·721
24	9·3	237·8	206·1	953·7	1159·8	·0169	16·94	·350	1·367	1·717
25	10·3	240·1	208·4	952·1	1160·6	·0169	16·30	·353	1·361	1·714
26	11·3	242·3	210·6	950·7	1161·3	·0169	15·72	·356	1·354	1·711
27	12·3	244·4	212·8	949·3	1162·0	·0170	15·17	·359	1·349	1·708
28	13·3	246·4	214·8	947·9	1162·7	·0170	14·66	·362	1·343	1·705
29	14·3	248·4	216·9	946·5	1163·4	·0170	14·19	·365	1·337	1·702
30	15·3	250·3	218·8	945·3	1164·1	·0170	13·75	·368	1·331	1·699
31	16·3	252·2	220·7	944·0	1164·7	·0170	13·33	·371	1·326	1·697
32	17·3	254·1	222·6	942·8	1165·4	·0170	12·94	·373	1·321	1·694
33	18·3	255·8	224·4	941·6	1166·0	·0171	12·57	·376	1·316	1·692
34	19·3	257·6	226·2	940·3	1166·5	·0171	12·23	·378	1·311	1·689
35	20·3	259·3	227·9	939·2	1167·1	·0171	11·90	·381	1·306	1·687
36	21·3	261·0	229·6	938·0	1167·6	·0171	11·59	·383	1·302	1·685
37	22·3	262·6	231·3	936·9	1168·2	·0171	11·29	·385	1·297	1·683
38	23·3	264·2	232·9	935·8	1168·7	·0171	11·02	·388	1·293	1·681
39	24·3	265·7	234·5	934·7	1169·2	·0171	10·75	·390	1·288	1·678
40	25·3	267·3	236·0	933·7	1169·7	·0172	10·50	·392	1·284	1·676
42	27·3	270·2	239·0	931·6	1170·7	·0172	10·03	·396	1·276	1·672
44	29·3	273·1	242·0	929·6	1171·6	·0172	9·601	·400	1·269	1·669
46	31·3	275·8	244·8	927·7	1172·4	·0172	9·209	·404	1·261	1·665
48	33·3	278·5	247·5	925·8	1173·3	·0173	8·848	·408	1·254	1·662
50	35·3	281·0	250·1	924·0	1174·1	·0173	8·515	·411	1·248	1·659
52	37·3	283·5	252·6	922·2	1174·8	·0173	8·208	·414	1·241	1·655
54	39·3	285·9	255·1	920·5	1175·6	·0173	7·922	·418	1·235	1·652
56	41·3	288·2	257·5	918·8	1176·3	·0173	7·656	·421	1·229	1·649
58	43·3	290·5	259·8	917·1	1176·9	·0174	7·407	·424	1·223	1·647
60	45·3	292·7	262·1	915·5	1177·6	·0174	7·175	·427	1·217	1·644
62	47·3	294·9	264·3	913·9	1178·2	·0174	6·957	·430	1·211	1·641
64	49·3	296·9	266·5	912·3	1178·8	·0174	6·752	·433	1·206	1·639
66	51·3	299·0	268·6	910·8	1179·4	·0174	6·560	·436	1·201	1·636
68	53·3	301·0	270·6	909·4	1180·0	·0175	6·378	·438	1·196	1·634

TABLE 1. PROPERTIES OF SATURATED STEAM—*continued.*

Pressure		Temperature	Heat B.Th.U. per pound			Volume Cu. ft. per pound		Entropy B.Th.U. per °F. per pound		
Pounds/ sq. in. absolute	Pounds/ sq. in. gauge	°F.	Sensible liquid	Latent	Total vapour	Liquid	Vapour	Liquid	Evaporation	Total vapour
70	55·3	302·9	272·6	907·9	1180·6	·0175	6·206	·441	1·191	1·632
75	60·3	307·6	277·4	904·5	1181·9	·0175	5·816	·447	1·179	1·626
80	65·3	312·0	282·0	901·1	1183·1	·0176	5·472	·453	1·168	1·621
85	70·3	316·3	286·4	897·8	1184·2	·0176	5·168	·459	1·157	1·616
90	75·3	320·3	290·6	894·7	1185·3	·0177	4·896	·464	1·147	1·611
100	85·3	327·8	298·4	888·8	1187·2	·0177	4·432	·474	1·129	1·603
110	95·3	334·8	305·7	883·2	1188·9	·0178	4·049	·483	1·112	1·595
120	105·3	341·3	312·4	877·9	1190·4	·0179	3·728	·492	1·096	1·588
130	115·3	347·3	318·8	872·9	1191·7	·0180	3·455	·500	1·082	1·581
140	125·3	353·0	324·8	868·2	1193·0	·0180	3·220	·507	1·068	1·575
150	135·3	358·4	330·5	863·6	1194·1	·0181	3·015	·514	1·056	1·569
160	145·3	363·5	335·9	859·2	1195·1	·0182	2·834	·520	1·044	1·564
170	155·3	368·4	341·1	854·9	1196·0	·0182	2·675	·527	1·032	1·559
180	165·3	373·1	346·0	850·8	1196·9	·0183	2·532	·533	1·022	1·554
190	175·3	377·5	350·8	846·8	1197·6	·0183	2·404	·538	1·012	1·550
200	185·3	381·8	355·4	843·0	1198·4	·0184	2·288	·544	1·002	1·545
220	205·3	389·9	364·0	835·6	1199·6	·0185	2·087	·554	·984	1·537
240	225·3	397·4	372·1	828·5	1200·6	·0186	1·918	·563	·967	1·530
260	245·3	404·4	379·8	821·8	1201·5	·0187	1·775	·572	·951	1·523
280	265·3	411·1	387·0	815·3	1202·3	·0188	1·651	·580	·936	1·516
300	285·3	417·3	393·8	809·0	1202·8	·0189	1·543	·588	·923	1·510
350	335·3	431·7	409·7	794·2	1203·9	·0191	1·326	·606	·891	1·497
400	385·3	444·6	424·0	780·5	1204·5	·0193	1·161	·621	·863	1·484
450	435·3	456·3	437·2	767·4	1204·6	·0195	1·032	·636	·838	1·473
500	485·3	467·0	449·4	755·0	1204·4	·0197	·928	·649	·815	1·463
600	585·3	486·2	471·6	731·6	1203·2	·0201	·770	·672	·773	1·445
700	685·3	503·1	491·5	709·7	1201·2	·0205	·655	·693	·737	1·430
800	785·3	518·2	509·7	688·9	1198·6	·0209	·569	·711	·705	1·415
900	885·3	532·0	526·6	668·8	1195·4	·0212	·501	·728	·674	1·402
1000	985·3	544·6	542·4	649·4	1191·8	·0216	·446	·743	·647	1·390
1100	1085·3	556·3	557·4	630·4	1187·8	·0220	·400	·758	·621	1·378
1200	1185·3	567·2	571·7	611·7	1183·4	·0223	·362	·771	·596	1·367
1300	1285·3	577·5	585·4	593·2	1178·6	·0227	·329	·784	·572	1·356
1400	1385·3	587·1	598·7	574·7	1173·4	·0231	·301	·796	·549	1·345
1500	1485·3	596·2	611·6	556·3	1167·9	·0235	·277	·808	·527	1·335
1600	1585·3	604·9	624·1	538·0	1162·1	·0239	·255	·820	·505	1·325
1800	1785·3	621·0	648·3	501·1	1149·4	·0247	·218	·841	·464	1·305
2000	1985·3	635·8	671·7	463·4	1135·1	·0257	·188	·862	·423	1·285
2500	2485·3	668·1	730·6	360·5	1091·1	·0287	·131	·913	·320	1·232
3000	2985·3	695·4	802·5	217·8	1020·3	·0346	·0858	·973	·189	1·162
3206·2	3191·5	705·4	902·7	0	902·7	·0503	·0503	1·058	0	1·058

TABLE 2. SUPERHEATED STEAM

Pressure lb./sq.in. abs. (Sat. temp. °F.)	Saturation: Liquid. Volume. Total heat. Entropy.	Saturation: Vapour. Volume. Total heat. Entropy.	Volume: Cu. ft. per pound; Total heat: B.Th.U. per pound; Entropy: B.Th.U. per °F. per pound — of superheated steam at temperatures of :—																	
			120° F.	140° F.	160° F.	180° F.	200° F.	220° F.	240° F.	260° F.	280° F.	300° F.	320° F.	340° F.	360° F.	380° F.	400° F.	500° F.	600° F.	700° F.
	0·016	334	345	357	369	381	393	405	417	428	440	452	464	476	488	500	512	572	631	691
1·0	69·7	1106	1114	1123	1132	1141	1150	1160	1169	1178	1187	1196	1205	1214	1223	1233	1242	1288	1336	1384
(102)	0·133	1·978	1·993	2·008	2·023	2·037	2·051	2·065	2·078	2·091	2·103	2·115	2·127	2·139	2·150	2·161	2·172	2·223	2·270	2·314
	0·016	90·6			91·7	94·8	97·8	101	104	107	110	113	116	119	122	125	128	143	158	173
4·0	121	1127			1131	1140	1149	1159	1168	1177	1186	1195	1204	1214	1223	1232	1241	1288	1336	1384
(153)	0·220	1·863			1·868	1·883	1·897	1·911	1·924	1·937	1·950	1·962	1·974	1·985	1·997	2·008	2·019	2·07	2·117	2·161
	0·016	53·6				53·9	55·7	57·5	59·2	60·9	62·7	64·4	66·1	67·8	69·6	71·3	73·0	81·6	90·1	98·6
7·0	145	1137				1138	1148	1157	1167	1176	1185	1195	1204	1213	1222	1232	1241	1288	1335	1384
(177)	0·258	1·817				1·819	1·834	1·848	1·861	1·874	1·887	1·900	1·912	1·923	1·935	1·946	1·957	2·008	2·056	2·099
	0·017	38·4					38·9	40·1	41·3	42·6	43·8	45·0	46·2	47·4	48·6	49·8	51·0	57·1	63·0	69·0
10·0	161	1143					1147	1156	1166	1175	1185	1194	1203	1213	1222	1231	1241	1288	1335	1383
(193)	0·284	1·788					1·793	1·807	1·821	1·834	1·847	1·860	1·872	1·883	1·895	1·906	1·917	1·969	2·016	2·060
	0·017	26·8						27·2	28·0	28·9	29·7	30·5	31·4	32·2	33·0	33·9	34·7	38·8	42·9	46·9
14·696	180	1150						1154	1164	1174	1183	1193	1202	1212	1221	1231	1240	1287	1335	1383
(212)	0·312	1·757						1·762	1·777	1·790	1·803	1·816	1·828	1·840	1·852	1·863	1·874	1·926	1·973	2·017
	0·017	20·1							20·5	21·1	21·7	22·4	23·0	23·6	24·2	24·8	25·4	28·5	31·5	34·5
20·0	196	1156							1162	1172	1182	1192	1201	1211	1220	1230	1239	1287	1334	1383
(228)	0·336	1·732							1·741	1·755	1·768	1·781	1·793	1·805	1·817	1·829	1·840	1·892	1·939	1·983
	0·017	16·30								16·82	17·33	17·84	18·34	18·83	19·33	19·82	20·3	22·7	25·2	27·6
25·0	208	1161								1171	1181	1191	1200	1210	1219	1229	1239	1286	1334	1383
(240)	0·353	1·714								1·728	1·742	1·755	1·768	1·780	1·792	1·803	1·814	1·867	1·914	1·958
	0·017	13·75								13·96	14·39	14·82	15·24	15·66	16·07	16·49	16·90	18·93	21·0	23·0
30·0	219	1164								1169	1179	1189	1199	1209	1219	1228	1238	1286	1334	1382
(250)	0·368	1·699								1·706	1·720	1·734	1·746	1·759	1·771	1·782	1·794	1·846	1·894	1·938
	0·017	11·90								11·91	12·29	12·66	13·03	13·39	13·75	14·10	14·46	16·21	17·94	19·67
35·0	228	1167								1168	1178	1188	1198	1208	1218	1228	1237	1285	1334	1382
(259)	0·381	1·687								1·688	1·702	1·715	1·728	1·741	1·753	1·765	1·776	1·829	1·877	1·921
	0·017	10·50									10·71	11·04	11·36	11·68	12·00	12·32	12·63	14·17	15·69	17·20
40·0	236	1170									1177	1187	1197	1207	1217	1227	1237	1285	1333	1382
(267)	0·392	1·676									1·686	1·699	1·713	1·725	1·738	1·749	1·761	1·814	1·862	1·906

TABLE 2. SUPERHEATED STEAM—*continued*

Pressure lb./sq.in. abs. (Sat. temp. °F.)	Saturation		Volume : Cu. ft. per pound Total heat : B.Th.U. per pound Entropy : B.Th.U. per °F. per pound } of superheated steam at temperatures of :—																
	Liquid. Volume. Total heat. Entropy.	Vapour. Volume. Total heat. Entropy.	320° F.	340° F.	360° F.	380° F.	400° F.	420° F.	440° F.	460° F.	480° F.	500° F.	520° F.	540° F.	560° F.	580° F.	600° F.	700 °F.	800° F.	900° F
	0·017	8·52	9·04	9·30	9·56	9·81	10·07	10·32	10·57	10·82	11·06	11·31	11·56	11·80	12·04	12·29	12·53	13·74	14·95	16·15
50	250	1174	1195	1205	1215	1225	1235	1245	1255	1265	1274	1284	1294	1303	1313	1323	1333	1381	1431	1481
(281)	0·411	1·659	1·686	1·699	1·711	1·723	1·735	1·746	1·757	1·768	1·778	1·789	1·799	1·809	1·818	1·828	1·837	1·881	1·922	1·960
	0·017	7·18	7·49	7·71	7·93	8·14	8·36	8·57	8·78	8·99	9·20	9·40	9·61	9·81	10·02	10·22	10·43	11·44	12·45	13·45
60	262	1178	1193	1203	1213	1224	1234	1244	1254	1263	1273	1283	1293	1303	1312	1322	1332	1381	1431	1481
(293)	0·427	1·644	1·663	1·677	1·689	1·702	1·714	1·725	1·736	1·747	1·758	1·768	1·778	1·788	1·798	1·807	1·816	1·861	1·902	1·940
	0·017	6·21	6·38	6·57	6·76	6·95	7·14	7·32	7·50	7·68	7·86	8·04	8·22	8·40	8·57	8·75	8·92	9·80	10·66	11·52
70	273	1181	1190	1201	1212	1222	1232	1242	1252	1262	1272	1282	1292	1302	1312	1321	1331	1380	1430	1481
(303)	0·441	1·632	1·644	1·658	1·671	1·683	1·695	1·707	1·718	1·729	1·740	1·750	1·760	1·770	1·780	1·789	1·799	1·843	1·884	1·923
	0·018	5·47	5·54	5·72	5·89	6·06	6·22	6·38	6·54	6·70	6·86	7·02	7·18	7·33	7·49	7·64	7·80	8·56	9·32	10·08
80	282	1183	1188	1199	1210	1220	1231	1241	1251	1261	1271	1281	1291	1301	1311	1321	1331	1380	1430	1480
(312)	0·453	1·621	1·627	1·641	1·654	1·667	1·679	1·691	1·702	1·713	1·724	1·735	1·745	1·755	1·765	1·774	1·784	1·828	1·869	1·908
	0·018	4·90		5·05	5·21	5·36	5·51	5·65	5·80	5·94	6·08	6·23	6·37	6·51	6·64	6·78	6·92	7·60	8·28	8·95
90	291	1185		1197	1208	1219	1229	1240	1250	1260	1270	1280	1290	1300	1310	1320	1330	1379	1429	1480
(320)	0·464	1·611		1·626	1·640	1·652	1·665	1·677	1·688	1·700	1·710	1·721	1·731	1·741	1·751	1·761	1·770	1·815	1·856	1·895
	0·018	4·43		4·52	4·66	4·80	4·94	5·07	5·20	5·33	5·46	5·59	5·72	5·84	5·97	6·09	6·22	6·84	7·45	8·05
100	298	1187		1194	1206	1217	1228	1238	1249	1259	1269	1279	1289	1299	1309	1319	1329	1379	1429	1480
(328)	0·474	1·603		1·612	1·626	1·639	1·652	1·664	1·676	1·687	1·698	1·709	1·719	1·729	1·739	1·749	1·758	1·803	1·844	1·883
	0·018	3·73			3·84	3·96	4·08	4·20	4·31	4·42	4·53	4·64	4·74	4·85	4·96	5·06	5·17	5·68	6·20	6·70
120	312	1190			1202	1213	1224	1235	1246	1257	1267	1277	1287	1298	1308	1318	1328	1378	1428	1479
(341)	0·492	1·588			1·602	1·616	1·629	1·641	1·653	1·665	1·676	1·687	1·697	1·708	1·718	1·727	1·737	1·782	1·824	1·863
	0·018	3·22			3·26	3·37	3·47	3·57	3·67	3·76	3·86	3·95	4·05	4·14	4·23	4·32	4·41	4·86	5·30	5·74
140	325	1193			1197	1209	1221	1232	1243	1254	1265	1275	1286	1296	1306	1316	1326	1377	1427	1478
(353)	0·507	1·575			1·580	1·595	1·609	1·622	1·634	1·646	1·657	1·668	1·679	1·689	1·700	1·709	1·719	1·765	1·806	1·845
	0·018	2·83				2·91	3·01	3·10	3·19	3·27	3·36	3·44	3·53	3·61	3·69	3·77	3·85	4·24	4·63	5·02
160	336	1195				1206	1218	1229	1241	1252	1262	1273	1284	1294	1304	1315	1325	1376	1426	1478
(364)	0·520	1·564				1·577	1·591	1·604	1·617	1·629	1·641	1·652	1·663	1·673	1·684	1·694	1·703	1·749	1·791	1·830
	0·018	2·53				2·56	2·65	2·73	2·81	2·89	2·97	3·04	3·12	3·19	3·27	3·34	3·41	3·76	4·11	4·45
180	346	1197				1201	1214	1226	1238	1249	1260	1271	1282	1292	1303	1313	1324	1375	1426	1477
(373)	0·533	1·554				1·560	1·575	1·588	1·602	1·614	1·626	1·637	1·648	1·659	1·669	1·680	1·689	1·736	1·778	1·817

TABLE 2. SUPERHEATED STEAM—*continued*

Pressure lb./sq.in. abs. (Sat. temp. °F.)	Saturation		Volume : Cu. ft. per pound Total heat : B.Th.U. per pound Entropy : B.Th.U. per °F. per pound } of superheated steam at temperatures of :—																
	Liquid. Volume. Total heat. Entropy.	Vapour. Volume. Total heat. Entropy.	400° F.	420° F.	440° F.	460° F.	480° F.	500° F.	520° F.	540° F.	560° F.	580° F.	600° F.	620° F.	640° F.	660° F.	680° F.	700° F.	800° F.	900° F.
	0·018	2·29	2·36	2·44	2·51	2·59	2·66	2·73	2·79	2·86	2·93	2·99	3·06	3·12	3·19	3·25	3·32	3·38	3·69	4·00
200	355	1198	1210	1223	1235	1247	1258	1269	1280	1291	1301	1312	1322	1333	1343	1353	1363	1374	1425	1476
(382)	0·544	1·545	1·559	1·574	1·587	1·600	1·612	1·624	1·635	1·646	1·657	1·667	1·677	1·686	1·696	1·705	1·714	1·723	1·766	1·805
	0·019	2·09	2·13	2·20	2·27	2·34	2·40	2·47	2·53	2·59	2·65	2·71	2·77	2·83	2·89	2·95	3·01	3·07	3·35	3·63
220	364	1200	1207	1220	1232	1244	1255	1267	1278	1289	1300	1310	1321	1331	1342	1352	1362	1373	1424	1476
(390)	0·554	1·537	1·545	1·560	1·574	1·587	1·600	1·612	1·623	1·634	1·645	1·655	1·665	1·675	1·685	1·694	1·703	1·712	1·755	1·794
	0·019	1·918	1·928	1·996	2·06	2·13	2·19	2·25	2·31	2·36	2·42	2·48	2·53	2·59	2·64	2·70	2·75	2·80	3·07	3·33
240	372	1201	1203	1216	1229	1241	1253	1265	1276	1287	1298	1309	1319	1330	1340	1351	1361	1372	1423	1475
(397)	0·563	1·530	1·532	1·548	1·562	1·575	1·588	1·600	1·612	1·623	1·634	1·644	1·655	1·664	1·674	1·684	1·693	1·702	1·744	1·784
	0·019	1·775		1·826	1·888	1·948	2·01	2·06	2·12	2·17	2·23	2·28	2·33	2·38	2·43	2·48	2·53	2·58	2·83	3·07
260	380	1202		1212	1226	1238	1251	1262	1274	1285	1296	1307	1318	1328	1339	1350	1360	1370	1422	1474
(404)	0·572	1·523		1·535	1·550	1·564	1·577	1·590	1·602	1·613	1·624	1·634	1·645	1·655	1·664	1·647	1·683	1·692	1·735	1·775
	0·019	1·651		1·679	1·739	1·796	1·851	1·905	1·957	2·01	2·06	2·11	2·16	2·20	2·25	2·30	2·35	2·39	2·62	2·85
280	387	1202		1209	1222	1235	1248	1260	1272	1283	1294	1305	1316	1327	1338	1348	1359	1369	1422	1474
(411)	0·580	1·516		1·524	1·539	1·554	1·567	1·580	1·592	1·603	1·614	1·625	1·635	1·646	1·655	1·665	1·674	1·683	1·727	1·766
	0·019	1·543		1·551	1·609	1·664	1·717	1·768	1·817	1·865	1·913	1·959	2·01	2·05	2·10	2·14	2·18	2·23	2·44	2·65
300	394	1203		1205	1219	1233	1245	1258	1270	1281	1293	1304	1315	1326	1336	1347	1358	1368	1421	1473
(417)	0·588	1·510		1·513	1·529	1·543	1·557	1·570	1·582	1·594	1·605	1·616	1·627	1·637	1·647	1·657	1·666	1·675	1·718	1·758
	0·019	1·326			1·348	1·398	1·446	1·492	1·537	1·580	1·622	1·663	1·704	1·743	1·783	1·822	1·860	1·898	2·08	2·27
350	410	1204			1210	1225	1239	1252	1264	1276	1288	1300	1311	1322	1333	1344	1355	1366	1419	1471
(432)	0·606	1·497			1·504	1·520	1·534	1·548	1·561	1·573	1·585	1·596	1·607	1·617	1·628	1·637	1·647	1·656	1·700	1·740
	0·019	1·161				1·198	1·243	1·285	1·326	1·365	1·403	1·441	1·477	1·513	1·548	1·583	1·617	1·651	1·816	1·977
400	424	1205				1217	1231	1245	1258	1271	1283	1295	1307	1318	1330	1341	1352	1363	1416	1469
(445)	0·621	1·484				1·498	1·514	1·528	1·542	1·555	1·567	1·578	1·589	1·600	1·611	1·621	1·630	1·640	1·684	1·725
	0·020	1·032				1·040	1·083	1·123	1·161	1·198	1·233	1·267	1·301	1·333	1·365	1·397	1·428	1·458	1·607	1·752
450	437	1205				1208	1224	1238	1252	1266	1278	1291	1303	1315	1326	1338	1349	1360	1414	1468
(456)	0·636	1·473				1·477	1·494	1·510	1·524	1·537	1·550	1·562	1·574	1·585	1·595	1·605	1·615	1·625	1·670	1·711
	0·020	0·928					0·954	0·993	1·029	1·063	1·096	1·128	1·159	1·189	1·219	1·248	1·276	1·304	1·441	1·572
500	449	1204					1215	1231	1246	1260	1273	1286	1299	1311	1323	1334	1346	1357	1412	1466
(467)	0·649	1·463					1·475	1·492	1·507	1·521	1·535	1·547	1·559	1·570	1·581	1·592	1·602	1·612	1·657	1·698

TABLE 2. SUPERHEATED STEAM—*continued*

Pressure lb./sq.in. abs. (Sat. temp. °F.)	Saturation		Volume : Cu. ft. per pound Total heat : B.Th.U. per pound Entropy : B.Th.U. per °F. per pound } of superheated steam at temperatures of :—																	
	Liquid. Volume. Total heat. Entropy.	Vapour. Volume. Total heat. Entropy.	500° F.	520° F.	540° F.	560° F.	580° F.	600° F.	620° F.	640° F.	660° F.	680° F.	700° F.	720° F.	740° F.	760° F.	780° F.	800° F.	900° F.	1000° F
	0·020	0·770	0·795	0·829	0·860	0·890	0·919	0·946	0·973	0·999	1·024	1·049	1·073	1·097	1·121	1·144	1·167	1·190	1·301	1·410
600	472	1203	1216	1233	1248	1263	1277	1290	1303	1315	1327	1339	1351	1363	1374	1385	1397	1408	1463	1517
(486)	0·672	1·445	1·459	1·476	1·492	1·506	1·520	1·532	1·544	1·556	1·567	1·577	1·588	1·597	1·607	1·616	1·625	1·634	1·676	1·715
	0·021	0·655		0·683	0·713	0·742	0·768	0·793	0·818	0·841	0·864	0·886	0·908	0·929	0·950	0·970	0·991	1·011	1·108	1·202
700	492	1201		1218	1235	1251	1266	1281	1294	1308	1320	1333	1345	1357	1369	1380	1392	1403	1459	1514
(503)	0·693	1·430		1·446	1·464	1·480	1·495	1·508	1·521	1·533	1·545	1·556	1·567	1·577	1·587	1·596	1·606	1·615	1·657	1·696
	0·021	0·569		0·572	0·602	0·629	0·654	0·678	0·701	0·722	0·743	0·764	0·783	0·803	0·822	0·840	0·858	0·876	0·963	1·047
800	510	1199		1201	1221	1239	1255	1271	1285	1299	1313	1326	1339	1351	1363	1375	1387	1399	1455	1511
(518)	0·711	1·415		1·417	1·438	1·455	1·472	1·486	1·500	1·513	1·525	1·537	1·548	1·558	1·568	1·578	1·588	1·597	1·641	1·680
	0·021	0·501			0·512	0·540	0·564	0·587	0·609	0·629	0·649	0·668	·0686	0·704	0·722	0·739	0·755	0·772	0·851	0·926
900	527	1195			1204	1225	1243	1260	1276	1291	1305	1319	1332	1345	1358	1370	1382	1394	1452	1508
(532)	0·728	1·402			1·411	1·431	1·449	1·465	1·480	1·494	1·507	1·519	1·530	1·541	1·552	1·562	1·572	1·581	1·626	1·666
	0·022	0·446				0·467	0·492	0·514	0·535	0·555	0·573	0·591	0·608	0·625	0·641	0·657	0·673	0·688	0·760	0·829
1000	542	1192				1210	1230	1249	1266	1282	1297	1311	1325	1339	1352	1364	1377	1389	1448	1505
(545)	0·743	1·390				1·407	1·427	1·445	1·461	1·476	1·489	1·502	1·514	1·526	1·537	1·547	1·557	1·567	1·612	1·653
	0·023	0·345					0·355	0·379	0·399	0·418	0·435	0·452	0·467	0·482	0·496	0·510	0·524	0·537	0·598	0·655
1250	579	1181					1192	1217	1238	1257	1275	1292	1307	1322	1337	1350	1364	1377	1439	1498
(572)	0·778	1·361					1·372	1·395	1·415	1·433	1·449	1·464	1·477	1·490	1·502	1·514	1·524	1·535	1·582	1·624
	0·024	0·277						0·282	0·305	0·324	0·341	0·357	0·372	0·386	0·399	0·411	0·424	0·435	0·489	0·539
1500	612	1168						1175	1204	1228	1250	1269	1287	1304	1320	1335	1350	1364	1429	1490
(596)	0·808	1·335						1·341	1·369	1·391	1·411	1·428	1·443	1·458	1·471	1·484	1·496	1·507	1·557	1·600
	0·026	0·188								0·194	0·216	0·234	0·249	0·262	0·275	0·286	0·297	0·307	0·353	0·394
2000	672	1135								1146	1185	1215	1240	1262	1283	1301	1319	1336	1409	1475
(636)	0·862	1·285								1·295	1·330	1·356	1·378	1·397	1·415	1·430	1·444	1·458	1·514	1·560
	0·035	0·086											0·098	0·125	0·142	0·155	0·166	0·176	0·216	0·248
3000	803	1020											1061	1136	1180	1214	1242	1267	1365	1442
(695)	0·973	1·162											1·197	1·261	1·298	1·326	1·349	1·369	1·444	1·498
	0·050	0·050												0·102	0·122	0·136	0·148	0·158	0·198	0·229
3206·2	903	903												1091	1150	1191	1223	1251	1355	1435
(705)	1·058	1·058												1·219	1·269	1·302	1·329	1·351	1·431	1·487

APPENDIX III

ENTROPY

ENTROPY is a function which is very useful for certain steam calculations. Unfortunately while it is easy to use, its exact meaning may be difficult to appreciate.

When heat is added to a substance there must be a temperature drop between the heating medium and the heated substance. Although the conception may appear to be unscientific, in a high temperature substance the heat may be regarded as " concentrated." In a substance at low temperature the heat can be considered as " diluted " or " spread."

Wherever heat is added to a fluid there is a decrease in the concentration of the heat in the heated substance, or an increase in the spread or dilution of the heat energy.

For example, when heat is added to water at boiling point the energy addition does not cause any rise of temperature, therefore the concentration of heat continually diminishes, or the " dilution " or " spread " of the heat continually increases.

The measure of this dilution or spread of the heat energy is called " Entropy " and its numerical value is :—

$$\frac{\text{Quantity of heat added}}{\text{Mean absolute temperature during the heat addition}}$$

When heat is removed from a substance there is a decrease in the spread, or dilution, of the heat energy in the cooled substance. Similarly there is an increase in the spread or dilution of the heat energy in the substance which has received the heat.

If some of the heat energy in a substance is converted by expansion into mechanical energy there is no equivalent to a temperature drop between the heat and the mechanical work—the actual conversion is perfect—so that the concentration, dilution or spread of the heat remains unchanged—the entropy remains constant. Similarly when mechanical compression adds energy to a substance the conversion is perfect and the temperature rises without any dilution or spread, that is at constant entropy.

A perfect mechanical compression or expansion is called an " Adiabatic " (" not-passing-through ") compression or expansion because no heat passes through the walls of the cylinder or vessel into or out of the substance during the process.

Thus, expressed in more concise language, when heat enters a body the entropy of the heat energy in the body increases ; when heat leaves a body the entropy of the heat energy remaining in the body decreases. When adiabatic changes take place the entropy of the fluid remains constant.

Water at 32° F. is considered for purposes of calculation as having zero heat content, and it has, therefore, an entropy of zero. If the water is then heated to 212° F., it receives 180 B.Th.U., at a mean absolute temperature of

$$\frac{492 + 672}{2}$$

or 582° absolute. Its entropy is $\frac{180}{582}$ or 0·309. This is the entropy of the liquid. Actually the entropy of the liquid is 0·312. The heat was added in infinitely small increments and caused minute temperature increments. Without using

the calculus a near approximation can be made by taking 10 increments of 18 B.Th.U. each, giving temperature increases of 18° F. The calculation is as follows :—

Mean temperature °F.	41	59	77	95	113	131	149	167	185	203
Mean absolute temperature	501	519	537	555	573	591	609	627	645	663
Entropy increase ..	·0359	·0347	·0335	·0324	·0314	·0305	·0296	·0287	·0279	·0272

The sum of these ten entropy increases is 0·3118. This figure is larger than the figure of 0·309 obtained by simply using the mean absolute temperature over the whole temperature rise. Had even smaller heat additions been taken the figure would have become 0·312.

These entropy increases for equal heat additions show quite clearly that the higher the temperature at which the heat addition is made the more concentrated is the added heat energy—the smaller its dilution, spread or entropy.

If the water is then converted into steam at 212° F. the temperature remains constant (Chapter VII) and 970 B.Th.U. are added to it. The entropy of evaporation is therefore $\frac{970}{672}$, i.e. 1·444.

The total entropy is 1·444 + 0·312 = 1·756.

Slight divergencies between these figures and those in the steam table are due to approximations in the last place of decimals. For a full discussion of entropy and its use, readers are referred to works on thermodynamics. Some illustrative examples showing the use of the entropy columns of the steam tables forming Appendix II are here given.

EXAMPLE 1

To calculate the dryness and heat drop of steam expanded adiabatically from a higher pressure to a lower pressure. Assume that the steam be expanded from 100 lb. per square inch absolute to 14·7 lb. per square inch absolute in a perfect engine.

From the tables,

Initial pressure	100 lb.	Final pressure	14·7 lb.
Total heat ..	1,187 B.Th.U.	Latent heat ..	970 B.Th.U.
Total entropy ..	1·603	Total entropy ..	1·757
		Evaporation entropy ..	1·445

The entropy difference is 1·757 − 1·603 = 0·154.

This difference represents the wetness in the steam at exhaust.

The entropy of evaporation is 1·445.

$$\text{The wetness} = \frac{0{\cdot}154}{1{\cdot}445} = 0{\cdot}1065 \text{ water}$$

$$= 10{\cdot}65 \text{ per cent.,}$$

Therefore dryness $= 89{\cdot}35$ „ „

The total heat remaining after expansion is made up of the sensible heat of 1 lb. of water at 212° F. plus the latent heat in (1 − 0·1065) lb. steam.

Therefore total heat $= 180 + (1 - 0{\cdot}1065)\ 970$

$= 1{,}047$ B.Th.U.

Adiabatic heat drop $= 1{,}187 - 1047 = 140$ B.Th.U.

EXAMPLE 2

To determine the extent to which steam must be superheated if it is to be dry at exhaust.

Let the steam be initially at 100 lb. per square inch absolute dry, and as in Example 1 let it be expanded to atmospheric pressure in a perfect engine. Total entropy of dry, saturated steam at 14·7 lb. absolute = 1·757. This must be the entropy of the saturated steam if it is to be dry. From inspection of superheat tables :—

At 100 lb. per square inch and 600° F., entropy = 1·758
At „ „ „ and 580° F., „ = 1·749

From these figures 20° F. corresponds to 0·009 entropy. By simple proportion, the excess of steam temperature over 580° F. is given by

$$\frac{20 \times (1{\cdot}757 - 1{\cdot}749)}{0{\cdot}009} \text{ i.e. } 18° \text{ F.}$$

Therefore, since entropy remains constant for adiabatic expansion, 598 °F. must be the temperature to which the steam must be superheated.

EXAMPLE 3

To find the power available from steam under given conditions.

An exhaust turbine takes steam exhausted from a reciprocating engine at atmospheric pressure. The engine exhaust is 4 per cent. wet, and the amount of steam is 20,000 lb. per hour. What power can be expected from the turbine when exhausting into a 28 inch vacuum ?

Conditions at engine exhaust and turbine inlet—atmospheric pressure—are found in the steam table :—

Liquid entropy	0·312
Evaporation entropy ..	1·445
Sensible heat	180 B.Th.U./lb.
Latent heat	970 B.Th.U./lb.

As exhaust is 4 per cent. wet, the entropy at engine exhaust is

$$0{\cdot}312 + 96 \text{ per cent. of } 1{\cdot}445$$
$$= 0{\cdot}312 + 1{\cdot}387 = 1{\cdot}699.$$

Total heat in engine exhaust is

$$180 + 96 \text{ per cent. of } 970$$
$$= 180 + 931 = 1{,}111 \text{ B.Th.U./lb.}$$

Steam table gives the following for steam at 28 inches vacuum :—

Total entropy	1·983
Evaporation entropy	1·854
Sensible heat	68 B.Th.U./lb.
Latent heat	1,037 B.Th.U./lb.

Wetness of turbine exhaust (by method of Example 1)

$$= \frac{1{\cdot}983 - 1{\cdot}699}{1{\cdot}854} = \frac{0{\cdot}284}{1{\cdot}854} = 15{\cdot}31 \text{ per cent. wet.}$$

Adiabatic heat drop in turbine (by method of Example 1) :—

Total heat in turbine exhaust = 68 + (84·69 per cent. of 1,037)
= 946 B.Th.U./lb.
Adiabatic heat drop = 1,111 − 946 = 165 B.Th.U./lb.

Theoretical power available :—

Heat units available per hour = 20,000 × 165 B.Th.U.
Energy available per hour = 20,000 × 165 × 778 ft.-lb.
1 H.P. = 33,000 ft.-lb./min., or 1,980,000 ft.-lb./hr.

$$\text{Therefore theoretical H.P.} = \frac{20{,}000 \times 165 \times 778}{1{,}980{,}000} = 1{,}297 \text{ H.P.}$$

In practice losses occur and the efficiency of the turbine must be taken into account ; the figure in practice would be about 70 per cent. of this, the real effective heat drop being about 70 per cent. of 165 B.Th.U., or 120 B.Th.U.

The effect of taking the efficiency of the machine into account is that the total heat in the exhaust would be 1,111 — 120 or 991 B.Th.U. instead of 946 B.Th.U., and hence instead of calculated wetness of the exhaust of 15·3 per cent. it will, in practice, be only 11 per cent., i.e.

$$\text{Dryness of steam} = \frac{991 - 68}{1{,}037} = 89 \text{ per cent.}$$

INDEX

ACCUMULATORS, 602
Air :
  Detection of leaks, 102
  Distribution in regenerators, 455
  Excess air, 63, 84
  Films in steam heating, 562–568
  Flow through fuel beds, 83
  Heaters, 151
  Inleakage effect of, 100–102, 229, 230, 490–492
  Preheating, 29, 30, 75, 412, 463, 705
  Primary air, 81
  Requirements for combustion—*see* Combustion.
  Secondary air, 82
Air leaks—*see* Air.
Analysis of solid fuel, 776
Annealing furnaces :
  Glass, 449, 452
  Metals, etc., 471–478
Arc furnaces—*see* Electric furnaces.
Ashes :
  Loss of combustibles in, 248–250
  Recovery of combustible from, 249

BARNES' Formula, 225, 226
Bernouilli's Theorem, 167, 168, 173
Blast furnace gas, 30
Blow-down, 245, 332–334
Boiler auxiliaries, 297–324
  Air preheaters, 312
  Economisers, 305
  Fans, 322
  Feed pumps, 315
  Feed water heaters, 310
  Lancing, 299
  Mountings and fittings, 297
  Power requirements, 313
  Soot removal, 298
  Steam jets, 321
  Superheaters, 299
Boiler mountings and fittings, 297, 619
Boiler pressure, selection of, 110
Boilers :
  Air inleakage, 100–102, 229
  Banking fires, 245
  Carbon losses, 247–250
  Central heating, 616–619, 625, 626
  Cornish, 282
  Design, 273
  Economic, 285–288
  Firing, 233–244, 625
  Heat balance, 341, 349–352
  Heat flow diagram, 712
  Heat transmission in, 120, 160–163, 276, 277, 283, 289, 295
  Lancashire, 281–284
  Oil-fired, 291, 707
  Stationary locomotive, 279
  Testing, 340–354
  Vertical, 274–279
  Waste heat, 288, 486–498
  Water—*see* Feed water.
  Water circulation, 138
  Water level, 245
Boilers—*continued.*
  Water-tube, 289–296
Boiler scale :
  Effects of, 120, 126, 326
  prevention, 327
Boiler settings : leakages, 100–102
Bunsen burners, 412, 678
Burners :
  Fuel oil, 702
  Furnace, 412–415
  Gas—*see* Town gas supply.
  Pulverised fuel, 257

CANNEL, 6
Carbon : calorific value, 71
Carbonisation, 23
Caustic embrittlement, 331
Cement kilns, 433, 714, 717
Central heating, 606–630. *See also* Space heating.
Channelling in gas producer firebeds, 364
Chimney :
  Dimension, 81, 102
  Pressure loss in, 101
Clarain, 6
Clinker :
  Formation, 94–96, 249
  In gas producers, 363
  Prevention, 97–100
CO_2 indicators, 720–726
Coal :
  Analysis, 776
  Ash, 3, 4, 17, 777
  Caking properties, 4, 5, 15, 44
  Calorific value, 5, 7, 9, 44, 340, 342, 344
  Classification, 7–10
  Cleaning, 18–21
  Composition, 3, 4 ; industrial significance, 14–18
  Draught required, 103
  Flue gas losses, 67
  Fuel Research Survey, 788
  Industrial uses, 13
  Measurement, 202
  Moisture, 14
  Moisture, effect on density, 202, 203
  Open cast—*see* Open cast coal.
  Origin, 1
  Oxidation, 40
  Properties, 14–18, 776–778
  Reserves, 1
  Sampling, 747–756
  Screen analysis, 755
  Selection for industrial purposes, 13, 14, 779–786
  Size :
    As marketed, 21
    Effect of weathering, 45
    Effect on properties, 17
    Effect on use, 22
  Storage, 39–48
  Sulphur, 4, 18
  Types, 5
  Use in furnaces, 405

Coal—*continued.*
Volatile matter, 4, 15, 777
Weighing, 202–204
Coalfields, 10–13, 788
Coal gas :
Calorific value, 28
Composition, 28, 30
Flue gas heat losses, 67–70
Heat flow diagram for gas retorts, 713
Manufacture, 23, 27
Preheating, 29, 393
Selection for industrial purposes, 786
Sulphur content, 29
Transmission of, 178, 179, 200
Use in dryers, 655
Use in furnaces, 405
Use in glass manufacture, 454
Coal tar fuels, 32–37
Application, 36
Calorific values, 35
Composition, 35
Manufacture, 32
Nomenclature, 32, 34
Pulverised pitch, 37
Coke :
Combustion, 91, 143
Combustion rate, 27
Composition, 24
Crucible furnace, 427
Density, 26, 27, 203
Impurities, 26
Radiation, 143
Reactivity, 25, 26
Sampling, 756
Shatter test, 24
Size, 25
Sulphur content, 26
Use in dryers, 657
Use in furnaces, 405
Coke oven gas—*see* Coal gas.
Combustion, 49–57, 90–96
Air requirements, 54, 55, 63
Calculations, 50–54
Chemistry, 49
Data, 54, 71
Excess air, 63, 64
Temperature, 89
Thermal effects, 70–72
Time factor, 87
Turbulence, 87
Condensate, 116, 317–320, 571, 579
Condensers, 151
Conduction of heat, 119, 122–127, 417. *See also* Heat transfer.
Conductivity, 122–124
Convection, 119, 128–139. *See also* Heat transfer.
Conveyor furnaces, 475, 476
Cornish boilers, 280, 282
Creosote, 33
Crosstube boilers, 274, 276
Crucible furnaces, 427
Cupola furnaces, 428

D'Arcy Formula, 134
De-aeration of boiler feed water, 331
Demineralisation process for feed water treatment, 329
De-superheating, 562
Diffusivity, 481
Distillation, 115, 603
Draught, 77–81
Calculation of, 79
Furnaces, 387
Management of, 230–232, 622
Measurement, 231, 232, 726–729
Primary air, 81
Production of, 79, 321–324
Secondary air, 82
Utilisation of, 100
Draught gauge, 230, 232, 726
Drying :
Control of, 648
Data for calculations, 650–654
Dust collection, 646
Furnaces and fuels, 655–660
Heat losses, 654
Practice, 633–635, 646
Principles, 631–633
Thermal efficiency, 648
Types of dryers, 635–646
Vapour pressure of water, 631, 632, 651
Durain, 6

Economic boilers, 285
Economisers :
Design, 309
Economics, 306
Efficiency and draught loss, 164
Heat transmission, 151, 163, 308
Electric furnaces :
Arc furnaces, 505, 506, 508, 510
Arc heating, 503
H.F. induction furnace, 505, 506, 508, 511
Induction heating, 504
L.F. induction furnace, 508, 509, 511, 512
Resistance furnace, 507, 508, 509, 512, 513
Resistance heating, 504, 512
Supply of current, 513
Theory, 499–503
Electricity :
For furnace heating, 37, 513
Selection for industrial heating, 787
Entropy, 797
Evaporation, 112, 559, 631–635
Exhaust steam utilisation, 320, 554, 581, 584–594

Fan draught, 322
Fanning's equation, 173, 174, 187
Feed water :
Aluminate treatment, 328
Circulation in boilers, 138, 282
Corrosive action, 331
Deaeration, 331
Demineralisation, 329
Hardness, 325
Heaters, 310
Heating by means of exhaust steam, 314, 321
High pressure boilers, 330
Internal treatment, 320
Measurement, 224–226
Phosphate treatment, 329
Pumping, 315–320

Feed water—*continued.*
Sampling boiler water, 338
Scale in boilers, effect of, 120, 121, 126, 326, 327 ; prevention of, 327
Specifications, 336, 337
Temperature, 306, 319
Testing, 335–339
Treatment, 327–331
Waste heat boilers, 492
Firebars, 246
Firebed—*see* Fuel bed.
Firebricks—*see* Refractory materials.
Firebridge, 89, 103
Flame temperature, 74
Flash steam, 117, 581
Flash tanks, 584
Flow meters, 204–224
Flue gases :
Analysis of, 719–726
Composition, calculation of :
From coal and coke, 52–54
From coal gas, 54–56
From oil fuel, 705
Furnaces : heat recovery, 392–396
Heat losses in, 64–70, 85–87, 417
Heat recoverable from, 487, 488
Latent heat, 63
Temperature measurement, 736
Fluid flow, 166–201
Changes of section, 176
Changes of direction, 176
Convection heating, 169
Equations of, 173–176
Examples, 198–201
Flow outside tubes, 177
Furnaces, calculation of flow in, 185–197
Gas flow, 178
Measurement of, 179–182
Orifice plate, 179, 204–224
Pitot tube, 181
Venturi tube, 180
Steam flow, 182, 212–224
Stratification, 182
Stream line flow, 168,
Turbulent flow, 168
Viscosity, 170
Friction : fluid flow, 173
Fuel :
Conservation of, 1
Manufacture, 23
Selection of, 377, 779–787
Fuel bed :
Air flow, 83, 232
Cooling, 96–100
Physical characteristics, 89
Thickness, 18, 234
Zones, 82
Fuel oil—*see* Oil fuel.
Furnace atmospheres, 377–383
Furnaces :
Accessories, 411–416
Brickwork construction, 548–551
Combustion conditions, 383–389
Control of, 478–480
Gas velocity, 387
Heat balance, 391, 403
Heat losses, sources of, 389, 391-403, 417-419
Heating by oil, 707
Furnaces—*continued.*
Insulation, 396–401
Loading, 406, 463
Operational methods, 404–406
Preheating, 459–471
Selection 376
Structural features, 407–411
Testing, 416–419
Thermal efficiency, 390
Furnace types :
Batch, 463
Continuous, 465, 475
Conveyor, 475, 476
Crucible, 420, 427
Cupola, 428–433
Dryer furnace, 655–659
Electric arc, 505, 506, 508, 510
Electric induction, 505, 506, 508, 509, 511
Electric resistance, 507, 508, 509, 512
General types, 384, 385
Glass-making, 447–459
Heat treatment, 471–478
Kilns and ovens, 435–447
Lead pots, 477
Melting, 420–426
Moving hearth, 476
Muffle, 473
Open hearth, 420–426
Reheating, 459–471
Salt baths, 477
Shaft, 420, 428
Single purpose, 473
Sintering, 433, 434
Soaking pits, 470
Top hat, 477
Utility, 472
Fusain, 6

Gases :
Leakage, 419
Sampling, 759–775
Transmission of, 178
Gasification :
Chemistry of, 57–59
Coal and coke, 29, 30
Thermal effects, 73
Gas measurement, 179–182, 204–217
Gas meters, 604
Gas producers, 355–375
Blast, 360–363, 365, 368
Built on, 461
Channelling in, 364
Clinker, 363, 369
Control, 370
Efficiency, 366, 367
Fuels, 358, 365
Layout of plant, 357, 479
Operation, 367–370
Glass-making furnaces, 447–459
Annealing furnaces, 449, 452
Efficiency of, 458
Fuels used, 453
Furnace control, 455–458
Heat flow diagram, 75
Operation, 450–453
Pot furnaces, 448, 451
Tank furnaces, 447, 450
Use of fuels, 454, 455

Gourley and Crimp's Formula, 225
Governors, gas, 673
Graphical heat balances :
  Construction, 710–717
  Principles, 709, 710
  Utilisation of heat, 717, 718
Gray-King Assay, 5
Grit emission, 240, 247

HAND-FIRING methods, 233–237, 620
Heat : definition of units, 60
Heat balances :
  Boilers, 341–349, 351, 712
  Cement kilns, 714
  Continuous vertical gas retorts, 713
  Furnaces, 403, 416
  Glass tank, 715
  Graphical, 709–719
  Open hearth steel furnace, 716
Heat drop, 108, 789
Heat exchangers, 151
Heat-flow gauges, 742
Heat flow in furnace stock, 387, 388, 466, 467, 480
Heat losses, 64, 85, 154, 417, 741
Heat transfer :
  Boilers, 160–163
  Coke fire, radiation from, 143
  Condensing vapours, 137
  Conduction, 119, 122–127
  Conduction through composite walls, 126
  Conduction through pipe walls, 127
  Conductivities, table of, 123
  Convection, 119, 128–139, 169
  Convection in liquids, 136
  Cornish and Lancashire boilers, 283
  Economisers, 164
  Emissivity, 139, 141
  Emissivity of CO_2, 145
  Emissivity of water vapour, 146
  Forced convection, 132–135
  From superheated steam, 135
  Furnaces, 383-389, 466, 467, 480–485
  Gas radiation, 144–151
  Heat flow in pipes, 128
  Losses, 154–160
    From surfaces, 155
    Through furnace wall, 157
  Luminous flames, radiation from, 150
  Methods of heat transfer, 119
  Modes of heat transfer in furnaces, 151
  Natural convection, 130
  Radiation, 120, 139–151
  Radiation, laws of, 139–142
  Smoke-and water-tube boilers, 277
  Through boiler plate, 120
  Transmittance, 124
  Vertical crosstube boilers, 276
Heat treatment furnaces, 471–480
  Conveyor furnaces, 475–477
  High temperature furnaces, 478
  Lead pots, 477
  Salt baths, 477
  Temperature distribution, 473–475
  Top hat furnaces, 477
  Use and design, 472
Hot-face insulation, 397–401, 542–545
Hot water :
  Heating by—*see* Space heating,
  Lagging pipes, etc., 547
  Supply, 626
Hydrogen :
  Effect on C.V., 71, 72
  Effect on flame, 74, 375
  Effect on flame speed, 74
  Flame temperature, 76

IGNITION limits, 73
Ignition temperature, 73
Induction furnaces—*see* Electric furnaces.
Instruments, 230, 403, 420, 425, 426, 470, 472, 647, 719–746
Insulating materials, 540–548
  Building methods for furnace insulation, 550, 551
  Classification, 541
  Effect of insulating firebricks, 543, 544, 545
  Effect of lagging, 547
  Insulating firebricks, 542–545
  Lagging, 545
  Method of application of lagging, 545, 546, 547
  Properties, 542, 543, 546
  Selection, 540–548
  Thermal conductivity, 542
Insulation
  Furnaces, 396–401
  Hot face, 399, 400, 542–545

KILNS, 435–447
  Belgian, 435, 440, 441, 442, 445
  Dressler, 439
  Firing, 442–445
  Fuel efficiency, 446
  Fuels used, 441
  Hoffmann, 440, 441, 445
  Thrift, 439
  Types, 436–441
  Wilkinson, 438
  Woodall-Duckham, 440
Kirchoff's law, 140

LACEY's Formula, 178
Lagging, 545
Lagging materials, 541, 546
Lambert's law, 140
Lancashire boilers, 281, 282
Latent heat of flue gases, 63
Latent heat of steam :
  Definition, 106
  Re-use in process work, 584–594
  Utilisation in process work, 556
Lead pots, 477
Lehrs, glass, 449, 452, 453
Lessing test, 5
Lignite, 6
Lime-soda process for feed water treatment, 328
Locomotive boilers, stationary, 279, 280, 282

MEASUREMENT, 202–228
  Examples, 226–228
  Gases, liquids, steam, 204–224
  Solid fuels, 202
  Water, 224

Melting furnaces, 420–426, 447
Multiple effect evaporation, 587–594
Multitubular boilers, externally fired, 284

Nusselt number, 129

Oil fuel :
  Burner operation, 706, 707
  Burners, 702
  Combustion, 703–707
  Fuel oils, 699, 700
  Preheating, 701
  Properties and testing, 699, 707, 708
  Storage, 700
  Transmission of, 198, 199, 701
  Use in dryers, 656, 657
  Use in furnaces, 405
  Use in glass industry, 454
Open cast (outcrop) coal :
  Composition, 47
  Storage, 47
Open hearth furnaces :
  Description, 424
  Fuel saving, 425, 426
  Heat balance, 422, 423
  Heat flow diagram, 716
Orifice plates for measurement of fluid flow, 179, 205–210
Orsat apparatus, 720–723
Overfeed firing : combustion, 92

Panel heating, 609
Peak steam loads, 594–603
Peat, 5
Petroleum :
  Composition, 696
  Cracking, 698
  Refining and cracking, 697
Pilot tube, 181
Poiseuille's Equation, 173, 174
Pole's formula, 178
Preheaters, air, 312
Pressurestats, 693
Producer gas :
  Analysis of, 370
  Composition, 29, 30, 361, 362
  Furnace heating, 374
  Impurities, 359, 371
  Limitations, 30, 356, 358
  Manufacture, 29, 355–370
  Purification, 371
  Selection for industrial purposes, 786
  Transmission of, 178, 198, 373
  Use in glass industry, 454
  Utilisation, 372–375
Producers—*see* Gas producers.
Pulverised fuel, 251–272
  Applications, 259
  Bin and feeder system, 251–255
  Burners, 257
  Cement industry, 260, 717
  Fineness, 254
  Furnace design, 262
  Metallurgical applications, 260
  Practice, 257
  Pulverisation, 252–255
  Ring main system, 256
Pulverised fuel—*continued.*
  Storage, 263
  Unit system, 255
  Use in boilers, 259, 263–272
  Use in furnaces, 404, 462
Pulverisers, 252, 254
Pumps :
  Boiler feed, 315
  Design, 315
  Centrifugal, 316
  Hot feed, 317–320
  Maintenance, 520
Pyrometers, 732–736

Radiation, 120, 139–151, 401, 419. *See also* Heat transfer.
Radiators for central heating, 607–609, 611, 613, 614, 615
Rake, 235
Records, need for keeping, 403
Recuperators, 151, 152, 392–396, 467
Refractory materials :
  Chrome, 517, 521, 524
  Composition, 515, 516, 517–521
  Cristobalite, 519, 520
  Dolomite, 517, 521
  Expansion, 526–529
  Firebrick, 517, 523
  Furnace construction, 548, 549, 551
  Influence on fuel economy, 515
  Insulating refractories, 538, 542–545
  Inversions of silica, 519, 520
  Jointing cements, 537–538
  Magnesite, 517, 521, 524
  Manufacture, 516, 518
  Mechanical properties, 532, 533
  Quartz, 519, 520
  Refractoriness, 521–524
  Selection, 539–540
  Silica brick, 517, 523
  Sillimanite, 517, 521, 524
  Slagging, 534, 537
  Spalling, 533, 534
  Specific gravity, 525
  Testing, 539–540
  Texture, 524
  Thermal properties, 528, 530–532, 533
  Tridymite, 519, 520
Regenerators :
  Air infiltration, 492
  Description, 153
  Distribution of gases, 455
  Purpose of, 392, 468, 486, 497
  Temperature control, 457
Reheating furnaces, 459–471
  Batch, 463, 464
  Coal fired, 460–463
  Continuous, 465, 406
  Efficiency, 468–470
  Gas fired, 463
  Heat recovery, 467, 468
  Oil fired, 463
  Pulverised fuel, 462
Resistance furnaces—*see* Electric furnaces.
Reynolds' number, 173, 175, 181, 184
Riddlings, 248

SAFETY valves, 297
Salt baths, 477
Sampling :
  Effect of segregation, 103
  Gases, 759–775
  Solid fuels, 747–758
Sankey diagrams, 711
Segregation, 103
Sensible heat :
  Flue gases, 63
  Steam, 105
Shaft furnaces, 428–433
Shatter test, coke, 24
Silica bricks—*see* Refractory materials.
Sillimanite—*see* Refractory materials.
Sintering furnaces, 433, 434
Slack :
  Combustion of, 90
  Size grading, 22
Slice bar, 235
Slurry : Hand firing on boilers, 236, 237
Smoke, 92
  Measurement, 744, 745
  Prevention, 235
Smoketube boilers, 276, 277
Soaking pits, 470
Soot blowers, 298
Soot removal, 298
South Wales coals :
  Characteristics, 10
  Classification, 9
  Combustion of slack, 90
Space heating :
  Boiler, 616–619
  Boiler fittings, 619, 620
  Boiler operation, 625, 626
  Draught, 622
  Firing methods, 620–623
  Fuel economy in, 624, 625
  General principles, 606
  Hand firing, 620
  Heating and ventilating 627–630
  Mechanical stoker, 621
  Principles, 606, 607
  Systems used, 607–618
  Thermostatic control, 623, 661–663
Spalling, 533
Specific heat, 61
Steam :
  De-super heating, 562
  Draining pipes, 578
  Dryness, 107, 218, 219
  Flash, 117, 581, 584
  Formation, 104–108
  Heat transfer, 564–571
  Latent heat, 106, 556, 584–594
  Losses, 604
  Measurement of, 212–224
  Pass out, 554–556
  Peak loads, 594–603
  Power production, 111, 554, 556
  Pressure, 104
  Sensible heat, 105, 556
  Superheat—definitions, 108
  Superheat for power production, 111
  Superheat in process work, 112–116, 135, 558–561, 604
  Total heat, 106
Steam—*continued.*
  Transmission of, 182, 562, 578
  Traps :
    Group trapping, 576
    Position of, 577
  Use of, 109–116, 553–556
  Wiredrawing, 109, 561
Steam traps, 572–580
Stefan-Boltzmann Law, 139, 142
Stock, rate of heating, 387, 388, 466, 467, 480–485
Stokers :
  Central heating boilers, 621
  Chain grate, 244
  Coking, 240, 241
  Sprinkler, 237–240
  Underfeed, 241, 243
Storage of coal, 39–48, 263
  Precautions, 42, 46, 263
Stratification in flues, 182–185
Sulphur in furnace work, 378, 379
Superheaters, 299–305, 493
Survey : Fuel Research—of British coals, 788, 789

TAR—*see* Coal tar.
Temperature measurement :
  Flue gas temperature, 736–740
  Heat loss from surfaces, 741–744
  Optical pyrometer, 732, 734
  Suction pyrometer, 738–740
  Surface temperatures, 740–741
  Thermocouples, 732, 733
  Thermometer, 729–732
Thermo-compression, 594
Thermostatic control :
  Boiler control, 666
  Equipment, 667–670
  Drying plant, 648
  Hot water supply, 663–665
  Maintenance, 670, 671
  Process work, 664, 665, 668–670
  Space heating, 623, 661–663
  Waste heat recovery, 665
" Throttling," 109
" Throttling " calorimeter, 218
Town gas—*see* Coal gas.
Town gas supply :
  Burners, 678–688
  Dampers, 691
  Flue gas composition, 54–56
  Flue gas heat losses, 67–70
  Flues, 688–691
  Gas fitting, 672, 673
  Governors, 673–678
  Meters, 694, 695
  Pipes, 178, 179, 200, 673
  Thermostatic control, 691–693
  Valves and cocks, 691, 694
Transmittance, 124
Traps, steam, 572–580

UNDERFEED firing :
  Combustion, 94
  Stokers in operation, 237–244
Unit heaters, 614
Unwin's Formula, 179

VENTILATION, 614, 627
Venturi meter, 180
Viscosity :—
  Definition, 170, 171
  Gases, 172, 173
  Liquids, 170, 171
Vitrain, 6
V-notch, measurement of water by, 224

WASTE heat boilers, 288, 486–498
  Air inleakage, effect of, 490–492
  Draught, 489
  Economics, 496, 497
  Economisers, 494
  Efficiency, 494–496, 780
  Gas temperatures, 488, 493
  Heat recovery by, 393, 394, 468, 487–489
  Selection of boiler, 497–498
  Superheaters, 493
Water :
  Level in boilers, 245
  Measurement of, 224–226
  Vapour pressure of, 631, 632, 651
Water gas, 31, 32
  Transmission of, 199
Water jets for firebed cooling, 97–100
Water meters : Principles of construction, 224–226
Water-tube boilers, 289–296
Weir tanks, 224
" Wiredrawing," 109, 561
" Wiredrawing " calorimeter, 218
Woodall-Duckham method of coal analysis, 5, 778

ZEOLITE process for feed water treatment, 329

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