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APPLIED HEAT

ADAPTED FROM

DER WÄRMEINGENIEUR

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PREFACE

As civilization advances and the requirements of mankind increase and expand, the demand for energy to meet these requirements increases in corresponding manner. At the present time, this demand is met almost entirely by the heat energy of various forms of fuel, the supply of which is as yet equal to the demand. There are signs, however, that this happy state of affairs may not continue indefinitely, and the winning of both solid and liquid fuel is becoming less and less easy. It is of the utmost importance, therefore, that the utilization of the available fuel should be conducted in the most efficient manner. In *Der Wärmeingenieur*, the author has made a survey of the principal uses to which heat energy is applied, particular attention being paid to the efficiency of the processes and the prevention of waste. In translating and adapting the work for the English reader, considerable revision has been necessary, and some sections which were entirely inapplicable to British conditions have been omitted. The occurrence of large quantities of inferior fuels on the Continent necessitates the use of appropriate furnaces and plants, some descriptions of which have been retained as their utilization is of increasing importance universally. Considerable attention has also been paid to the increase of efficiency which can be obtained by a more common use of the heat in exhaust and flue gases and in exhaust steam. A subject of so wide a character can only be treated within the confines of a single volume by a process of selection. The heat reactions in connexion with furnaces (Chap. III and V) have been treated in considerable detail as the text-books on the subject are mostly confined to qualitative descriptions of the chemical actions, quantitative estimates of the heat being often omitted. The same treatment of the heat reactions in connexion with steam and internal-combustion engines is not necessary owing to the existence of adequate books on these subjects, such as Ricardo's *Internal-Combustion Engine* and Sim's *Steam Condensing Plant*.

Illustrations of certain Continental plants have been replaced by others of English origin, and many additional figures of important plants and apparatus have been introduced. For permission to use these, thanks are due to various firms and publications. Individual acknowledgment is made in each case under the diagrams concerned.

H. Moss.

August, 1927.

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APPLIED HEAT

CHAPTER I

General Principles

All manufacture can be divided into two factors, work expended in alteration of form or chemical transformation and heat consumption or production. The two factors are intimately connected, one depending on the other. In the ensuing chapters, particular study is made of heat production and consumption, the expenditure of work in alteration of form or chemical transformation being considered only in so far as it affects the former problems. Efficiency is the essence of successful commercial production, and a knowledge of the principles underlying heat transformations is essential to the technical engineer who desires to eliminate unnecessary losses. Industrial processes are often of a very complex character, but the fundamental laws of the theory of heat are applicable in all cases, and serve as a guide to the required efficiency.

1. CONSERVATION OF ENERGY

The law of the conservation of energy states that all kinds of energy, potential and kinetic, chemical, optical, magnetic, electric, and thermal, are fundamentally identical, and that the total energy in an isolated system withdrawn from all external influences is of constant magnitude. As energy of the different kinds enumerated is usually expressed in different systems, the conversion of one into the other in a numerical calculation is effected by equivalent values. In the English technical system:

Unit of mechanical energy = 1 ft.-lb. = 0·001285 B.Th.U.
Unit of electrical energy = 1 B.O.T. unit = 3412 B.Th.U.
(1 Board of Trade unit = 1000 watts for 1 hr.)
Unit of heat = 1 B.Th.U. = 778 ft.-lb.
(1 British Thermal Unit = Heat to raise 1 lb. of water through 1° F.)

2. TOTAL HEAT AND ENTROPY

The total heat of a substance is given by the expression

$$H = U + pv,$$

where U is the internal energy of the body possessed in virtue of the arrangement and velocities of the molecules, while p and v are the pressure and volume respectively. The product pv represents the work done in forcing the substance into the space occupied against a constant pressure p. For solids pv is small and varies to a negligible extent with temperature in comparison to the changes in the internal energy. For liquids it is larger, but is still only a small proportion of the total heat. For gases pv is important and increases appreciably with temperature. During a change of state, an increment of total heat is received of which a considerable amount may be due to an increment in pv. This is particularly the case when water changes from liquid to vapour.

If a body receives an amount of heat dH and is allowed to expand against a constant external pressure p, part is retained as an increase dU in internal energy of the body and part dW is utilized in doing external work; thus:

$$dH = dU + dW = dU + pdv.$$

The heat can also be expressed by the equation

$$dH = Td\phi,$$

where T is the absolute temperature and ϕ the entropy. In an adiabatic change no heat is received or emitted, so that $dH = 0$ and $d\phi = 0$. Consequently the entropy is a constant for an adiabatic action. If a quantity of heat dH pass from a body at a temperature T_1 to another at a lower temperature T_2, the loss of entropy of the first will be dH/T_1 and the gain of entropy of the second will be dH/T_2. The gain of the second is greater than the loss of the first, and it may be stated generally that in any system of bodies protected from outside influences, any interchanges of heat tend to increase the total entropy. The entropy of a body is a function of its state only, and if the body is brought back to the same condition of pressure, temperature, and volume it will have the same entropy. The internal energy of a body is also a function of its state, and hence in a reversible cycle there are two conditions to be fulfilled. If a quantity of heat dH_1 is given up by the body at temperature T_1, while a quantity dH_2 is returned to it at the lower temperature T_2,

$$\frac{dH_1}{T_1} = \frac{dH_2}{T_2}.$$

Hence $dH_1 > dH_2$. The loss of internal energy dU_1 being equal to the gain of internal energy dU_2,

$$dH_1 - dW_1 = dH_2 - dW_2,$$
$$dH_1 - dH_2 = dW_1 - dW_2.$$

The body being restored to its original state, a quantity of heat dH_1 has been given, and the excess heat $dH_1 - dH_2$ transformed to work $dW_1 - dW_2$, the efficiency

$$\frac{dW_1 - dW_2}{dH_1} = \frac{dH_1 - dH_2}{dH_1} = \frac{T_1 - T_2}{T_1}.$$

If a substance passes through a single action and not through a cycle, and its total heat falls from H_1 to H_2 while an amount of *useful* work W_1 is done, the efficiency of the action is $W_1/(H_1 - H_2)$.

3. SPECIFIC AND LATENT HEAT

In order to utilize these general theorems in a particular case, it is necessary to know the behaviour of the medium used when subjected to heat under various circumstances, also the character of the expansion curve and the amount of work done when it is allowed to expand under different conditions. If the volume is maintained constant and heat dH is applied to unit mass while the specific heat at constant volume is C_v,

$$\begin{aligned} dH &= dU + p dv \\ &= dU; \\ C_v &= \left(\frac{dH}{dT}\right)_{v \text{ const}} = \frac{dU}{dT}. \end{aligned}$$

If the substance is allowed to expand under constant external pressure p, and C_p is the specific heat at constant pressure,

$$\begin{aligned} C_p &= \left(\frac{dH}{dT}\right)_{p \text{ const.}} \\ &= \frac{dU}{dT} + p\left(\frac{dv}{dT}\right)_p \\ &= C_v + p\left(\frac{dv}{dT}\right)_p. \end{aligned}$$

When the unit mass is at its boiling-point under the pressure p and the heat applied results in its vaporization,

$$dH = L = dU + p(v_2 - v_1),$$

where L is the latent heat and v_2 and v_1 are the volumes of vapour and liquid

respectively. A similar expression applies to a solid at the melting-point. For a solid, the expansion due to heating is so small that $C_p = C_v$, and only for a gas or vapour is the difference important.

4. WORK OF EXPANSION

For unit mass of a perfect gas

$$pv = kT,$$
$$p\left(\frac{dv}{dT}\right)_{p \text{ const.}} = k,$$
$$C_p = C_v + k.$$

The isothermal expansion of a perfect gas is according to the law

$$pv = \text{constant}.$$

The adiabatic expansion law is

$$pv^\gamma = \text{constant},$$

where $\gamma = C_p/C_v$.

The work done in isothermal expansion from volume v_1 to volume v_2,

$$W = kT \log_e v_2/v_1.$$

The work done in adiabatic expansion,

$$W = \frac{p_1v_1 - p_2v_2}{\gamma - 1} = \frac{k(T_1 - T_2)}{\gamma - 1}$$
$$= C_v(T_1 - T_2).$$

The change in total heat per unit mass,

$$H_1 - H_2 = U_1 + p_1v_1 - U_2 - p_2v_2.$$

For isothermal expansion,

$$H_1 - H_2 = U_1 - U_2 = C_v(T_1 - T_2).$$

For adiabatic expansion,

$$H_1 - H_2 = C_v(T_1 - T_2) + p_1v_1 - p_2v_2$$
$$= C_p(T_1 - T_2).$$

The change in entropy per unit mass,

$$\phi_2 - \phi_1 = \int \frac{dH}{T} = C_v \log \frac{T_2}{T_1} + k \log \frac{v_2}{v_1}$$
$$= C_p \log_e \frac{T_2}{T_1} - k \log_e \frac{p_2}{p_1}.$$

For isothermal expansion

$$\phi_2 - \phi_1 = -k \log_e \frac{p_2}{p_1} = +k \log_e \frac{v_2}{v_1}.$$

For adiabatic expansion,

$$\phi_2 - \phi_1 = 0.$$

If a perfect gas has undergone expansion which is neither isothermal nor adiabatic, but can be represented by the equation $pv^x = \text{constant}$, then

$$W = \frac{p_1v_1 - p_2v_2}{x - 1} = \frac{k(T_1 - T_2)}{x - 1},$$

$$H_1 - H_2 = (C_v + k)(T_1 - T_2),$$

$$\phi_2 - \phi_1 = C_v \log\left(\frac{T_2}{T_1}\right)\left(\frac{x - \gamma}{x - 1}\right).$$

5. IMPERFECT GAS OR VAPOUR

The behaviour of an actual gas deviates from the law $pv = kT$. At high temperatures there is an apparent increase in specific heat. In many cases this is due to dissociation of the molecules which reunite on subsequent cooling. If

$$C_p = \alpha + \delta T,$$
$$C_v = \beta + \delta T,$$

the equation for adiabatic expansion or compression is

$$p^\beta v^\alpha e^{\delta T} = \text{constant},$$

where e is the base of Napierian logarithms, 2·71828.

When an actual gas is cooled and compressed so that it is approaching condensation to liquid form, the volume diminishes more rapidly with increase of pressure than would be expected from the perfect gas law. This deviation has been expressed by Van der Waals in the characteristic equation, where a and b are constants:

$$\left(p + \frac{a}{v^2}\right)\left(v - b\right) = kT.$$

Although this form of equation represents the properties of a vapour more accurately than the simple form, there are discrepancies in the neighbourhood of the critical pressure. Callendar's form of equation is given by

$$v - b = \frac{kT}{p} - c,$$

where $c = c_0 T_0^n / T^n$. In both forms of equation b, which is termed the co-volume, represents the volume of the molecules and is practically equal to the volume of the substance in the liquid state. c is the co-aggregation

volume and represents the diminution in volume due to the pairing of molecules. From an equation of this character with the constants applicable to the vapour under consideration, general expressions for the total heat, entropy, and work of expansion may be obtained in a manner similar to that for the perfect gas, but for practical purposes it is more convenient to use tables or charts giving these functions under various conditions of pressure, volume, and temperature. Such tables are available for all the substances in common use, examples of which are given in the next section.

6. PROPERTIES OF COMMON SUBSTANCES

The tables are arranged in different ways according to the preference of the computers. Selected extracts of various forms are given, more detailed figures being available from the sources quoted. In all the tables,

t = temperature in degrees Fahrenheit;
T = absolute temperature = $460 + t$;
p = pressure in pounds per square inch absolute = $14{\cdot}7$ + gauge pressure;
vacuum = pressure below atmospheric in inches of mercury $= 30(14{\cdot}7 - p)/14{\cdot}7$;
v = volume of 1 lb. in cubic feet;
H = total heat in B.Th.U. per pound;
L = latent heat in B.Th.U. per pound;
$H_{\text{saturated vapour}} = H_{\text{liquid at B.P.}} + L$;
ϕ = entropy per pound;
$\phi_{\text{saturated vapour}} = \phi_{\text{liquid at B.P.}} + L/T$;
U = internal energy in B.Th.U. per pound $= H - pv144/778$;
External work during vaporization in B.Th.U. per pound $= p(v_{\text{saturated vapour}} - v_{\text{liquid at B.P}})\ 144/778$.

PROPERTIES OF STEAM[1]

LIQUID AND SATURATED VAPOUR (LOW PRESSURES)

Vacuum.	p	t	$v_{vap.}$	$H_{liq.}$	H_{vap}	$\phi_{vap.}$
−29	0·490	78·9	653·0	46·8	1091·9	2·032
−28	0·979	101·0	339·8	68·8	1102·1	1·974
−27	1·469	114·9	231·9	82·7	1108·5	1·941
−25	2·448	133·6	143·5	101·4	1117·0	1·899
−20	4·896	161·4	74·9	129·2	1129·4	1·844
−10	9·793	192·2	39·12	160·1	1142·6	1·789
0	14·690	212·0	26·79	180·0	1150·7	1·757
+10	19·59	226·8	20·46	194·9	1156·7	1·735
+20	24·48	238·9	16·62	207·2	1161·3	1·717

[1] From *The Enlarged Callendar Steam Tables*, by permission of the Author and the Publishers, Messrs. Edward Arnold & Co.

p =	30			40			60			100		
$t_{sat.}$ =	250·3			267·2			292·6			327 7		
t	H	v	φ	H	v	φ	H	v	ϕ	H	v	ϕ
$t_{sat.}$	1165·6	13·74	1·702	1171·7	10·50	1·679	1180·4	7·184	1·648	1191·3	4·451	1·608
300	1191·0	14·80	1·736	1188·8	11·03	1·702	1184·4	7·269	1·653	1175·5	4·254	1·588
350	1216·2	15·84	1·768	1214·4	11·83	1·735	1210·8	7·815	1·687	1203·7	4·606	1·624
400	1241·0	16·88	1·798	1239·6	12·61	1·765	1236·7	8·353	1·718	1230·8	4·944	1·656
450	1265·6	17·90	1·826	1264·4	13·38	1·793	1262·0	8·878	1·746	1257·2	5·272	1·686
500	1290·1	18·91	1·852	1289·1	14·15	1·820	1287·1	9·399	1·773	1283·1	5·596	1·714
550	1314·4	19·92	1·877	1313·6	14·92	1·844	1311·9	9·925	1·798	1308·6	5·913	1·739
600	1338·7	20·93	1·900	1338·0	15·68	1·868	1336·6	10·43	1·822	1333·7	6·226	1·764
650	—	—	—	1362·3	16·44	1·890	1361·1	10·94	1·845	1358·6	6·537	1·787

p =	200			500			1000			2000		
$t_{sat.}$ =	381·8			468·6			547·3			636·8		
t	H	v	φ	H	v	ϕ	H	v	ϕ	H	v	ϕ
$t_{sat.}$	1205·4	2·320	1·554	1222·2	0·977	1·481	1230·9	0·506	1·424	1232·3	0·260	1·363
400	1216·3	2·387	1·567	—	—	—	—	—	—	—	—	—
450	1245·2	2·568	1·599	1209·2	0·944	1·467	—	—	—	—	—	—
500	1273·1	2·741	1·629	1243·2	1·029	1·503	1193·3	0·459	1·386	—	—	—
550	1300·2	2·910	1·657	1275·0	1·109	1·536	1233·0	0·509	1·426	—	—	—
600	1326·6	3·075	1·682	1305·3	1·185	1·565	1269·8	0·554	1·462	1198·8	0·239	1·332
700	1378·2	3·396	1·729	1362·7	1·327	1·617	1336·8	0·637	1·522	1285·2	0·289	1·410
800	—	—	—	1417·0	1·463	1·662	1397·7	0·714	1·573	1359·1	0·339	1·471
900	—	—	—	—	—	—	1454·8	0·784	1·617	1425·5	0·379	1·522
1000	—	—	—	—	—	—	—	—	—	1486·8	0·419	1·565

PROPERTIES OF AMMONIA, NH_3 [1]

Liquid, Saturated Vapour, and Superheated Vapour

p		Liquid.	Saturated Vapour.	Superheated Vapour. Degree of Superheat.				
				50° F.	100° F.	150° F.	200° F.	250° F.
1	t	−103·7	−103·7					
	v	0·022	225·0					
	H	—	—	—	—	—	—	—
	ϕ	—	—					
3	t	−76·5	−76·5					
	v	0·022	80·0					
	H	—	—	—	—	—	—	—
	ϕ	—	—					
5	t	−62·0	−62·0					
	v	0·023	49·3					
	H	−98·1	519·1	—	—	—	—	—
	ϕ	−0·221	1·332					
10	t	−40·4	−40·4					
	v	0·023	25·75					
	H	−75·7	526·4	—	—	—	—	—
	ϕ	−0·166	1·270					
15	t	26·4	−26·4					
	v	0·024	17·60					
	H	−61·2	530·9	—	—	—	—	—
	ϕ	−0·132	1·234					
20	t	−15·9	−15·9	34·1	84·1	134·1	184·1	234·1
	v	0·024	13·45	15·2	16·9	18·6	20·2	21·8
	H	−50·3	534·0	561·4	587·4	612·7	638·0	663·4
	ϕ	−0·107	1·209	1·268	1·318	1·362	1·403	1·441
30	t	+0·1	+0·1	50·1	100·1	150·1	200·1	250·1
	v	0·024	9·17	10·41	11·55	12·65	13·74	14·81
	H	−33·6	538·5	567·1	593·7	619·5	645·2	670·9
	ϕ	−0·071	1·174	1·233	1·283	1·327	1·367	1·405
50	t	22·1	22·1	72·1	122·1	172·1	222·1	272·1
	v	0·025	5·67	6·44	7·14	7·81	8·46	9·11
	H	−10·3	544·3	574·4	602·1	628·7	655·0	681·0
	ϕ	−0·0216	1·130	1·189	1·239	1·283	1·323	1·360
75	t	41·3	41·3	91·3	141·3	191·3	241·3	291·3
	v	0·025	3·86	4·39	4·86	5·32	5·76	6·20
	H	10·3	548·8	580·4	609·2	636·5	663·2	689·8
	ϕ	0·020	1·095	1·155	1·205	1·249	1·288	1·325

[1] Reprinted, by permission, from Macintire's *Mechanical Refrigeration.* Published by John Wiley & Sons, Inc.

PROPERTIES OF AMMONIA, NH_3

LIQUID, SATURATED VAPOUR, AND SUPERHEATED VAPOUR (*Cont.*)

p		Liquid.	Saturated Vapour.	Superheated Vapour. Degree of Superheat.				
				50° F.	100° F.	150° F.	200° F.	250° F.
100	t	56·0	56·0	106·0	156·0	206·0	256·0	306·0
	v	0·026	2·94	3·34	3·70	4·05	4·38	4·71
	H	26·5	552·1	584·8	614·3	642·2	669·4	696·3
	ϕ	0·052	1·071	1·131	1·181	1·225	1·264	1·301
150	t	78·5	78·5	128·5	178·5	228·5	278·5	328·5
	v	0·027	1·99	2·26	2·51	2·75	2·97	3·19
	H	51·8	556·7	591·2	621·9	650·7	678·6	706·0
	ϕ	0·099	1·037	1·099	1·149	1·192	1·231	1·267
250	t	110·1	110·1	160·1	210·1	260·1	310·1	360·1
	v	0·028	1·21	1·38	1·53	1·68	1·81	1·95
	H	89·7	562·5	599·4	631·9	661·9	690·8	719·0
	ϕ	0·167	0·996	1·059	1·109	1·152	1·191	1·227
400	t	142·9	142·9	192·9	242·9	292·9	342·9	392·9
	v	0·029	0·75	0·86	0·96	1·06	1·14	1·23
	H	132·9	567·9	607·1	642·3	672·7	702·6	731·7
	ϕ	0·239	0·961	1·023	1·074	1·117	1·156	1·191
600	t	174·7	174·7	224·7	274·7	324·7	374·7	424·7
	v	0·032	0·49	0·57	0·64	0·70	0·76	0·82
	H	180·4	572·7	613·8	649·5	682·2	713·1	743·0
	ϕ	0·314	0·932	0·995	1·045	1·088	1·126	1·161

PROPERTIES OF CARBON DIOXIDE, CO_2[1]

LIQUID AND SATURATED VAPOUR

t.	p.	$v_{liq.}$	$v_{vap.}$	$H_{liq.}$	$H_{vap.}$	$\phi_{liq.}$	$\phi_{vap.}$
−20	218	0·016	0·416	−20·5	102·7	−0·046	0·234
−10	260	0·016	0·348	−17·1	102·9	−0·039	0·228
0	308	0·016	0·293	−13·5	103·0	−0·030	0·224
10	362	0·017	0·245	−9·7	103·0	−0·022	0·218
20	422	0·017	0·205	−5·6	102·7	−0·013	0·212
30	489	0·017	0·172	−1·1	101·8	−0·003	0·206
40	565	0·018	0·144	+4·3	100·4	+0·007	0·198
50	650	0·019	0·120	10·8	98·2	0·017	0·189
60	744	0·020	0·099	17·9	95·3	0·030	0·179
70	848	0·021	0·080	26·4	91·1	0·045	0·168
80	962	0·024	0·062	37·2	84·7	0·065	0·154

[1] Reprinted, by permission, from Greene's *Elements of Refrigeration*. Published by John Wiley & Sons, Inc.

PROPERTIES OF CARBON DIOXIDE, CO_2[1]

Superheated Vapour

p	Entropy.											
	0·18.			0·20			0·22.			0·24.		
	t	H	v	t	H	v	t	H	v	t	H	v
200	—	—	—	—	—	—	—	—	—	−16·6	103·3	0·460
240	—	—	—	—	—	—	—	—	—	+1·4	106·6	0·385
280	—	—	—	—	—	—	—	—	—	23·0	109·6	0·344
320	—	—	—	—	—	—	—	—	—	42·8	112·1	0·316
360	—	—	—	—	—	—	14·0	104·4	0·250	59·0	114·7	0·288
400	—	—	—	—	—	—	28·4	106 0	0·222	71 6	116·3	0 263
450	—	—	—	—	—	—	44·6	107 6	0·203	87·8	118·4	0·248
500	—	—	—	—	—	—	60 8	109·6	0·195	98·6	120·6	0·223
550	—	—	—	39·2	100·8	0 147	73·4	111·2	0·181	111·2	122·8	0 210
600	—	—	—	51·8	102·1	0 139	86·0	112 7	0·169	118 4	124·6	0 194
700	—	—	—	77 0	104·8	0·127	107·6	115 9	0 150	140 0	128·2	0 170
800	71·6	96·3	0·092	95 0	107·1	0 113	122·0	118 4	0 130	159·8	130 9	0 153
900	89 6	98 1	0·081	113 0	109 4	0 104	141·8	121·1	0·120	188 6	133·7	0·138
1000	100·4	99 7	0·076	127 4	111·2	0 095	158 0	123·3	0·112	199·4	136 3	0·130

PROPERTIES OF SULPHUR DIOXIDE, SO_2[1]

Liquid and Saturated Vapour

t	p	$v_{liq.}$	$v_{vap.}$	H_{liq}	$H_{vap.}$	$\phi_{liq.}$	$\phi_{vap.}$
−20	5·9	0·01	12·02	−15·9	149·0	−0·035	0·340
−10	7·9	0·01	9·12	−12·9	150·7	−0·028	0·336
0	10·4	0·01	7·12	−9·9	152·1	−0·021	0·331
10	13·3	0·01	5·56	−6·8	153·7	−0·015	0·327
20	17·0	0·01	4·42	−3·8	155·3	−0·008	0·324
30	21·5	0·01	3·56	−0·6	156·8	−0·001	0·320
40	26·7	0·01	2·88	+2·6	158·0	+0·005	0·316
60	40·4	0·01	1·96	9·3	160·4	+0·018	0·309
80	59·3	0·01	1·37	16·1	162·5	0·031	0·302
100	84·4	0·01	0·96	23·1	164·0	0·044	0·296

[1] From Greene's *Elements of Refrigeration.*

PROPERTIES OF SULPHUR DIOXIDE, SO_2[1]

SUPERHEATED VAPOUR

p	Entropy.											
	0·30			0·32			0·34			0·36		
	t	H	v	t	H	v	t	H	v	t	H	v
6	—	—	—	—	—	—	—	—	—	12	158·4	12·75
8	—	—	—	—	—	—	−4	152·5	9·37	25	162·0	9·76
10	—	—	—	—	—	—	+7	155·2	7·54	36	164·9	7·32
15	—	—	—	—	—	—	32	161·3	5·15	61	171·5	5·35
20	—	—	—	—	—	—	52	165·6	3·95	84	176·0	4·10
30	—	—	—	52	161·6	2·60	82	172·3	2·71	113	183·6	2·82
40	—	—	—	75	166·1	2·00	104	176·3	2·08	138	189·0	2·16
50	—	—	—	91	169·4	1·64	122	181·3	1·69	156	193·3	1·76
60	—	—	—	106	172·6	1·38	138	184·5	1·43	172	197·1	1·48
70	91	163·8	1·15	120	175·1	1·19	153	187·4	1·24	—	—	—
80	103	165·6	1·02	133	177·3	1·07	167	190·1	1·10	—	—	—
90	112	167·2	0·91	144	179·5	0·95	180	192·4	0·99	—	—	—
100	120	168·7	0·84	153	181·3	0·87	192	194·4	0·92	—	—	—

A selected list of the physical properties of some other substances with which the heat engineer has to deal are given in the following tables. It must be noted that in the case of manufactured articles, the values are dependent upon the mechanical state and mode of manufacture. With some materials the heat conductivity increases greatly at high temperatures.

Melting-points at Atmospheric Pressure

	° F.		° F.
Platinum	3180	Paraffin wax	129
Mild steel	2450 to 2650	Benzene	42
Steel	2350 to 2550	Water	0
Cast iron, grey	2200	Mercury	−38
Glass (softens at)	750 to 1450	Sulphur dioxide	−105
Copper	1980	Ammonia	−108
Silver	1760	Carbon dioxide	−110
Brass	1650	Chlorine	−152
Aluminium	1212	Ethyl alcohol	−166
Zinc	788	Ether	−181
Lead	621	Nitrogen	−346
Sulphur	240	Oxygen	−376

[1] From Greene's *Elements of Refrigeration*.

Boiling-points at Atmospheric Pressure

	°F.		°F.
Zinc	1680	Ammonia	−27
Sulphur	834	Carbon dioxide	−108
Mercury	675	Acetylene	−119
Paraffin wax	570	Ethylene	−155
Water	212	Oxygen	−298
Benzene	176	Air	−317
Ethyl alcohol	173	Nitrogen	−321
Ether	97	Hydrogen	−424
Ethyl chloride	54	Helium	−449
Sulphur dioxide	14		

Critical Temperatures and Pressures

	°F.	Pounds per Sq. Inch.		°F.	Pounds per Sq. Inch.
Water	705	3200	Oxygen	−181	784
Sulphur dioxide	313	1160	Air	−220	572
Chlorine	286	1195	Carbon monoxide	−222	528
Ammonia	266	1680	Nitrogen	−231	516
Carbon dioxide	88	1068	Hydrogen	−404	295

Latent Heat of Fusion at Atmospheric Pressure

	B.Th.U. per Pound.		B.Th.U. per Pound.
Ice	144	Benzene	54
Aluminium	138	Zinc	50
Blast-furnace slag	90	Tin	25
Copper	77	Sulphur	16
Paraffin	63	Lead	11
Iron	54	Mercury	5

Latent Heat of Vaporization at Atmospheric Pressure

	B.Th.U. per Pound.		B.Th U. per Pound.
Water	971	Turpentine	126
Sulphur	540	Mercury	122
Ammonia	590	Hydrogen	222
Ethyl alcohol	378	Oxygen	92
Benzene	169	Nitrogen	86
Sulphur dioxide	161		

Specific Heats

Aluminium	0·210	Ether	0·520
Lead	0·031	Ethyl alcohol	0·600
Iron	0·114	Ammonia (liquid)	1·000
Gold	0·031	Liquid sulphur dioxide	0·320
Silver	0·560	Glycerine	0·580
Zinc	0·096	Oil	0·400
Tin	0·056	Petroleum	0·510
Copper	0·095	Aniline	0·514
Brass	0·092	Turpentine	0·420
Nickel	0·110		
Basalt	0·200	Water at 22° F.	1·016
Concrete	0·270	,, 32° F.	1·011
Ice	0·505	,, 70° F.	1·000
Glass	0·195	,, 100° F.	0·999
Wood	0·510 to 0·650	,, 150° F.	1·001
Compressed cork	0·250	,, 200° F.	1·005
Bricks	0·200 to 0·300	,, 212° F.	1·007
Paraffin	0·600	,, 250° F.	1·012
Sandstone	0·220	,, 300° F.	1·021
Slag	0·180	,, 350° F.	1·031
Pitch pine	0·650		
Tiles	0·190 to 0·241		
Limestone	0·220	Mercury at 32° F.	0·0335
Coke	0·200	,, 100° F.	0·0331
Rubber	0·820	,, 200° F.	0·0327
Oil (solid)	0·350	,, 300° F.	0·0324

	Percentage of Water.	Specific Heat Before the Freezing Process.	Specific Heat After the Freezing Process.	Melting Heat, B.Th.U. per Pound.
Lean meat	72	0·77	0·41	104
Fat meat	51	0·60	0·34	73
Fat pork	39	0·51	0·30	56
Veal	63	0·70	0·39	91
Eggs	70	0·76	0·40	101
Potatoes	74	0·80	0·42	106
Cabbage	91	0·93	0·48	131
Milk	87	0·90	0·47	125
Eel	62	0·69	0·38	89
Fowl	74	0·80	0·42	106

The specific heat of gases which are far removed from the saturation-point are given in the following table:

Gas.	Formula.	Molecular Weight.	Density, Pounds per cubic foot at 1 atm. and 60° F.	Specific Heat.		$\frac{C_p}{C_v}$
				C_p	C_v	
Air	—	28·95	0·0762	0·242	0·173	1·40
Oxygen ..	O_2	32·00	0·0843	0·242	0·173	1·40
Nitrogen ..	N_2	28·02	0·0738	0·235	0·17	1·41
Hydrogen ..	H_2	2·02	0·0053	3·405	2·402	1·407
Carbon monoxide	CO	28·00	0·0738	0·250	0·179	1·40
Carbon dioxide	CO_2	44·00	0·1165	0·202	0·165	1·30
Steam (212° F.)	H_2O	18·02	0·0353	0·478	0·390	1·305
Sulphur dioxide	SO_2	64·07	0·1725	0·154	0·123	1·26
Ammonia ..	NH_3	17·03	0·510	0·530	0·400	1·33
Methane ..	CH_4	16·03	0·0423	0·590	0·450	1·31
Ethylene ..	C_2H_4	28·03	0·0745	0·400	0·320	1·26
Acetylene ..	C_2H_2	26·02	0·0696	0·370	0·290	1·26

Coefficients of Heat Conduction in B.Th.U. per square foot per hour for a thickness of 1 ft. and a temperature difference of 1° F.

Aluminium	122	Firebrick	0·3 to 1·0
Iron	35	Cement	0·5
Copper	220	Water	0·35
Brass	63	Asbestos	0·13
Zinc	64	Cotton	0·035
Ice	1	Sawdust	0·035
Concrete	0·43	Cork	0·025
Porcelain	0·6	Wool	0·02
Wood	0·1 to 0·15	Fused silica ..	0·56

7. REVERSIBLE CYCLE

The Carnot reversible cycle with a gaseous medium is represented in fig. 1 on a pressure-volume diagram and in fig. 2 on a temperature-entropy diagram. From the former, the external work performed in the various stages can be obtained, thus:

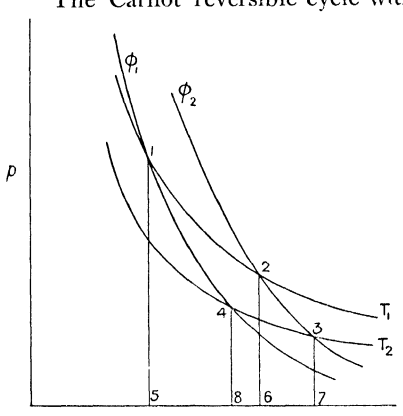


Fig. 1.—The Carnot Reversible Cycle with a Gaseous Medium on a Pressure-volume Diagram

Work by the gas during isothermal expansion at T_1 = area 1265;
Work by the gas during adiabatic expansion from T_1 to T_2 = area 2376;
Work on the gas during isothermal compression at T_2 = area 3487;
Work on the gas during adiabatic compression from T_2 to T_1 = area 4158;
Net work done by gas W = area 1234.

From fig. 2 the heat received is determined:

Heat received by the gas during isothermal expansion at T_1 = area 1265;
Heat received by the gas during adiabatic expansion from T_1 to T_2 = 0;
Heat given up by the gas during isothermal compression at T_2 = area 3487;
Heat given up by the gas during adiabatic compression from T_2 to T_1 = 0;
Net heat received by gas = area 1234.

$$H_1 - H_2 = (\phi_2 - \phi_1)(T_1 - T_2) = W.$$

$$\text{Efficiency} = \frac{W}{H_1} = \frac{(\phi_2 - \phi_1)(T_1 - T_2)}{(\phi_2 - \phi_1)T_1}$$

$$= \frac{T_1 - T_2}{T_1}$$

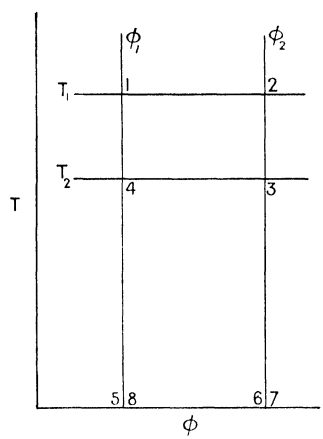


Fig. 2.—The Carnot Reversible Cycle on a Temperature-entropy Diagram

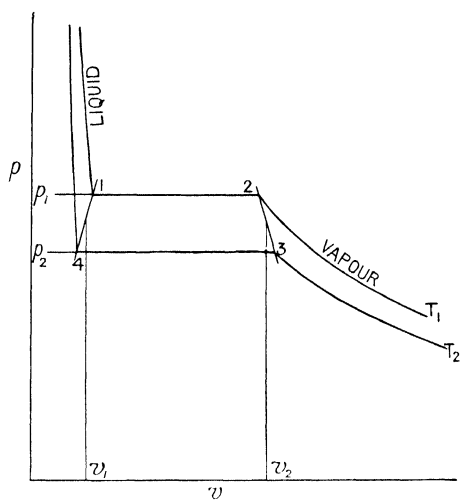


Fig. 3.—Two Isothermals for Liquid-vapour

8. CHANGE OF STATE

Two isothermals for liquid-vapour are represented in fig. 3 by T_1 and T_2, the constant pressure lines 12 and 34 representing the condensation conditions. If unit mass of the substance be taken round the cycle 1234, then

Heat given to medium during vaporization at T_1 = L_1,
Heat given up by vapour falling from T_1 to T_2 = $s_2(T_1 - T_2)$,
Heat given up by vapour condensing at T_2 = L_2,
Heat given to liquid heated from T_2 to T_1 = $s_1(T_1 - T_2)$,
Net heat given = $L_1 - L_2 + (s_1 - s_2)(T_1 - T_2)$.

If the temperature interval is very small,

$$\text{Net heat given} = d\mathrm{L} + (s_1 - s_2)d\mathrm{T}.$$

This is equivalent to the work done by the medium which for the small temperature interval is $(v_2 - v_1)dp$. Hence

$$s_1 - s_2 = \frac{d\mathrm{L}}{d\mathrm{T}} + (v_2 - v_1)\frac{dp}{d\mathrm{T}}.$$

The efficiency of the cycle being the ratio of the work done to the heat absorbed,

$$= \frac{(v_2 - v_1)dp}{\mathrm{L}_1 + s_1 d\mathrm{T}}.$$

For a small temperature interval $s_1 d\mathrm{T}$ is negligible in comparison with L_1, which may be replaced by L representing either L_1 or L_2. Hence

$$\frac{(v_2 - v_1)dp}{\mathrm{L}} = \frac{d\mathrm{T}}{\mathrm{T}},$$

or

$$\frac{\mathrm{L}}{\mathrm{T}} = (v_2 - v_1)\frac{dp}{d\mathrm{T}}.$$

A similar relationship holds for the fusion of a solid. As the left-hand side of the equation is positive, it follows that the two terms of the right-hand side must both be positive or both negative. In the case of all vaporization $dp/d\mathrm{T}$ is positive, i.e. an increase in pressure results in an increase of the temperature of vaporization and therefore the volume of the vapour v_2 is greater than the volume of the liquid v_1. In some cases of the fusion of a solid, however, $dp/d\mathrm{T}$ is negative, an increase of pressure lowering the melting-point, and the volume of the liquid is less than the volume of the solid. The most notable case is that of ice and water. With some solids $dp/d\mathrm{T}$ is positive and the solid has a smaller volume than the liquid. Such solids cannot be satisfactorily cast, as they shrink on solidifying and do not fill the moulds.

9. KINETIC ENERGY

In addition to energy changes due to heat transfer, a substance may possess kinetic energy due to its velocity which may be transformed into other forms of energy. A gas or a liquid flowing in a closed channel of varying section experiences an interchange of kinetic and pressure energy. For a perfect fluid having no viscosity, and hence experiencing no frictional resistance, the transformation is given by Bernoulli's theorem. Neglecting the weight of the fluid and variation in height, if u is its velocity, p the pressure, and ρ the density, then

$$\frac{u^2}{2} + \int\frac{dp}{\rho} = \text{constant}.$$

For liquids which are nearly incompressible the density may be assumed constant and independent of p, so that

$$\tfrac{1}{2}\rho u^2 + p = \text{constant}.$$

A liquid flowing through a pipe, the cross section, velocity, and pressure at one point being a_1, u_1, and p_1 and at another point a_2, u_2, and p_2, experiences a varying pressure and velocity according to the equations

$$\begin{aligned} p_1 + \tfrac{1}{2}\rho u_1^2 &= p_2 + \tfrac{1}{2}\rho u_2^2; \\ a_1 u_1 &= a_2 u_2, \\ \therefore u_1 &= \sqrt{\frac{2(p_1 - p_2)}{\rho\left\{\left(\frac{a_1}{a_2}\right)^2 - 1\right\}}}. \end{aligned}$$

The velocity can hence be measured by the difference of pressure between the two points, and the weight flowing in unit time obtained from the simple relationship $w = \rho a_1 u_1$. An actual fluid, however, experiences frictional resistance and there is a degeneration of some energy into heat. The destruction of kinetic energy under circumstances in which it cannot be transformed into pressure energy likewise results in its reduction to heat.

In the case of a gas, the density depends upon the pressure, the relationship between the two being determined by the heat characteristics of the flow. For adiabatic flow $p \propto \rho^\gamma$, and for isothermal flow $p \propto \rho$. With large flows and high velocities attained in short distances, the former is more nearly correct, and the equations of the flow if $p = k\rho^\gamma$ are

$$\begin{aligned} \frac{u^2}{2} + \frac{\gamma}{\gamma - 1} k^{1/\gamma} p^{\frac{\gamma-1}{\gamma}} &= \text{constant}, \\ a\rho u &= \text{constant}. \end{aligned}$$

A case of special interest is that in which the first cross section is so large that the velocity may be considered negligible. An example is that of a gas issuing from a large vessel through an orifice of cross section a; then

$$\begin{aligned} \frac{\gamma}{\gamma - 1} k^{\frac{1}{\gamma}} p_1^{\frac{\gamma-1}{\gamma}} &= \tfrac{1}{2}u_2^2 + \frac{\gamma}{\gamma - 1} k^{\frac{1}{\gamma}} p_2^{\frac{\gamma-1}{\gamma}}, \\ \therefore u_2^2 &= \frac{2\gamma}{\gamma - 1}\frac{p_1}{\rho_1}\left\{1 - \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}}\right\}. \end{aligned}$$

The weight flowing out in unit time,

$$\begin{aligned} w &= a\rho_2 u_2 = a\left(\frac{p_2}{p_1}\right)^{\frac{1}{\gamma}}\rho_1 u_2 \\ &= a\left(\frac{p_2}{p_1}\right)^{\frac{1}{\gamma}}\sqrt{\frac{2\gamma}{\gamma - 1}\rho_1 p_1\left\{1 - \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}}\right\}}. \end{aligned}$$

If the pressure p_2 is diminished the value of w reaches a maximum when

$$\frac{p_2}{p_1} = \left(\frac{2}{\gamma + 1}\right)^{\frac{\gamma}{\gamma-1}}.$$

The value of u_2 is then

$$u_2 = \sqrt{\frac{2\gamma}{\gamma + 1}\frac{p_1}{\rho_1}}$$

and

$$w = a\left(\frac{2}{\gamma + 1}\right)^{\frac{1}{\gamma-1}}\sqrt{\frac{2\gamma}{\gamma + 1}p_1\rho_1}$$

This is the maximum weight which can flow through the orifice. If the pressure p_2 is still further diminished the velocity increases but the weight flowing in unit time decreases. The maximum velocity is attained with no external pressure, and is equal to

$$u_2' = \sqrt{\frac{2\gamma}{\gamma - 1}\frac{p_1}{\rho_1}}.$$

This is greater than the velocity of sound under the conditions within the vessel in the ratio $\sqrt{\frac{2}{\gamma - 1}}$.

10. CHEMICAL ENERGY

In applying any mechanical or thermodynamical formula to a substance, attention must be paid to the possibility of chemical changes occurring, particularly when large temperature ranges are used with complex gases. When combustion occurs heat is added, and the action is evident not only by the energy changes but by the permanent change in the constitution of the substances. With many gases subjected to high temperatures, dissociation occurs with accompanying absorption of heat, but the gases reunite when low temperatures are restored and heat is emitted. There is thus no permanent evidence of the chemical reaction, but the efficiency of a working cycle is affected by the difference in temperature at which the heat is absorbed and formed. The degree of dissociation in certain common cases is illustrated by the following table:

Carbon dioxide. $2CO_2 \rightleftharpoons 2CO + O_2$.

Temperature.		Amount Dissociated (Atmospheric Pressure).
2550° F.	....	less than 0·1 per cent
3100	....	0·23 per cent
3650	....	2·1 ,,
4550	....	17·0 ,,
5450	....	55·1 ,,

Steam. $2H_2O \rightleftharpoons 2H_2 + O_2$.

Temperature.		Amount Dissociated (Atmospheric Pressure).
2750° F.	....	0·03 per cent
3650	....	0·57 ,,
4550	....	4·1 ,,
5450	....	13·0 ,,

Similar chemical reactions which take place under the influence of high temperature and reverse when low temperature is restored are:

$$CO + H_2O \rightleftharpoons CO_2 + H_2,$$
$$C + CO_2 \rightleftharpoons 2CO.$$

The heat of formation for some common compounds is given in the equations below. This heat is absorbed when the compounds are dissociated and reproduced when association is restored. The values are those for reactions under constant atmospheric pressure at 60° F. If the products are retained at a higher temperature, their heat content, whether sensible or latent, is not available for external purposes. The free heat of formation at constant pressure is less than that at constant volume by the amount of the external work,

$$H_p = H_v - p\delta v.$$

If the number of gaseous molecules after the reaction is less than that before, while the original pressure and temperature are maintained, there will be a diminution in volume, in which case the free heat at constant pressure will be greater than that at constant volume.

1 mol. C (12 lb.) + 1 mol. O_2 (32 lb.) = 1 mol. CO_2 (44 lb.) + 176,000 B.Th.U.

1 mol. C (12 lb.) + ½ mol. O_2 (16 lb.) = 1 mol. CO (28 lb.) + 53,000 B.Th.U.

1 mol. H_2 (2 lb.) + ½ mol. O_2 (16 lb.) = 1 mol. H_2O (liquid) (18 lb.) + 123,000 B.Th.U.

1 mol. H_2 (2 lb.) + ½ mol. O_2 (16 lb.) = 1 mol. H_2O (gas) (18 lb.) + 103,000 B.Th.U.

1 mol. S (32 lb.) + 1 mol. O_2 (32 lb.) = 1 mol. SO_2 (64 lb.) + 128,000 B.Th.U.

1 mol. CS_2 (76 lb.) + 3 mol. O_2 (96 lb.) = 1 mol. CO_2 (44 lb.) + 2 mol. SO_2 (128 lb.) + 478,000 B.Th.U.

1 mol. C (12 lb.) + 2 mol. S (64 lb.) = 1 mol. CS_2 (76 lb.) − 35,000 B.Th.U.

1 mol. CH_4 (16 lb.) + 2 mol. O_2 (64 lb.) = 1 mol. CO_2 (44 lb.) + 2 mol. H_2O (liquid) (36 lb.) + 382,000 B.Th.U.

1 mol. C (12 lb.) + 2 mol. H_2 (4 lb.) = 1 mol. CH_4 (16 lb.) + 39,500 B.Th.U.

11. CONDUCTION OF HEAT

In an engine, heat is added to a system by chemical combustion, and in the cases just considered heat is added or withdrawn by chemical reactions. With an external source of heat the transfer occurs by conduction or radiation. Heat is likewise withdrawn from the system by conduction and radiation. In an efficient system heat flow and heat production must take place only when and where it is intended, and must be reduced to a minimum under other circumstances.

When heat is flowing through a material and a steady flow has been reached, if $d\mathrm{T}/dx$ is the temperature gradient in the direction of flow, A the area perpendicular to the flow, $d\mathrm{H}/dt$ the flow in unit time, and K the co-efficient of heat conductivity of the material,

$$\frac{d\mathrm{H}}{dt} = -\mathrm{KA}\frac{d\mathrm{T}}{dx}.$$

If the heat flow has not reached a steady state either because the material itself has not acquired its steady temperature or because the source of heat is varying,

$$\frac{d\mathrm{T}}{dt} = \frac{\mathrm{K}}{s\rho}\frac{d^2\mathrm{T}}{dx^2},$$

where s is the specific heat and ρ the density of the material. The coefficient $\mathrm{K}/s\rho$ is termed the diffusivity or the thermometric conductivity, and the time required for the material and the heat flow to take up a steady state increases as this coefficient decreases. In most engineering problems heat has to pass from a liquid, gas, or vapour into the material, and the conditions at the surface are very different to those within the body of the material. There is a thin film over the surface consisting of oxidized or dirty metal, oil, condensed vapour, or stagnant gas, or all these at the same time. The thickness of the film and its character depend upon the temperature and the velocity of the fluid from which the heat is being abstracted. In general, its resistance to heat conduction is many times that of any metal and it is the determining factor for the amount of heat transmitted. As the thickness of the film cannot be measured, its conductivity can only be expressed by a coefficient which includes the heat conductivity and the thickness. If heat is passing through a material of heat conductivity K and thickness x from a medium at temperature T_1 to a medium at temperature T_2, and the surface co-efficients are β_1 and β_2 respectively,

$$\frac{d\mathrm{H}}{dt} = \frac{\mathrm{A}(\mathrm{T}_1 - \mathrm{T}_2)}{\frac{1}{\beta_1} + \frac{x}{\mathrm{K}} + \frac{1}{\beta_2}}.$$

In cases where the material is a metal of small thickness, K/x being large in comparison to β_1 or β_2,

$$\frac{dH}{dt} = \frac{A(T_1 - T_2)\beta_1\beta_2}{\beta_1 + \beta_2}.$$

With a wall of insulating material, the surface resistances become of less importance. The dependence of β upon the velocity may be expressed approximately by an equation of the form $\beta = a + b\rho v$, where the term a gives the conduction for stationary fluid which takes place by gradual diffusion, while $b\rho v$ is a term introduced by the formation of eddies which bring fresh fluid continually in contact with the surface. As the quantity of fluid giving up or receiving heat is proportional to the velocity, the temperature change being proportional to $a/\rho v + b$ becomes practically constant for varying velocities above the value which make a/v negligible.

12. RADIATION AND ABSORPTION OF HEAT

In addition to the heat received by the wall by direct contact from adjacent fluid or other material, heat may be received by radiation. The dissipation of the heat after passing through the wall may likewise be by radiation. According to Stefan's law the radiation from a perfectly black body is expressed by the relation

$$H = EA(T_1^4 - T_2^4),$$

where T_1 is the absolute temperature of the body, T_2 that of the surroundings, A the area, and E a constant. The value of E for a perfectly black body when the temperatures are measured in degrees Fahrenheit is $1{\cdot}73 \times 10^{-9}$ B.Th.U. per square foot per hour. The radiation from other surfaces is smaller than this, diminishing according to the degree of polish. Some values of E for other surfaces are given below. The radiation from the centre of furnaces and flames approaches that from a black body.

Substance.	Surface.	Temperature Range. F.	E.
Black.	Dull.	—	$1{\cdot}73 \times 10^{-9}$
Glass.	Smooth	60	1·6
Brass.	Dull.	100 to 650	0·37
Lamp black.	Smooth	30 to 120	1·6
Copper.	Lightly polished.	100 to 550	0·3
Wrought iron.	Dull, oxidized.	60 to 700	1·6
Wrought iron.	Polished.	90 to 230	0·6
Wrought iron.	Highly polished.	110 to 500	0·5
Cast iron.	Rough, oxidized.	110 to 500	1·6
Basalt.	Dull, ground smooth.	140 to 400	1·2
Mortar.	Rough, white.	50 to 200	1·55
Slate.	Dull, ground smooth.	140 to 400	1·2
Sandstone.	Dull, ground smooth.	140 to 400	1·05

The perfectly black body absorbs all radiation which falls upon it, while other surfaces absorb in the same proportion as their emissive powers to that of the black body. Hence if heat is radiated from one body in proportion to its emissive power E_1 and the heat falls upon a second surface, the amount absorbed by the latter is proportional to $E_1E_2/1{\cdot}73$ and also to the area subtended by the second surface. If the two areas are A_1 and A_2, the distance between them d and the inclinations of the normals to the line joining them α_1 and α_2,

$$H = \frac{E_1E_2}{1{\cdot}73 \times 10^{-9}} \frac{A_1A_2 \cos\alpha_1 \cos\alpha_2}{d^2\pi} (T_1^4 - T_2^4).$$

13. FRICTION

Waste of energy and the production of undesired heat occurs wherever there is friction. Solid friction between two rubbing surfaces is proportional to the pressure between them and independent of the velocity. It can be greatly reduced by lubrication of the surfaces to a minimum value, below which little further improvement can be made. The friction of liquids or gases flowing past surfaces or in tubes is a more complicated problem. For small velocities the fluid follows fixed paths, one particle following another along definite lines of flow. Under these circumstances the frictional resistance is proportional to the velocity of the fluid and to its viscosity. Above a certain critical velocity the flow becomes turbulent and there are no fixed lines of flow. The resistance is then approximately proportional to the square of the velocity and is dependent only on the density and not on the viscosity of the fluid. As the velocity is still further increased, the resistance follows this simple law more and more nearly.

For the flow of liquids or gases in pipes, the complete law of resistance can be represented by the following formula, where p is the pressure difference between the ends of the tube of length l and diameter d, v the mean velocity, and ρ and μ the density and viscosity coefficient of the fluid.

$$p = \frac{4l\rho v^2}{d} F\left(\frac{vd\rho}{\mu}\right),$$

F is a numerical coefficient depending on $vd\rho/\mu$ and is given in fig. 4. Over the range where $vd\rho/\mu$ is small, e.g. with small velocities or small diameter pipes, F changes very rapidly if any one of the quantities v, d, ρ, or μ is changed. This is the range over which stream-line flow occurs, and F has been calculated theoretically to be $8\mu/vd\rho$. Hence for this range

$$p = \frac{32l\mu v}{d^2} = \frac{128l\mu Q}{\pi d^4},$$

where Q is the quantity flowing in unit time. As the quantity μ changes greatly with temperature in the case of liquids, there is a corresponding

change in the loss of head. The break in the curve represents the critical conditions above which turbulence occurs, and the value for F has only been found experimentally. It diminishes for the lower increases of $vd\rho/\mu$ and becomes approximately constant for higher values. When this stage is reached,

$$p \propto \frac{l\rho v^2}{d}$$

$$\propto \frac{l\rho Q^2}{d^5}.$$

In the case of a liquid p may possibly be measured as head of the liquid itself i.e. $p = \rho gh$, when

$$h = K\frac{lQ^2}{d^5}.$$

As an example of the quantities involved, if h is in feet of head, l length in

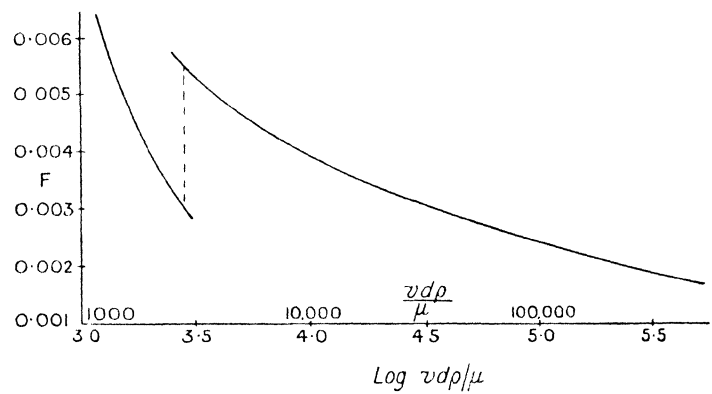


Fig. 4.—Flow of Fluids in Pipes of Circular Section

Values of F in formula $p = \frac{4l\rho v^2}{d} F\left(\frac{vd\rho}{\mu}\right)$

feet, d diameter in inches, Q in gallons per hour, then for a flow of water the diminishing value of K and its approach to constancy are shown in the following table.

$Q/d = 100$	....	$K = 0{\cdot}46 \times 10^{-6}$
250	....	0·385
500	....	0·315
1000	....	0·27
2000	....	0·23
5000	....	0·21

Although the viscosity of petrol is less than half that of water, the value of K is very little less, being 0·40 for $Q/d = 100$ and 0·195 for $Q/d = 2000$.

The frictional resistance of bends, taps, and valves may be very great. With pipes of small bore an additional resistance equivalent to that of a length of pipe from 5 to 50 times the diameter may be added by a rectangular bend, according as it is smooth or sharp. The resistance of taps, due to the bore through the barrel being often smaller than that of the pipe in which they are fitted, may be equal to that of many feet of the piping.

CHAPTER II

Fuels and their Combustion

Although there are in nature many possible sources of energy, including the water-power of the rivers and the tides, solar heat, and atomic energy, yet, with the exception of a very small percentage, the energy utilized by man is obtained from the combustion of various kinds of fuel. The principal combustible elements of fuels are carbon (C) and hydrogen (H), but most fuels also contain oxygen (O). Small proportions of sulphur (S) which is combustible may also be present; nitrogen (N), which usually acts only as a diluent; and other elements such as iron (Fe) and calcium (Ca), which remain in combinations as a solid residue. The elements are present in the fuels in many diverse combinations. The maximum amount of heat (except for certain unusual and unstable compounds) would be obtained from given amounts of combustible elements if they were uncombined with other elements. The combustion of a fuel in which they appear as compounds produces a less amount of heat, the difference being equivalent to the heat of formation or disruption of the compounds.

Fuels may be divided into natural and artificial fuels. The former are provided directly by nature and are made ready for use by mechanical or chemical treatment. The latter are manufactured from the former by chemical or physical conversion. The principal natural fuels are wood, peat, lignite, coal, oil shale, petroleum, and natural gas. Artificial fuels include charcoal, coke, coal and coke gases, petroleum distillates and oils, alcohol, benzene, and other products. Although metals are not used primarily as fuels, their oxidation in certain processes liberates considerable heat, which assists that produced by the combustion of ordinary fuel.

1. CALORIFIC VALUE

Combustion is the process of oxidation of the fuel, usually by means of the oxygen in the air. The amount of heat liberated by the oxidation of a given weight is independent of the amount of air, provided this exceeds the minimum amount for the complete oxidation. An excess of air reduces the temperature attained, which may be of importance for the purpose to which the heat is to be applied, but the quantity of heat is unaffected. The entropy of the combustion gases is increased according to the reduction in temperature.

An approximation to the amount of heat that would be liberated by the

combustion of 1 lb. of fuel can be calculated if the proportions of the elements present are known. If the combustion takes place at constant pressure and sufficient air is present to ensure the complete combustion of the hydrogen, carbon, and sulphur to water (H_2O), carbon dioxide (CO_2), and sulphur dioxide (SO_2) respectively, then the heat produced from 1 lb. of each of these elements is: H, 62,000 B.Th.U.; C, 14,600 B.Th.U.; S, 4000 B.Th.U. If the weights of these elements in 1 lb. of fuel are h lb., c lb., and s lb., the amount of heat produced per pound of fuel if they were uncombined would be:

$$H_{max.} = 62{,}000h + 14{,}600c + 4000s.$$

In the case of fuels which consist of chemical compounds of known constitution and in known proportion, the heat of formation of these compounds may be known and subtracted; thus

$$H = H_{max.} - H.F.$$

If, for example, h_1 lb. of the hydrogen is in the form of alcohol, C_2H_6O, and h_2 lb. is in the form of ether, $C_4H_{10}O$, then the heat of formation for these two compounds is $17{,}500h_1 + 12{,}600h_2$. If the total carbon content is also accounted for by these two compounds, no further deduction is necessary.

In many cases, however, the analysis of a fuel consists only of the proportions of the elements present and the compounds which they form are not known. The calorific value can then be calculated, though with less exactitude, by assuming that the oxygen present is already in combination with the hydrogen and that only the excess of hydrogen is available for combustion. As 8 lb. of oxygen unite with 1 lb. of hydrogen, the available hydrogen is thus $h - \frac{o}{8}$, where the weight of oxygen in 1 lb. of the fuel is o lb. The calorific value would thus be

$$H = 62{,}000\left(h - \frac{o}{8}\right) + 14{,}600c + 4000s.$$

If the products of combustion are reduced to the original temperature of the fuel and air (usually ordinary atmospheric temperature), the whole of this heat, except for a negligible increase in heat content referred to in a later paragraph, will be abstracted from them. In many cases, however, it is not possible to so reduce the final temperature, and the products of combustion retain some of this heat partly as sensible heat and partly as latent heat of the steam formed. The above value is known as the higher calorific value of the fuel, while the term lower calorific value is used when a correction is deducted for the heat retained by water vapour. The sensible heat of the other gaseous products is usually small and is neglected. If w is the weight of water vapour formed when 1 lb. of fuel is burnt, its sensible and latent heat when it remains as vapour is $1100w$ B.Th.U. The lower calorific value is hence

$$H_1 = H - 1100w.$$

Experimental methods of determining the calorific value of fuels are

dealt with in a later chapter on the measurement of the quantities involved in processes of heat engineering.

The sulphur content, if any, provides a negligible proportion of the total calorific value, which is determined therefore by: (1) the amount of inert matter present, such as water, ash, or nitrogen; and (2) the proportion of the hydrogen and carbon in the remaining portion. As an example, a specimen of brown coal briquettes consisting of 55·3 per cent C, 4·5 per cent H, 24·4 per cent O, 0·5 per cent S, 9·5 per cent H_2O, and 5·8 per cent ash has a calorific value of 9000 B.Th.U. per pound. If the water and ash are removed the remainder has a calorific value of 10,600 B.Th.U. per pound. As the quantity of water and ash may be often due to accidental circumstances during preparation and storage, and may also vary between preparation and utilization owing to methods of transit, it may in some cases be advisable to base tests of a comparison of fuels on the character of the " nuclear " substance after the removal of water and ash. With liquid fuels, the proportion of incombustible substance is usually small if sufficiently high temperature conditions are attained.

In the previous discussion on the calorific value of a fuel, the amount of heat which could be abstracted was found on the assumption that the combustion occurred at constant pressure and that the products of combustion were restored to the original atmospheric temperature. Even in the case of the combustion of a gas the final volume will usually be different from the original volume and external work will have been performed. Thus,

$$C_mH_n + \left(m + \frac{n}{4}\right)O_2 = mCO_2 + \frac{n}{2}H_2O.$$

If the fuel C_mH_n is a liquid or solid, and the water vapour is condensed, the ratio of initial to final volume is $\left(m + \frac{n}{4}\right)/m$, while if the water vapour remains such, the ratio is $\left(m + \frac{n}{4}\right) \Big/ \left(m + \frac{n}{2}\right)$. In the combustion of a gaseous fuel the two ratios are $\left(1 + m + \frac{n}{4}\right)/m$ and $\left(1 + m + \frac{n}{4}\right) \Big/ \left(m + \frac{n}{2}\right)$. If the volume variation per pound of fuel due to molecular variation is δv and the diminution due to water condensation is $\delta_1 v$ while the initial pressure is p, then the calorific value at constant volume can be derived from that at constant pressure by the expressions:

$$\text{Higher value: } H_v = H + p\delta v;$$
$$\text{Lower value: } H_{1v} = H_1 + p\delta v.$$

In the second expression the value of the external work done during evaporation of the water, viz. $p\delta_1 v$, has already been deducted with internal latent heat and sensible heat in the evaluation of H_1.

$$\begin{aligned} H_{1v} &= H - \delta U - p\delta_1 v + p\delta v \\ &= H - \delta U + p(\delta v - \delta_1 v). \end{aligned}$$

As an example of the magnitude of this external work, the simple case of hydrogen may be considered:

$$H_2 + \tfrac{1}{2}O_2 = H_2O.$$

Two pounds of hydrogen occupying 360 c. ft. unite with 16 lb. of oxygen occupying 180 c. ft. to form water which as vapour occupies 360 c. ft. all at N.T.P. The available heat is 124,000 B.Th.U., but if the original volume of 540 c. ft. had been maintained the heat would have been less by an amount

$$\frac{180 \times 14{\cdot}7 \times 144}{778} = 490 \text{ B.Th.U.}$$

This represents 0·4 per cent of the heat released. If the water is condensed so that the final volume is negligible an additional 980 B.Th.U. are added due to external work representing 0·8 per cent. The heat given up in internal energy as latent heat and sensible heat, however, is very much greater. The 18 lb. of water in condensing liberate $1100 \times 18 = 20{,}000$ B.Th.U., of which the 980 B.Th.U. form a part. This represents 17 per cent of the heat of combustion, and it is lost if the water is not condensed. With complex fuels these values diminish according to the hydrogen content.

Even where the products of combustion are restored to the original temperature conditions, their heat content has been changed owing to the change in specific heat. If the combustion of the hydrogen has been conducted at (say) 60° F. while the datum temperature is 32° F.,

$$\begin{aligned} \text{Heat content before combustion} &= \{2 \times 3{\cdot}405 + 16 \times 0{\cdot}242\}28 \\ &= 298 \text{ B.Th.U.} \\ \text{Heat content after combustion} &= 18 \times 1 \times 28 \\ &= 504 \text{ B.Th.U.} \\ \text{Difference} &= 206 \text{ B.Th.U.} \end{aligned}$$

This, again, is less than the accuracy with which the total heat can be measured.

In the case of gaseous fuels, it is often more convenient to state the calorific value in B.Th.U. per cubic foot instead of per pound. The molecular weight in pounds of all gases occupies 360 c. ft. at 32° F. and 14·7 lb. per square inch, or 380 c. ft. at 60° F. Hence the calorific value in B.Th.U. per cubic foot is given by the following expression, where M is the molecular weight and H the calorific value in B.Th.U. per pound:

$$h = \frac{HM}{380}.$$

Most gaseous fuels are composed of mixtures of the following gases in differing proportions:

$$H_2,\ CO,\ CO_2,\ CH_4,\ C_2H_2,\ C_2H_4,\ N_2.$$

The calorific value of the mixture and the air required for combustion can be calculated as in the following example.

Constituent.	*v*. Vol., c. ft.	*h*. B.Th.U. per c. ft.	*hv*.	M.	M*v*.	Oxygen Required, c. ft.
CO	0·10	325	32·5	28	2·8	0·05
H_2	0·45	325	146·0	2	0·9	0·225
CH_4	0·35	1000	350·0	16	5·6	0·70
CO_2	0·05	—	—	44	2·2	—
N_2	0·05	—	—	28	1·4	—
Total	1·0	—	528·5	—	12·9	0·975

Therefore, calorific value of gas = 528·5 B.Th.U. per cubic foot,

equivalent molecular weight = 12·9,

density = $\frac{12·9}{380}$ lb. per cubic foot at 60 °F. and 14·7 lb. per square inch,

= 0·034,

density relative to air = $\frac{0·034}{0·0762}$ = 0·445,

air required for 1 c. ft. of gas = $\frac{0·975}{0·21}$ = 4·64 c. ft.,

mixture strength by weight = 4·64/0·445 = 10·4/1.

If an excess of air is supplied, say 7 c. ft. to 1 c. ft. of gas, the constitution of the products of combustion would be:

CO_2 = 0·10 + 0·35 + 0·05	= 0·50 c. ft.
Water vapour = 0·45 + 0·70	= 1·15 ,,
N_2 = 0·05 + 0·79 × 7	= 5·58 ,,
O_2 = 0·21 × 7 − 0·975	= 0·495 ,,
Total	7·725 ,,

There is thus a decrease in volume from a total of 8 c. ft. to 7·725 c. ft., i.e. 0·275 c. ft., if the water remains as vapour. If it is condensed there is a further decrease to 6·575 c. ft. In the latter case the higher calorific value of 528·5 B.Th.U. per cubic foot can be extracted. If the water remains gaseous, its weight being 1·15 × 18/380 = 0·0545 lb., the lower calorific value that can be utilized is

528·5 − 1100 × 0·0545 = 468·5 B.Th.U. per cubic foot.

2. CONDITIONS FOR EFFICIENT COMBUSTION

The completeness of the combustion of a fuel depends upon the intermingling of its molecules with those of the combining air. With a gaseous fuel the required contact between the oxygen atoms of the air and the hydrogen and carbon atoms of the fuel is usually obtained without difficulty. The hydrogen atom has a greater tendency to combine with the oxygen than has the carbon atom, and fuels containing higher hydrogen content burn more easily and give higher calorific values. Liquid fuels are first gasified before combustion. In a series of chemical compounds, the boiling-point rises with decreasing hydrogen content. Thus pentane (C_5H_{12}) boils at 97° F., while octadecane ($C_{18}H_{38}$), to which paraffin oil approximates, boils at 600° F. With many heavy hydrocarbons such as ethylene (C_2H_4), acetylene (C_2H_2), and benzene (C_6H_6), if the combustion air is not completely diffused throughout the vapour, the effect of heat on the parts weak in air is to produce a reduction to more simple substances such as methane, hydrogen, and carbon. The two former may burn when the air is brought into contact with them, but some of the carbon may be deposited on the sides of the containing chamber and remain unburnt. With solid fuels a high ratio of surface to volume increases the speed and improves the completeness of the combustion. Thus coal dust burns more readily than solid carbon if both fuels are similar in constitution. The combustion of a fuel of complex constitution, as are most natural fuels, therefore takes place in the following order:

1. Gaseous parts.
   - (*a*) Hydrogen.
   - (*b*) Hydrocarbons with large proportion of hydrogen.
   - (*c*) Hydrocarbons with small proportion of hydrogen.
2. Vaporized parts from solid or liquid fuel.
   - (*a*) Hydrocarbons with large proportion of hydrogen.
   - (*b*) Hydrocarbons with small proportion of hydrogen.
3. Solids with high surface-volume ratio followed by solids with small surface-volume ratio.
   - (*a*) Hydrocarbons with large proportion of hydrogen.
   - (*b*) Hydrocarbons with small proportion of hydrogen.
   - (*c*) Carbon.

A furnace should therefore be constructed so that the early burning of the gaseous parts may produce distillation of volatile contents, while the solid part is raised in temperature by the heat of combustion of both gas and distillates. The increase in temperature thus produced assists the combination of the oxygen with the fuel, and is particularly necessary with the solid portion.

As an example of the difference produced by reducing a fuel to small

pieces, the following figures were obtained for the residual coke produced by heating lump coal and powdered coal from the same source in a platinum crucible to a gentle heat.

Anthracite coal in lumps produced	97·8	per cent coke, in powder,	93·9	per cent.
Forge	,,	,,	95·2	,,	,,	87·6	,,
Bituminous	,,	,,	86·0	,,	,,	81·1	,,
Gas	,,	,,	67·9	,,	,,	66·5	,,

In the first two cases the quantity of gas produced was thus considerably greater with the pulverized fuel than with lump fuel. The proportionate increase in gas production due to pulverizing is less in the case of bituminous coal, but the lump coal had a blistered surface with comparatively large area. With the gas coal there was little effect produced by reducing to powder as the original coal was in small lumps and was porous.

Although the rate of combustion of coal, which is simply the oxidation of its carbon and hydrogen constituents, is greatly accelerated by increase of temperature, nevertheless it takes place to some extent at all temperatures. When coal is stored in large quantities and exposed to the atmosphere, the oxidation of a very small proportion will suffice to raise its temperature owing to the small cooling surface in comparison to the bulk. The raising of the temperature increases the rate of oxidation and the action may even progress until inflammation occurs. The greater part of the action is between the atmospheric oxygen and the hydrogen, and the loss of calorific value may be important even when there is no great rise in temperature. With inferior fuels of high water content, the heat may produce some drying, in which case it is not altogether a disadvantage.

3. ATTAINABLE TEMPERATURE

In the case of internal-combustion engines or with explosives, the heat is not abstracted from the products of combustion, at any rate in the first instance. The most efficient engine would be that in which all the heat was retained by the products. If 1 lb. of fuel with a calorific value at constant pressure of H B.Th.U. requires w_o lb. of air for its combustion, the initial and final temperatures being $t_1°$ F. and $t_2°$ F., then the heat content before and after combustion is given by

$$H + (s - C_{p1}w_o)(t_1 - 32) = wL + (1 + w_o)C_{p2}(t_2 - 32),$$

where s is the specific heat of the fuel and C_{p1} that of the air between 32° F. and $t_1°$ F., C_{p2} is the mean specific heat of the products of combustion between 32° F. and $t_2°$ F., while w is the water vapour formed and L its latent heat. A similar formula applies for combustion at constant volume:

$$H_v + (s - C_{v1}w_o)(t_1 - 32) = wL + (1 + w_o)C_{v2}(t_2 - 32).$$

When high temperatures are attained the value of C_{v2} depends upon the

temperature, the specific heat of many gases increasing appreciably owing to dissociation. With combustion at constant volume, the ratio of initial to final pressures is given by

$$\frac{p_1}{p_2} = \frac{R_1(t_1 + 460)}{R_2(t_2 + 460)}.$$

The ratio of the constants R_1/R_2 is the ratio of the number of gaseous molecules before and after, and it is affected both by the chemical action of combustion already referred to, and by the dissociation at high temperatures.

In no practical case is it possible for the combustion products to retain the whole of the heat of combustion, and it is equally impossible to abstract from them the whole of their heat. The gases escape to waste, carrying a certain amount of heat dependent upon the temperature and heat capacity. On a commercial scale it is usually necessary to use an amount of oxygen or air in excess of that required for the combustion, as the technical conditions render it impossible to bring every oxygen molecule into contact with a carbon, hydrogen, or sulphur molecule heated to ignition temperature and ready for combination. The greater the amount of excess oxygen or air used, the greater the amount of heat carried away by the waste gases. The excess should therefore be made as small as possible. Although this is a general rule, modification may be desirable in certain cases, such as in the use of weak mixtures for internal-combustion engines, where the efficiency is not determined solely by the heat in the waste gases or where the volume is not constant as assumed.

In order to obtain a quantitative idea of the maximum temperature possible, the case of an internal-combustion engine burning the gaseous fuel referred to in Section I (p. 29) may be taken as an example. In the ideal case combustion takes place at constant volume and during the process there is no heat loss. In addition to the fuel and air mixture, a residue of combustion products is retained from a previous combustion. If v_1 is the volume of any constituent and C_{v1} its mean specific heat at constant volume *per unit volume* between 32° F. and its initial temperature t_1 before the combustion, while v_2 and C_{v2} refer to the volume and specific heat *per unit volume* between 32° F. and the final temperature t_2, the calorific value of the fuel at constant volume being H_v per unit volume, then for the combustion of unit volume of gas,

$$H_v + (t_1 - 32)\Sigma v_1 C_{v1} = (t_2 - 32)\Sigma v_2 C_{v2}.$$

Now the mass of unit volume of a gas, 1 c. ft., is $M/380$ under standard conditions, where M is the molecular weight in pounds, so that the specific heat per unit volume,

$$C_v = \frac{M}{380} C_v,$$

where C_v is the specific heat of unit mass, which is the more usual manner of measuring specific heat. Hence

$$380H_v + (t_1 - 32)\Sigma v_1 M_1 C_{v1} = (t_2 - 32)\Sigma v_2 M_2 C_{v2}.$$

Continuing the example already utilized, a mixture of 1 c. ft. of gas with 7 c. ft. of air may be used in a gas-engine, a residue of 1·5 c. ft. of combustion gases being retained. The residue will contain $1{\cdot}15 \times 1{\cdot}5/7{\cdot}725$ c. ft. of water vapour, $0{\cdot}5 \times 1{\cdot}5/7{\cdot}725$ of CO_2, and $6{\cdot}075 \times 1{\cdot}5/7{\cdot}725$ of $N_2 + O_2$. Assuming that the temperature before ignition is 600° F., the left-hand side of the equation is obtained from the summation of the following table.

Constituent.	Volume, v_1	M_1C_{v1}	$v_1M_1C_{v1}$
$CO + H_2 + O_2 + N_2$	$0{\cdot}1 + 0{\cdot}45 + 0{\cdot}05 + 7{\cdot}0 + \frac{6{\cdot}075}{7{\cdot}725}1{\cdot}5 = 8{\cdot}780$	5·0	43 90
CO_2	$0{\cdot}05 + \frac{0{\cdot}5}{7{\cdot}725}1{\cdot}5 = 0{\cdot}147$	7·76	1·14
H_2O	$\frac{1{\cdot}15}{7{\cdot}725}1{\cdot}5 = 0{\cdot}223$	6·16	1·37
CH_4	$0{\cdot}35 = 0{\cdot}35$	8·00	2·80
Totals	9 50	—	49 21

$$(t_1 - 32)\Sigma v_1M_1C_{v1} = 568 \times 49{\cdot}21 = 27{,}900$$
$$380H_v = 380 \times 468{\cdot}5 = 178{,}000$$
$$\text{Total} \quad .. \quad .. = 205{,}900$$

For the right-hand side of the equation it is necessary to have some idea of the resulting temperature in order to insert the correct mean specific heat between 32° F. and the temperature. Assuming that the temperature is about 3600° F., then the following table is obtained.

Constituent.	Volume, v_2	M_2C_{v2}	$v_2M_2C_{v2}$
$O_2 + N_2$	$6{\cdot}075 + \frac{6{\cdot}075}{7{\cdot}725}1{\cdot}5 = 7{\cdot}254$	5·70	41·35
CO_2	$0{\cdot}5 + \frac{0{\cdot}5}{7{\cdot}725}1{\cdot}5 = 0{\cdot}597$	10·30	6·14
H_2O	$1{\cdot}15 + \frac{1{\cdot}15}{7{\cdot}725}1{\cdot}5 = 1{\cdot}373$	8·27	11·36
Totals	9·224	—	58·85

$$205{,}900 = (t_2 - 32)58{\cdot}85,$$
$$t_2 = 3560° \text{ F.}$$

A second approximation, using the mean specific heats between 32° F. and the neighbourhood of 3560° F., furnishes the corrected answer,

$$t_2 = 3570° \text{ F.}$$

The ratio of the pressures after and before combustion is

$$\frac{p_2}{p_1} = \frac{3570 + 460}{600 + 460} \times \frac{9{\cdot}224}{9{\cdot}500} = 3{\cdot}7.$$

4. MIXTURE STRENGTH AND EXHAUST GAS LOSS

If the temperature t at which the used combustion gases are allowed to escape is measured, then the heat lost can be calculated in a similar manner. From the constitution of the fuel and the amount of air, the products of combustion can be found by simple chemical equations. Their heat content can then be estimated as in the last table from the expression $(t - 32)\Sigma v M C_p$, where C_p is the specific heat at constant pressure if the gases escape at the original pressure. The proportion of the calorific value thus lost is given below for typical fuels and mixture strengths. As the excess air co-efficient λ increases, so also does the heat loss owing to the greater mass of exhaust gases at the temperature of escape.

λ	CO_2 Contents of the Gases from Analysis.	Combustion Temperature. ° F.	Heat Loss for an Exhaust Gas Temperature. ° F.		
			950	550	200
Carbon ; 14,600 B.Th.U. per pound.					
1	0·210	4200	0·181	0·103	0·029
2	0·105	2375	0·346	0·199	0·055
3	0·070	1730	0·511	0·295	0·082
Coal; 3% H_2O, 5% H_2, 75% C; 13,500 B.Th.U. per pound.					
1	0·181	3960	0·193	0·110	0·031
2	0·089	2340	0·363	0·207	0·058
3	0·059	1670	0·534	0·304	0·086
Lignite; 36% H_2O, 3% H_2, 40% C; 6500 B.Th.U. per pound.					
1	—	3490	0·231	0·136	0·045
2	—	2135	0·414	0·244	0·080
3	—	1560	0·596	0·352	0·113
Blast-furnace Gas; 60% N_2, 8% CO_2, 29% CO, 3% H_2; 99 B.Th.U. per cubic foot.					
1	0·236	2900	0·279	0·158	0·044
2	0·159	2135	0·399	0·227	0·063
3	0·121	1705	0·518	0·295	0·082

5. EXHAUST GAS ANALYSIS AND MIXTURE STRENGTH

The importance of maintaining the correct mixture strength between fuel and air is evident from the preceding table. In an internal-combustion engine it is of particular importance. The amount of fuel which can be burnt in the limited volume of the cylinders being determined by the fixed amount of air which they will contain, a strong mixture results in waste of fuel while a weak mixture results in smaller power for the volume and weight of engine in use. Owing to the difficulty of measuring large and irregular flows of air, it is not often possible to measure the mixture strength directly. It can be obtained, however, indirectly by analysis of the exhaust gases.

If 1 lb. of fuel contains h lb. of hydrogen, c lb. of carbon, and o lb. of oxygen, the weight of oxygen required for combustion in pounds is

$$32\frac{c}{12} + \frac{16}{2}h - o = 2\tfrac{2}{3}c + 8h - o.$$

At 60° F. and 14·7 lb. per square inch, the molecular weight of oxygen in pounds, viz. 32 lb., occupies 380 c. ft., so that the volume of oxygen required is

$$\frac{380}{32}(2\tfrac{2}{3}c + 8h - o) \text{ c. ft.}$$

The oxygen of the air is 21 per cent by volume, the air required at 60° F. and 14·7 lb. being therefore

$$\frac{380}{32 \times 0{\cdot}21}(2\tfrac{2}{3}c + 8h - o) \text{ c. ft.}$$
$$= 151c + 452h - 56{\cdot}50 \text{ c. ft.}$$

If this volume of air is v_0 and that actually used is v, the excess air being $v - v_0$,

$$\text{excess air coefficient } \lambda = \frac{v}{v_0},$$

$$\text{air factor } \eta = \frac{v_0}{v}.$$

$$\text{Percentage of excess air } \alpha = \frac{100(v - v_0)}{v_0}.$$

Considering first the case of weak mixtures, if the *volumes* of the flue gases resulting from the combustion of 1 lb. of fuel are denoted by the capital letters CO_2, O_2, N_2,

$$O_2 + N_2 = v_0(\lambda - 0{\cdot}21).$$

Where the water is condensed from the gases, the ratio of dry combustion gases to the air supplied

$$= 1 + \eta\left(\frac{c}{12v_0} - 0{\cdot}21\right).$$

If $\frac{h - \frac{o}{8}}{c} = a$, the relative volumes of the combustion gases are given by

$$CO_2 = \frac{0{\cdot}21}{\lambda + 3(\lambda - 0{\cdot}21)a},$$

$$O_2 = \frac{0{\cdot}21(\lambda - 1)(1 + 3a)}{\lambda + 3(\lambda - 0{\cdot}21)a},$$

$$N_2 = \frac{0{\cdot}79\lambda(1 + 3a)}{\lambda + 3(\lambda - 0{\cdot}21)a}.$$

As one molecule of oxygen becomes one molecule of carbon dioxide when it unites with carbon in complete combustion, the volume when reduced to the initial temperature and pressure conditions is unchanged. Hence the maximum value of the carbon dioxide plus oxygen in the combustion gases cannot exceed 21 per cent of the volume of the gases. This maximum value occurs in the combustion of pure carbon and if hydrogen or sulphur are present in the fuel, the proportion of carbon dioxide plus oxygen in the exhaust gases must be less than this. The maximum percentage of carbon dioxide alone is $21/\lambda$.

The mixture strength of a combustible mixture of fuel and air is usually denoted by the weight of air divided by the weight of fuel. For a weak mixture the number is high and for a strong mixture it is low. For perfect combustion, as oxygen is 23 per cent by weight of air, the weight of air required is

$$m_0 = \frac{1}{0{\cdot}23}(2\tfrac{2}{3}c + 8h - o).$$

With excess air

$$m = m_0\lambda.$$

The mixture strength can hence be deduced from a knowledge of the constitution of the fuel and the value of λ. The latter can be obtained from the composition of the combustion gases.

$$\left.\begin{aligned}
\lambda &= 0{\cdot}21\frac{\frac{1}{CO_2} + 3a}{1 + 3a} \\
&= 0{\cdot}21\frac{\frac{1 - CO_2}{O_2} - 1}{0{\cdot}21\frac{1 - CO_2}{O_2} - 1} \\
&= 1 + \frac{0{\cdot}79}{0{\cdot}21\frac{1 - CO_2}{O_2} - 1} \\
&= \frac{1}{1 - \frac{0{\cdot}79}{0{\cdot}21}\frac{O_2}{N_2}}.
\end{aligned}\right\}$$

With deficient air, less than that required for perfect combustion, some carbon monoxide, CO, would be formed in place of carbon dioxide and there would be no unburnt oxygen. Similar expressions could be derived. This method of obtaining the mixture strength requires a knowledge of the fuel constitution. Another method is to perform a number of combustions with various mixture strengths which have been measured by fuel and air measurements, and also to analyse the combustion gases in each case. The results obtained can then be used for the same fuel burning under the same conditions, the combustion gas analysis indicating the unknown mixture strength. Fig. 1 shows a chart for determining the strength of the petrol-air mixture

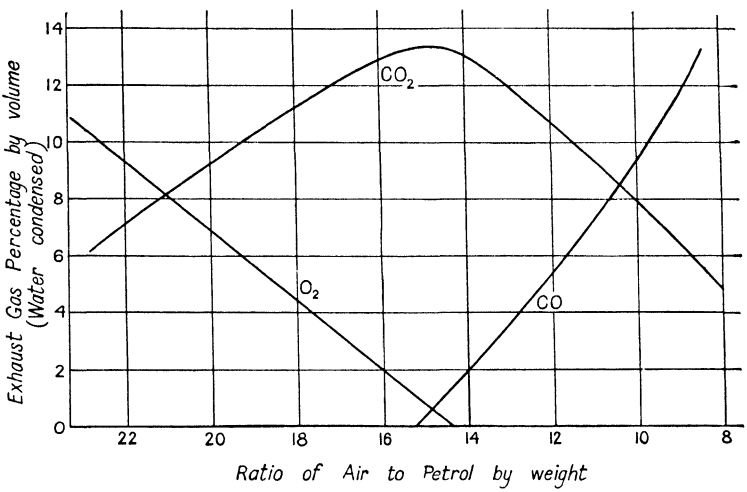


Fig. 1.—Chart for determining the strength of Petrol-air Mixture entering an Internal-combustion Engine

entering an internal-combustion engine, from an analysis of the exhaust gases.

The diagram of fig. 1 must only be used under the conditions by which it was constructed. Under other conditions the same mixture strength could result in different products. For example, a correct mixture strength giving production of CO_2 only at low temperatures would result in CO and O at high temperatures by dissociation. Diagrams have been constructed which represent the character of the combustion gases for various mixture strengths and under all combustion conditions. Such a diagram for coal containing 75 per cent C, 6 per cent H_2, and 8 per cent O_2 is shown in fig. 2. Ordinates represent the percentage of CO_2 in the combustion gases and abscissæ the percentage of O_2, the water being assumed condensed. With no oxygen or carbon monoxide, the exact amount of air required being provided, the percentage of CO_2 is 18 per cent, represented by the point C. With an infinite amount of air ($\lambda = \infty$, $\eta = 0$) the CO_2 would be negligible and the oxygen would be that in the air, 21 per

cent, represented by the point B. The line CB shows the increase of the CO_2 and diminution of O_2 as the air is reduced from a very large excess to the correct proportion, no monoxide being formed in any case. Now if the whole of the 18 per cent of CO_2 underwent dissociation, the total volume of combustion gas would increase and the CO would be about $16\frac{1}{2}$ per cent and the O_2 about $8\frac{1}{4}$ per cent. This is shown at the point D, which is on

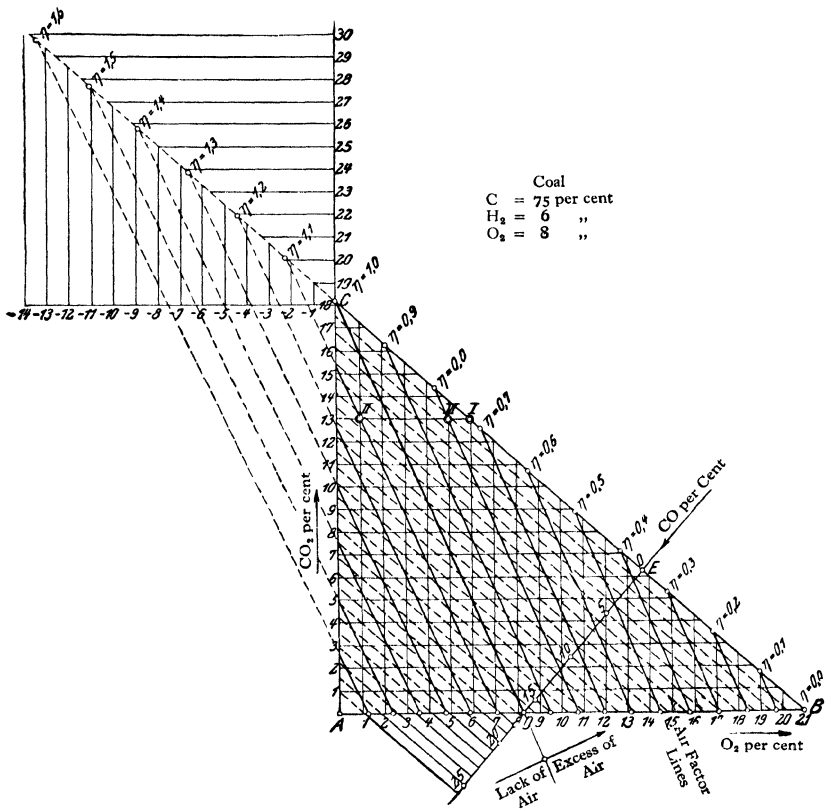


Fig. 2.—Exhaust Gas Diagram for Coal

the line $\eta = 1$ passing through C, the O_2 being given by the abscissæ as before and the CO by the perpendicular distance from CB, the line of no CO. At any point on the line between C and D there is CO_2, CO, and O all present, measured by the distances from AB, CB, and AC respectively. The same holds for the parallel lines for other mixture strengths.

A similar diagram is shown in fig. 3 for blast-furnace gas containing 2·6 per cent H_2, 32·5 per cent CO, 58·7 per cent N_2, and 6·2 per cent CO_2. As the original gas contains CO_2, the base line is inclined from the point D at 6·2 per cent to the point B at 0 per cent.

Instead of a number of plane diagrams, one for each kind of fuel, a model in three dimensions can be constructed which will give the data relative to fuels of varying composition. The relationship between mixture strength and combustion gas analysis can also be represented by means of nomograms. Further details of these methods can be found in *Beiträge zur graphischen Feuerungstechnik*, by W. Ostwald.

The combustion gas analysis thus not only supplies information on the

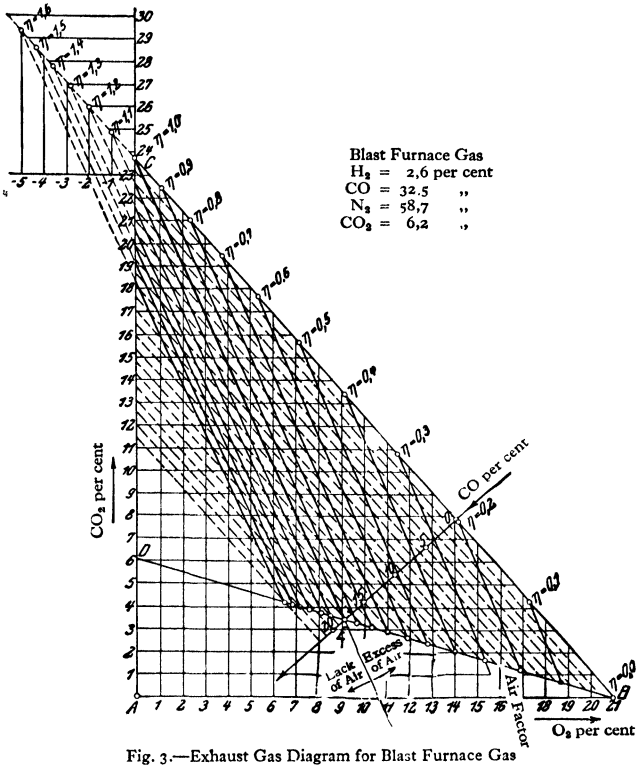


Fig. 3.—Exhaust Gas Diagram for Blast Furnace Gas

chemical conditions of the combustion, making it possible to determine whether or not perfect combustion and the maximum heat production have been obtained, but also indicates the physical condit[illegible]s to mixture strength and waste gas losses.

6. IGNITION TEMPERATURE AND VELOCITY

Although the combination of the elements of a fuel with oxygen may take place at a very slow rate at all temperatures, combustion as usually understood requires the previous attainment of what is known as the ignition

temperature. The heat of combustion of one part of the fuel serves to raise and to keep the remainder at the temperature, and combustion continues as long as the temperature is maintained and as long as there is a sufficient supply of oxygen. Gasification also takes place during the heating of the fuel, and usually there is present both gaseous combustion and combustion of solid carbon. The former results in the production of flame, while with the direct combustion of solid a glowing manifests itself without production of flame. The gases distilled from the fuel, such as methane and ethylene, may be decomposed before combustion according to the temperature attained, if conditions for rapid combustion are not present. The products of combustion also, carbon dioxide and water vapour, may be decomposed if the temperature is sufficiently high, the amount of heat released being reduced by the decomposition.

If the ignition temperature has been reached and combustion initiated at one point of a fuel, it then spreads at the ignition velocity. This velocity is of great importance, particularly in gaseous combustion. In a slow combustion its value is determined by the heat conductivity, specific heat, density, and temperature of the substance. In a rapid combustion a local rapid rise of temperature and pressure results in a compression of the unburnt portion with consequent temperature rise. Sufficient quantities of oxygen must, of course, be present throughout if ignition is to spread. Values of ignition velocities have been found by Le Chatelier and Mallard. It has been found

	Volume of Gas / Total Volume	Initial Pressure, Atmospheres Absolute.	Ignition Velocity, Ft./Sec.
Hydrogen—air Initial temp., 60° F. H_1 = 265 B.Th.U. per cubic foot	0·103	0·5	1·98
	0·101	2·5	1·66
	0·181	0·5	14·4
	0·182	2·5	20·1
	0·240	0·5	26·9
	0·244	2·5	45·6
Coal-gas—air Initial temp., 160° F. H_1 = 485 B.Th.U. per cubic foot	0·08	0·5	1·22
	0·08	2·0	1·36
	0·11	0·5	5·1
	0·11	3·5	4·7
	0·16	0·5	13·2
	0·16	3·5	11·9
Generator gas—air Initial temp., 160° F. H_1 = 133 B.Th.U. per cubic foot	0·30	0·5	1·72
	0·30	7·5	0·49
	0·465	0·5	7·2
	0·465	5·5	6·5

that the ignition velocity in open pipes is different from that in a closed bomb. Combustion in an open furnace therefore differs from that in an engine cylinder. Propagation in a moving eddying mass of gas is also different from that in a gas at rest. The value depends upon the initial pressure and mixture strength, and may reach a value of 60 ft. per second. This may be contrasted with the explosion velocity, i.e. the rate at which an explosive rise in pressure is transmitted, which attains a value of 3000 ft. per second.

The ignition temperature for certain pure gases with oxygen is given below according to Dixon.

	° F.
Hydrogen	1083
Carbon monoxide	1176 to 1216
Methane	1075 to 1093
Acetylene	780 to 825
Ethylene	932 to 967

From Wollers and Ehmke, for

Benzol vapour	1040 to 1058
Oil gas from light oil	1138 to 1202
Oil gas from tar oil	1193

The ignition temperatures of some common fuels at atmospheric pressure in air as given by Holm are:

	° F.
Vegetable oils	750
Brown coal-tar oil	750 to 1020
Coal-tar oil	1110 to 1160
Petroleum	715
Benzene	780 to 860
Alcohol	950
Benzol	970
Paraffin	590
Air-dried peat	535
Brown coal	480
Bohemian coal	735
Anthracite	825
Gas coke	750

The character of the combustion products and the ignition velocity are not only dependent upon the mixture strength, but combustion cannot be maintained at all unless the mixture strength lies within certain limits. If the mixture is either too strong or too weak burning will not take place. These limits for gaseous fuel are given below in percentage of gas to total mixture, by volume.

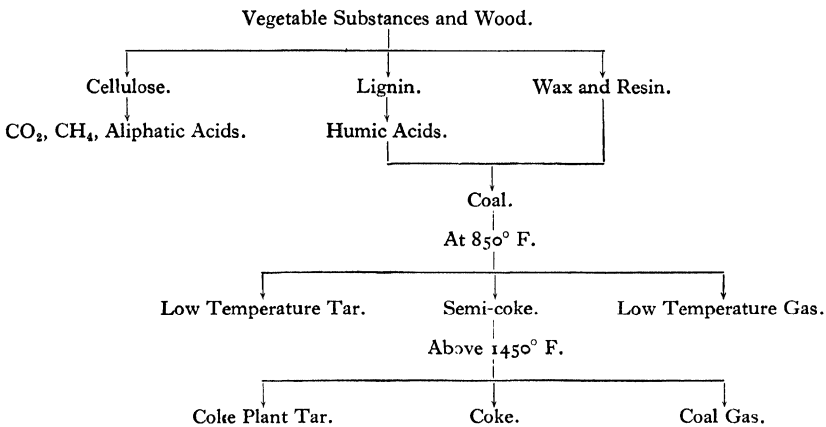
Carbon monoxide,	between	16·6 per cent	and	74·8 per cent
Hydrogen,	,,	9·5 per cent	,,	66·3 per cent
Water-gas,	,,	12·5 per cent	,,	66·6 per cent
Coal-gas,	,,	8·0 per cent	,,	19·0 per cent
Acetylene,	,,	3·5 per cent	,,	52·2 per cent
Benzene,	,,	2·5 per cent	,,	4·8 per cent

For liquid fuels in an internal-combustion engine the range of explosive mixture by weight is:

Petrol	.. ..	between $\frac{1}{8}$ and $\frac{1}{20}$
Benzol	.. ..	„ $\frac{1}{5}$ „ $\frac{1}{18}$
Alcohol	.. ..	„ $\frac{1}{4}$ „ $\frac{1}{20}$

7. SOLID FUELS

The composition of all vegetable substances may be divided into three parts, cellulose, lignin, wax and resin. From the cellulose, carbon dioxide, methane, and aliphatic acids are formed, while lignin is partly converted to humic acid. The derivation and derivatives of coal may be represented roughly by the following diagram.



This table may represent all classes of coal, but its composition and the proportions of the derivatives depend upon the time of formation, pressure, temperature, the presence of water, and the possibility of gases escaping. The variation in composition is illustrated by the following average data.

	Carbon. Per cent.	Wax and Resin. Per cent.
Wood	50	2·0
Peat, young	56	4·2
Peat, old	61	4·2
Brown coal, young ..	69	5·7
Brown coal, old ..	77	5·5
Gas coal	81	4·8
Bituminous coal ..	85	4·0
Non-bituminous coal ..	92	2·5
Anthracite	97	0·5

The proportions of the elements for certain of the fuels in the series are given in the following table. The figures refer to fuel after the expulsion of water and elimination of ash. A comparison of the two tables shows that in each case considerable variation of composition occurs.

	Percentage by Weight.				Atoms with 100 Atoms C.		Calorific Value in B.Th.U. per Pound.
	C.	H.	O.	N.	H.	O.	
Cellulose ..	44·4	6·2	49·4	—	166·7	83·3	7,550
Wood ..	50·0	5·9	44·0	0·1	139·3	66·0	8,100
Peat ..	56·9	6·5	35·0	1·5	138·4	48·4	9,350
Brown coal ..	60·9	5·5	37·7	1·0	90·4	49·7	11,700
Coal ..	85·0	4·9	9·0	1·1	67·5	9·0	14,400
Anthracite ..	95·3	1·9	2·3	0·5	23·8	1·8	15,300

The fifth and sixth columns show the gradual expulsion of the hydrogen and oxygen during the process of formation.

The main characteristics of some solid fuels are given in the table overleaf in which coal is classified according to its source. In the table below coal is differentiated according to its volatile constituents, beginning with anthracite, in which the content of these is low. The differentiation is based upon the result of a coking experiment. The character of the flame varies with the volatile constituents, and the conditions required in the efficient utilization of the various types of coal vary accordingly. The coke residue after distillation has also properties which vary greatly. Five main types of coal are differentiated and shown in the table below.

CLASSIFICATION OF COAL

Type of Coal.	Formation, per Cent.	O/H.	Coke after the Distillation, per Cent.	Specific Gravity of the Coke.	Condition of the Coke.
Non-bituminous coal or anthracite with a short flame.	90·0 to 93·8 C 4·5 to 4·0 H 5·5 to 3·0 O	1	82 to 92	1·35 to 1·41	Fritted or in the form of powder.
Bituminous coal with a short flame.	88·0 to 91·0 C 5·5 to 4·5 H 6·8 to 5·5 O	1	74 to 82	1·30 to 1·35	Smelted, very compact, slightly fissured.
Bituminous coal (forge coal).	84·0 to 89·0 C 5·5 to 5·0 H 11·0 to 5·5 O	1	68 to 74	1·30	Smelted until it is average, compact.
Bituminous coal with a long flame.	80·0 to 85·0 C 5·8 to 5·0 H 14·2 to 10·2 O	2 to 3	60 to 68	1·28 to 1·30	Smelted, but greatly fissured.
Dry coal with a long flame.	75·0 to 80·0 C 5·5 to 4·5 H 12·5 to 15·0 O	3 to 4	50 to 60	1·25	Powder caked together.

Solid Fuels

Fuel.	Air-dry Substance in 100 parts by weight.							Substances free from Water and Ashes in 100 parts by weight.				100 parts of Fuel give			Calorific Value, B.Th.U. per pound.	Calorific Value, B.Th.U. per pound of Combustible Substance.
	Carbon.	Hydrogen.	Oxygen and Nitrogen.	Sulphur.	Water.	Ashes.	Combustible Substance in 100 parts by weight.	Carbon.	Hydrogen.	Oxygen and Nitrogen.	Sulphur.	Coke Yield.	Solid Carbon.	Volatile Constituents.		
Carbon	100	—	—	—	—	—	100	100	—	—	—	—	100	—	14,600	14,600
Sulphur	—	—	—	100	—	—	100	—	—	—	100	—	—	—	4,000	4,000
Oak	50·2	6·0	43·4	—	—	0·4	—	—	—	—	—	—	75·0	24·6	8,300	8,350
Beech	49·1	6·1	44·2	—	0·6	—	—	—	—	—	—	—	75·0	26·4	8,600	8,650
Fir	50·4	5·9	43·4	—	—	0·3	—	—	—	—	—	—	75·0	24·7	9,050	9,100
Pine	50·3	6·2	43·1	—	—	0·4	—	—	—	—	—	—	75·0	24·6	9,150	9,200
Ruhr coal	80·0	4·7	6·0	1·5	1·3	6·5	92·2	86·8	5·1	6·5	1 6	77·0	70·0	22 0	13,750	14,900
Saar, Silesian, Saxony coal	75·0	5·0	10·0	1·0	2·5	6·5	91·0	82·5	5·5	11·0	1·0	64 0	57·0	33·0	12,800	14,000
Upper Bavarian coal ..	53·0	4·0	12·0	5·0	9 0	17·0	74·0	71·6	5·4	16·2	6 8	56 0	38·0	35·0	9,350	12,750
Coal briquettes ..	82·0	4·2	3·7	1·2	1·7	7·2	91·1	90·0	4 6	4·1	1·3	83 0	76·0	15 0	13,950	15,300
Gas coke	84·0	0·8	3·4	1·0	1·8	9·0	89·2	94·2	0·9	3·8	1·1	96·0	88·0	2·0	12,600	14,100
Westphalian anthracite	85·4	3·8	4·7	1·2	1·0	3·9	95·2	89 7	4·0	4·9	1·3	89·3	85·0	9·7	14,350	15,100
Bohemian brown coal ..	49·2	3·8	14·0	0·3	27·4	5·5	67·0	73·0	5 6	20·6	0·6	—	—	—	7,900	11,700
Saxony brown coal ..	32·1	2·6	12·2	1·1	46·2	5·8	48·0	67 0	5 4	25·5	1·1	25·0	28·0	27 0	5,000	11,900
Brown coal briquettes ..	52·0	4·3	16·0	2·0	17·0	9·0	74·0	70 0	5·8	21·5	2·7	40·0	32·0	43·0	8,650	11,750
Peat	44·0	4·5	25·0	0·5	20·0	6·0	74·0	59·4	6·1	33·8	0·7	31·0	25·0	49·0	6,850	9,700
Peat	10·4	1·2	8·7	0·4	22·4	56·9	20·7	50·2	5·8	42·0	2 0	9·0	—	10 0	6,000	7,650
Charcoal	75·5	2·5	12·0	—	9·0	1·0	90·0	83·9	2·8	13·3	—	—	—	—	12,600	14,000
Esthonian oil shale ..	29·7	3·5	11·6	2·2	15·0	38·0	47·0	63·1	7·5	24·6	4·8	14·0	—	33 0	5,950	16,300
Lignite	28·8	2·5	9·6	2·9	40·4	15·8	43·8	65·7	5·7	21·9	6·7	—	—	—	4,300	9,900
Württemberg oil shale	—	—	—	—	2 0	88·0	10·0	—	—	—	—	5 0	—	5·0	2,350	18,000

Coal may thus also be classified according to the character of the coke it produces, while for commercial purposes each class may be subdivided according to size into large coal, nuts, slack, and mixed coal.

Oil shales show very large variation in combustible content and calorific value according to their source. The proportion of gas, tar, and carbon in them varies between 10 and 70 per cent. The gas and tar is usually extracted, the carbon remaining in the residue, which has recently been utilized in the manufacture of Roman cement. The use of other fuels of low calorific value is becoming increasingly important. In addition to such fuels which occur naturally, e.g. peat and coals with high ash and water content, there are residues of low-value fuels in the grading of high-value fuels, e.g. coal dust and coke dross. Refuse from houses and many manufactures has also a calorific value which is not now allowed to be wasted.

The direct combustion of solid fuels furnishes a supply of heat at a minimum cost, but it has certain disadvantages. The handling of the fuel requires either more manual labour or more complicated machinery than with liquid or gaseous fuel. The temperatures attained are highest immediately above the fuel and fall off towards the more distant parts of the apparatus to be heated. In order to attain rapid combustion it is usually necessary to have high air velocities over the fuel and an excess of air above that theoretically required for complete combustion. This excess of air results in lower temperatures. Pulverized fuel has some advantage over lump fuel in this respect. The higher ratio of surface to volume enables the air and fuel to intermingle and combine with a minimum amount of excess air.

8. LIQUID FUELS

The properties of some liquid fuels are given in the table on the following page.

With liquid fuels it is possible to have a very fine atomization, i.e. a breaking up into small drops, thus increasing greatly the surface at which fuel and oxygen particles are brought together. Complete combustion with practically no excess of air can be obtained, giving high temperatures. Furthermore, a large space can be raised to a uniform temperature throughout. With the exception of alcohol and benzol, which form only a small percentage of the total fuel used, liquid fuels are obtained from natural oils. The convenience of the liquid form has caused attempts to be made to transform solid fuel into liquid. By using pressures of 100 to 150 atmospheres and temperatures of 700° F. to 900° F., a high output of liquid fuel can be obtained from coals, with the exception of those of the anthracite type and those with low water contents. From 100 lb. of Upper Silesian coal containing 28 per cent liquid, 6 per cent ashes, and 4 per cent water, Bergius obtained 22 lb. of oil of boiling-point below 450° F., 17 lb. of oil of higher boiling-point, 16 lb. of pitch, 15 lb. of gas (mostly methane), 10 lb. of water, 6 lb. of ash, 0·5 lb. of ammonia, and a residue of 15 lb. of unchanged coal. The cost

Name of Fuel.	Specific Gravity.	Chemical Formation.				Water Content per cent.	Lower Calorific Value, B.Th.U. per pound.	Combustion Air, c. ft. per pound.	Combustion Products, c. ft. per pound.		Flash Point, ° F.
		C per cent.	H per cent.	O + N per cent.	S per cent.				CO_2	H_2O	
Alcohol, 100 per cent ..	0·794	52·17	13·04	34·79	—	—	11,500	122	17·0	25·4	—
Alcohol, 90 per cent ..	0·823	—	—	—	—	10·0	10,150	109	15·3	25·1	—
Alcohol, 75 per cent ..	0·861	—	—	—	—	25·0	8,050	91	12·8	24·4	—
Benzol	0·885	92·31	7·69	—	—	—	17,250	179	30·1	15·0	—
Horizontal furnace tar ..	1·150	89·30	4·94	5·30	0·34	5·0	14,750	146	—	—	150 to 120
Vertical furnace tar ..	1·150	89·50	6·95	3·46	0·50	3·0	15,750	150	—	—	105 to 160
Chamber furnace tar ..	1·090	88·70	6·80	4·15	0·35	1·5	15,750	149	—	—	120 to 140
Coke oven tar	—	89·00	6·10	4·50	0·40	5·0	15,650	147	—	—	—
Tar oil	1·050	90·00	7·00	2·50	0·50	1·0	16,200	160	—	—	150 to 185
Naphthalene	1·150	93·75	6·25	—	—	—	17,300	160	—	—	180
Petrol	0·700	—	—	—	—	—	18,000	205	27·4	31·2	—
Brown coal tar	0·870	86·00	11·00	2·40	0·60	2·0	17,600	176	—	—	135 to 240
Solar oil	0·830	85·50	12·30	1·38	0·83	—	18,000	173	—	—	115 to 120
Paraffin oil	0·870	85·70	11·50	0·70	1·10	—	17,600	171	—	—	150 to 250
Creosole oil	0·940	80·1	9·70	8·90	1·30	—	15,650	176	—	—	195
Shale oil	0·880	—	—	—	—	—	14,400	—	—	—	—
Californian crude oil ..	0·960	86·90	11·80	3·00	—	—	17,100 to 20,700	176	—	—	< 60
Russian crude oil ..	0·880	86·00	13·00	1·00	—	—		176	—	—	180
Mexican crude oil ..	0·943	82·70	11·47	3·56	2·27	—		176	—	—	< 60
German crude oil ..	0·939	86·00	11·00	2·00	—	—		176	—	—	75
Gas oil	0·860	86·20	12·65	1·15	—	—	17,800	176	—	—	130 to 230

of the process is at present prohibitive for commercial purposes. The distillation of coal under ordinary pressures and primarily for gas or coke production gives an extract of tar and oil, some of which it is possible to use for internal-combustion engines.

9. GASEOUS FUELS

The advantages which it is possible to obtain by using atomized liquid fuel are inherent in a gaseous fuel. Its disadvantage is its bulk when it has to be transmitted considerable distances. In a few regions of the globe natural gas is found, but the chief supply is by manufacture from coal. A portion of the heat content of the coal is used in the process of manufacture, but this loss is more than counterbalanced by the increase in convenience. For domestic heating and illuminating the advantages are obvious, but in many industrial processes it has also been found of advantage to generate gas from the coal and then to burn the gas, rather than to use direct combustion of the coal. Heat is lost but much better regulation and more uniform temperature can be attained. Coal gas is obtained by the distillation of coal in retorts at high temperatures. Generator gas is manufactured by the partial combustion of coal, with air and water present. Blast-furnace gas is the gaseous combustion product ejected from blast furnaces. This, owing to the temperature conditions, retains some calorific value. The properties of various artificial gaseous fuels are detailed in the table on the following page.

It is only in the United States that natural gas is utilized to any large extent. All natural gases have high methane content and high calorific values of the order of 800 B.Th.U. per cubic foot. Manufactured gaseous fuels are obtained from solid or liquid fuels as indicated in the table.

The manufacture of gas from liquid fuels has not yet been conducted on any large scale, though several processes are in existence. It may be noted, however, that in the carburettor of a petrol-engine the liquid is partly or wholly gasified, and that the fuel which enters the engine is mainly gaseous.

The analyses of the various generator gases show that the principal loss of calorific value in the process of gasification is due to the oxidation of carbon to carbon monoxide. The production of some carbon dioxide represents a total loss of calorific value unless the generator is so near to the furnace that the sensible heat of the generator gases is utilized. The formation of CO and CO_2 takes place according to the following equations:

$$C + O = CO + 53{,}000 \text{ B.Th.U.}$$
$$C + O_2 = CO_2 + 176{,}000 \text{ B.Th.U.}$$

The loss from the carbon constituent of the solid fuel is thus 30 per cent. The greater part of the hydrogen is obtained in the free state with no loss, water being first formed and then decomposed at high temperatures.

Gaseous Fuels

Type of Gas.	Specific Gravity, Air = 1.	Chemical Formation in Volume Percentages.						Lower Calorific Value, B.Th.U. per c. ft.	Yield from Fuel.			
		CO.	CO_2.	CH_4.	C_mH_n.	H_2.	N_2.			C. ft. per ton.	Water, per cent.	Ash, per cent.
Blast-furnace gas	0·86	28·0	8·0	—	—	—	60·7	118	—	—	—	—
Coke-oven gas ..	0·42	8·0	2·0	29·0	4·0	50·0	7·0	540	Coal	11,300	1·8	9·0
Coal gas	0·40	8·0	2·0	32·0	4·0	51·0	3·0	553	,,	12,000	2·5	6·5
Low temperature distillation gas ..	0·58	7·0	3·0	48·0	13·0	27·0	2·0	780	,,	3,500	1·3	6·5
Water gas ..	0·52	42·0	5·0	0·5	—	49·0	3·0	293	Coke	49,500	1·8	9·0
Semi-water gas ..	0·83	22·0	7·0	0·5	—	18·0	52·0	133	,,	113,000	1·8	9·0
Generator gas ..	—	23·7	5·0	1·9	—	0·5	62·9	120	Coal	—	1·3	6·5
,,	—	26·8	7·2	0·6	—	18·4	47·0	152	Anthracite	—	1·0	4·0
Mond gas ..	0·82	11·6	16·0	3·8	—	24·4	44·2	150	Waste coal	—	11·1	21·5
,,	0·81	11·4	18·4	3·3	—	23·3	43·6	139	Peat	—	48·5	1·2
Generator gas ..	0·85	26·0	6·2	5·1	—	4·3	58·4	150	Wood	—	?	?
,,	—	30·6	5·7	8·1	—	6·1	49·5	171	Peat, Bohemian ..	—	32·3	3·3
,,	—	25·2	5·1	2·3	—	16·5	50·9	158	Brown coal, German	—	30·0	7·0
,,	—	22·7	10·5	1·7	—	22·7	42·4	133	Brown coal	—	46·8	7·0
,,	—	26·0	5·0	0·2	—	12·0	56·8	124	Smoke-box dust ..	—	2·9	19·2
,,	—	8·2	13·6	3·6	—	21·9	52·7	116	Saar waste coal ..	—	—	25·0
,,	—	21·0	9·3	1·8	—	16·3	51·6	136	Brown coal shale ..	—	15·9	19·7
,,	—	32·0	2·0	0·5	—	7·5	58·0	136	Coke	156,000	2·0	9·0
,,	—	30·0	3·7	2·0	—	10·7	53·9	139	Brown coal briquettes	111,000	15·0	9·0

10. COMBUSTION AND GASIFICATION

The production of artificial gaseous fuel accompanies and is occasioned by the process of combustion. The relative degree of the two processes depends upon the conditions of manufacture, notably upon the temperature and the availability of oxygen and water vapour. By suitable adjustment of the conditions action can be graded from perfect combustion, in which the heat production is a maximum and no calorific value remains in the products, to actions resulting in gases of various compositions and calorific values, with corresponding heat expenditure. The various chemical actions which may occur are:

$$\begin{aligned}
C + O_2 &= CO_2 + 176{,}000 \text{ B.Th.U.} \\
C + 2H_2O &= CO_2 + 2H_2 - 34{,}000 \text{ B.Th.U.} \\
C + CO_2 &= 2CO - 70{,}000 \text{ B.Th.U.} \\
2CO &= CO_2 + C + 70{,}000 \text{ B.Th.U.} \\
CO_2 + H_2 &= CO + H_2O - 18{,}000 \text{ B.Th.U.} \\
CO + H_2O &= CO_2 + H_2 + 18{,}000 \text{ B.Th.U.} \\
C + \tfrac{1}{2}O_2 &= CO + 53{,}000 \text{ B.Th.U.} \\
C + H_2O &= CO + H_2 - 52{,}000 \text{ B.Th.U.}
\end{aligned}$$

The later equations may represent intermediate stages in the first three actions, which in suitable proportions will give the result of any combustion or gasification process. The oxygen is, of course, usually introduced as a constituent of air, but the nitrogen acts only as a diluent of initial and final gases. The effect of other reactions such as those with sulphur impurities is negligible. If the first three reactions take place in proportion a, b, and c, the total is given by

$$\begin{aligned}
&(a+b+c)C + aO_2 + 2bH_2O \\
&\quad = 2cCO + (a+b-c)CO_2 + 2bH_2 + 176{,}000a - 34{,}000b - 70{,}000c.
\end{aligned}$$

The three reactions in any proportions can be represented by the triangular diagram given in fig. 4, in which the points A, B, and C indicate single reactions in the order given. Any combination of the second and third is given by a point on BC. Thus at E, where the reactions are in equal amounts, the net effect is that of a water-gas production. At the point D in AC representing the first and third reactions in equal amounts, air-gas production is given. Variation in water-gas and air-gas constitution will be shown by points in the neighbourhood of E and D respectively. The net reactions with mixed water gas and air gas are given by the point F on DE with practically no heat production or absorption. Cases in which combustion predominates must be within the triangle in the neighbourhood of A, while combustion and gasification combined will be within the area DABE. Cases of reduction of combustion gases due to high temperature and contact with carbon are shown in the neighbourhood of C. In the area HAG the net reaction is exothermal, and in CHGB it is endothermal. By the erection of ordinates perpendicular

to the area of the triangle which are proportional to the heat, a model in three dimensions can be formed. Similar models may be constructed for the reaction temperature, calorific value of products, coal and water consumption.

Triangular diagrams of the same character can be drawn to represent for

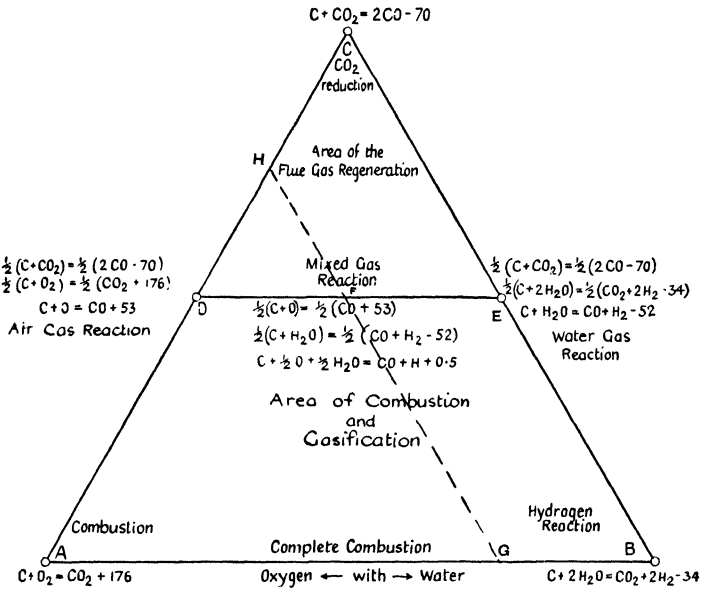


Fig. 4.—Gasification Triangle (W. A. Ostwald)

any process the constitution of products of combustion, which may consist of burnt gas, unburnt gas, and excess air in any proportions according to the conditions of the combustion process. The three corners of the triangle then give the cases in which the products consist of one only of the three possible constituents.

CHAPTER III

Plants for Combustion

In dividing the plants which are to be considered in this and the next chapter into (*a*) plants for combustion and (*b*) plants for distillation and gasification, the principal purpose of the plants determines the classification. In many cases, however, the plant serves a double purpose. For example, in a blast furnace the principal action is the combustion of coal for smelting, but blast-furnace gas may be produced which is comparable with that from special gas-producing plants. Heat may be used directly as heat, or may be transformed into another kind of energy such as mechanical energy, the blast furnace and the boiler furnace being representative of the two cases. The general factors to be considered in the production of the heat by combustion are the same for both fields of application. In many cases, however, where heat is used as heat, in addition to the combustion of the fuel by means of the air supplied there are also thermo-chemical reactions between fuel, gases, and the object of the heat application.

In the combustion of coal, the principal constituent being carbon, the maximum efficiency would be attained if the exhaust gases contained no carbon monoxide. At a low temperature, a reaction between the iron ore in an iron blast furnace and carbon monoxide may be represented by the following equation, the molecular weights being taken in pounds:

$$Fe_2O_3 + 3CO \rightleftharpoons 2Fe + 3CO_2 \pm 15{,}000 \text{ B.Th.U.} \quad .. \quad (1)$$

At a high temperature, however, CO_2 is reduced to CO by the action of iron, and a mixture of CO_2 and CO is produced, whose ratio depends upon the temperature and the reaction velocity. At low temperatures the formation of CO_2 and at high temperatures the formation of CO takes place, the reversible process $2CO = CO_2 + C$ occurring by catalysis at temperatures above 750° F. Hence the formation of CO_2 increases towards the top of the blast furnace, the final ratio of CO to CO_2 in practice varying from 2 to 3. The reduction of Fe_2O_3 with formation of CO is endothermal, and the degree to which this action takes place produces a correspondingly lower temperature.

$$Fe_2O_3 = Fe_2 + \tfrac{3}{2}O_2 - 350{,}000 \text{ B.Th.U.} \quad .. \quad (2)$$

$$Fe_2O_3 + 3C = Fe_2 + 3CO - 193{,}000 \text{ B.Th.U.} \quad .. \quad (3)$$

The free heat developed and the efficiency of the heat utilization increases

with the CO_2 content, if the blast furnace alone is considered, but as in modern plants the escaping gases are used for heating boilers and driving engines, the presence of unburnt CO exerts no decisive influence on the overall efficiency of the entire plant.

Heat is also developed in the combustion of metals and metalloids. The development of heat forms the basis of several important metallurgical processes. Among these are the Bessemer and Thomas processes in the iron industry and the blasting of copper ores in converters. Combustion is obtained by injecting air into the liquid metal, and is sufficient not only to keep the metal fluid but also to increase its temperature.

The heat production in various cases is as follows:

1 lb. of	Silicon	burnt to	SiO_2	produces	14,100	B.Th.U.
1 ,,	Manganese	,,	MnO	,,	2,970	,,
1 ,,	Iron	,,	FeO	,,	2,110	,,
1 ,,	Iron	,,	Fe_3O_4	,,	2,970	,,
1 ,,	Phosphorus	,,	P_2O_5	,,	10,600	,,
1 ,,	Sulphur	,,	SO_2	,,	4,100	,,
1 ,,	Carbon	,,	CO	,,	4,410	,,
1 ,,	Carbon	,,	CO_2	,,	14,600	,,

Impurities are to a great extent eliminated in this way by combustion when ingot iron or steel is produced from pig-iron. In the blasting of crude copper ore, the reactions occurring are as follows:

$$FeS + 3O = FeO + SO_2 + 190{,}000 \text{ B.Th.U.} \quad .. \quad (4)$$

$$Cu_2S + 3O = Cu_2O + SO_2 + 169{,}000 \text{ B.Th.U.} \quad .. \quad (5)$$

$$2Cu_2O + Cu_2S = 6Cu + SO_2 - 70{,}000 \text{ B.Th.U.} \quad .. \quad (6)$$

$$FeO + SiO_2 = FeSiO_3 + 18{,}000 \text{ B.Th.U.} \quad .. \quad .. \quad (7)$$

The crude ore may contain 40 to 55 per cent of copper and the net heat production is positive.

In the calcination of sulphur ores, the sulphur is burnt with oxygen to form sulphurous acid, while at the same time metals or metallic oxides are formed. Similar actions take place in the preparation of copper, zinc, lead, and other ores, while iron ores that are rich in sulphur are treated in order to obtain sulphuric acid. The most important ores in this class of cases are zincblende (ZnS), copper pyrites ($CuFeS_2$), galena (PbS), and iron pyrites (FeS_2).

1. COMBUSTION OF SOLID FUELS, FURNACES WITHOUT GRATES

Furnaces for the combustion of solid fuels may conveniently be divided into those with and without grates. The most notable example of the latter class is the iron blast furnace, a typical plant being shown in fig. 1. The principal parts are the blast furnace itself A, the charging apparatus B, and the regenerator C. The function of the last is to heat the air supplied to

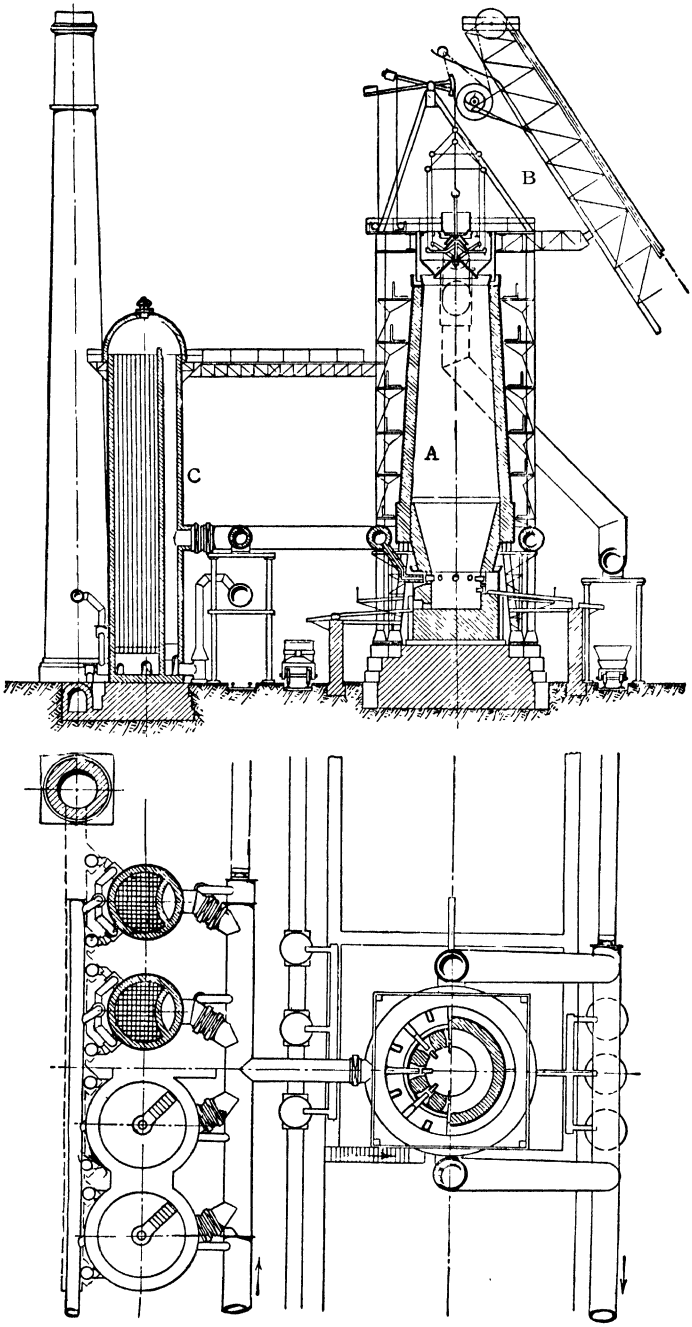


Fig. 1.—Blast Furnace Plant

the furnace to between 1250° F. and 1500° F., the air being furthermore introduced into the hottest part of the furnace. Equation (1) represents ideal conditions and any deviation from it results in inefficiency. In actual practice, a mean output of 200 tons of crude iron per day may require an equal quantity of coke and 1,000,000 c. ft. of air measured at 60° F. and atmospheric pressure, but heated and injected at an excess pressure of half an atmosphere. For heating the regenerator, 40 per cent of the exhaust may be required, the remaining 60 per cent being used under steam boilers or for power purposes.

Vertical furnaces without grates are used for many other purposes. The fuel is placed in the throat of the furnace together with the substance to be smelted, and sinks as the fuel burns and the smelted substance falls to the bottom. The air used for combustion is either drawn in from underneath or is injected under pressure, and is cold or hot according to the process. The fuel burns from the bottom, ensuring an economical utilization of the heat, all of which, except for radiation losses, reaches the substance above. The ascending gases also act upon the upper layers. The arrangements for utilizing the gas as it finally issues from the furnace depend upon its temperature and combustibility. The fuels used in these furnaces are charcoal, coke, coal, and coal dust.

An example of a cupola furnace is shown in fig. 2. The coke, which is arranged in layers between the iron, is burnt to carbon dioxide by means of the injected air so that the hot gases smelt the pig-iron. Enough air is injected to produce combustion products which are almost wholly CO_2, so that the maximum heat is obtained. The utilization of the escaping gases, which under such conditions contain only sensible heat, is discussed later. One hundred pounds of prepared castings, inserted cold in the cupola furnace, require the following amounts of coke in pounds:

Type of Casting.	For a Solid Casting.	For a Hollow Casting.
Cast iron	30	50
Malleable iron ..	140	160
Bessemer	60	90
Siemens-Martin ..	80	120
Electric steel ..	200	280
Crucible steel ..	400	500

One pound of coke produces 13,000 B.Th.U. when completely burnt to CO_2. One pound of pig-iron requires 410 B.Th.U. for liquefaction. Without heat loss, therefore, 1 lb. of pig-iron should require 410/13,000 or 0·03 lb. of coke. From the above table 1 lb. of pig-iron actually requires 30/100 or 0·3 lb. of coke. The total efficiency of the plant is therefore 10 per cent, and 90 per cent of the heat supplied is lost by conduction, radiation, and to the furnace gases. Now 1 lb. of coke requires 32/12 lb. of oxygen for complete combustion or 32/12 × 100/23 lb. of air. Together with the

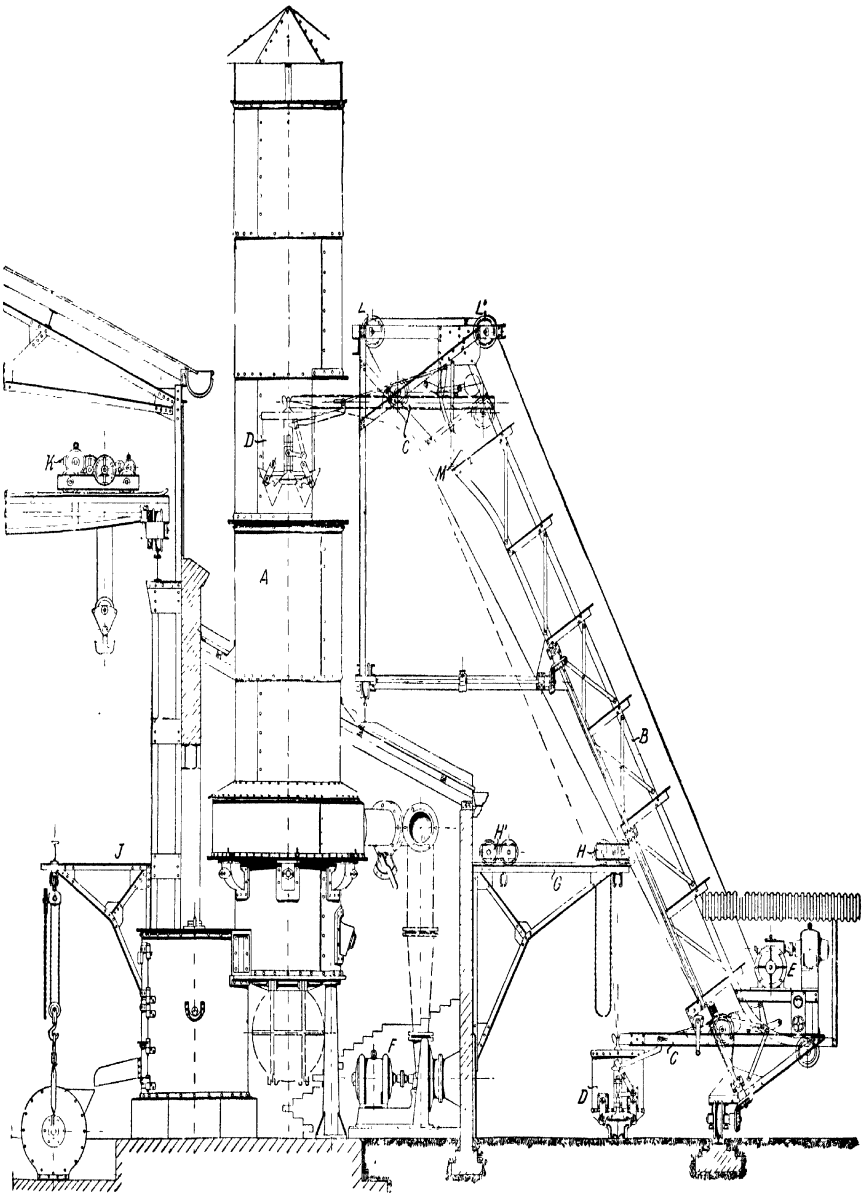


Fig. 2.—Cupola Furnace with Charging Apparatus

A, Furnace. B, Elevator. C, Elevator platform. D, Bucket. F, Winding engine. F, Blower. G, Jib crane. H, Travelling carriage on crane. J, Jib crane in foundry for removing molten metal in ladle. K, Foundry overhead travelling crane.

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1 lb. of coke, the weight of the escaping hot gases is thus 11·6 lb. To ensure complete combustion, excess of air is necessary, and if double that theoretically required is used, the quantity of hot gas per pound of coke would be 23 lb. If this leaves the furnace at 600° F., the quantity of heat carried away is 3400 B.Th.U. The heat balance per pound of melted iron is thus:

Heat supplied by 0·3 lb. of coke	= 3900	B.Th.U.
Heat of molten iron	= 400	,,
Heat of blast-furnace gases	= 1000	,,
Radiation and conduction losses	= 2500	,,

Coal is also burnt in vertical furnaces without a grate in the lime industry. The furnaces are similar in form to blast furnaces, and an example is shown in fig. 3. The removal of carbon dioxide from calcium carbonate is an endothermal process.

$$CaCO_3 = CaO + CO_2 - 77{,}000 \text{ B.Th.U.}$$

The molecular weight of $CaCO_3$ is 100, so that 1 lb. of $CaCO_3$ requires the application of 770 B.Th.U. by the combustion of the requisite amount of carbon. As 1 lb. of pure carbon in burning to carbon dioxide emits 14,600 B.Th.U., the quantity of carbon required per pound of $CaCO_3$ is 770/14,600 or 0·053 lb. The amount required per pound of CaO produced is 0·053 × 100/56 or 0·095 lb. of carbon or approximately 0·12 lb. of coal or coke. The heat required for raising the temperature of the carbonate to that of the furnace and for replacing the heat lost by radiation and conduction must also be provided. The production of this heat is estimated to require 0·18 to 0·25 lb. of coal or coke per pound of CaO produced in vertical furnaces in constant operation.

In more modern plants, annular kilns and tunnel furnaces often take the place of vertical furnaces. The air required for the combustion of the carbon is previously heated by means of the hot lime and the walls of the furnace, while the heat of the exhaust gases is used to pre-heat the substance to be burnt and the cooled furnace walls. An annular furnace for lime with a daily output of about 70 tons consists of ten to twelve compartments which are arranged in an oval ring and divided from one another by thin walls (see fig. 4). Each compartment has an opening for filling and emptying, the furnace being walled up during the burning process. The method of working is as follows. When burning is taking place in compartment 6, the compartments 1 to 5 will contain hot lime, the action having been completed in these in succession, while compartments 7 to 12 will contain unburnt limestone. The air required for combustion passes through compartments 1 to 5, becoming thereby heated more and more as it approaches 6, and cooling the burnt lime and furnace walls. After combustion in compartment 6, the hot exhaust gases pass through compartments 7 to 12, thus heating the limestone previous to burning. When the action is completed in 6, burning is commenced in 7, while compartment 1, now thoroughly cooled, is emptied of lime and filled with limestone. The fresh air now enters at 2, and the

exhaust gases leave at 1, which takes its place in the cycle after compartment 12. The larger annular kilns have two or three fires which are often of a radial shape, and daily output up to 120 tons of burnt lime. With this type

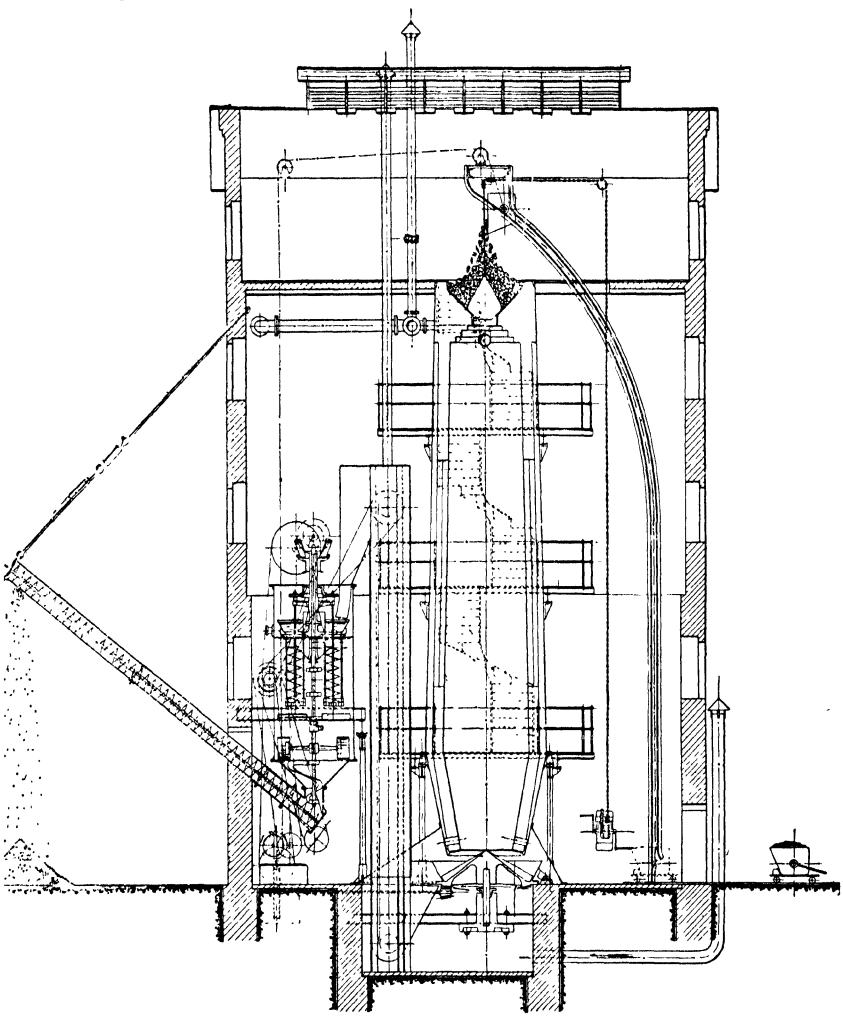


Fig. 3.—Vertical Lime Furnace with Elevators and Automatic Lime Extractor

the coal consumption falls to between 0·16 and 0·20 lb. of coal or coke per pound of burnt lime produced.

The tunnel furnace is a long furnace 50 yd. or more in length. The combustion is carried out at a point in the centre, and the substance to be burnt is moved slowly through the furnace on trucks. The air passes through the furnace in the reverse direction, so that the fresh air first meets the burnt

lime, which it cools while itself being pre-heated. After passing the place of combustion, the exhaust air passes over the unburnt limestone to which it imparts its heat. Thus both air and limestone are hot before entering the combustion portion.

Roman and Portland cement, hydraulic lime, and Puzzolana cement are burnt in vertical and annular furnaces. Roman cement is obtained by burning 60 to 70 per cent sulphurous lime with 25 per cent of clay below the sintering temperature. Portland cement is obtained from 75 per cent sulphurous lime and 25 per cent clay by burning above the sintering temperature at about 2500° F. to 2700° F. (Seger cone 14 to 18). For Portland cement, 0·18 lb. to 0·20 lb. of coke with low ash content is required per pound of burnt material, while slightly less is required for Roman cement. Portland cement and burnt lime are also prepared in revolving furnaces, which are described later. Hydraulic lime is obtained by burning limestone with 10 to 20 per cent of clay. Puzzolana cement, when not obtained in a natural state, is produced by burning siliceous stone poor in lime. Both these latter substances require less coal for burning than Roman cement.

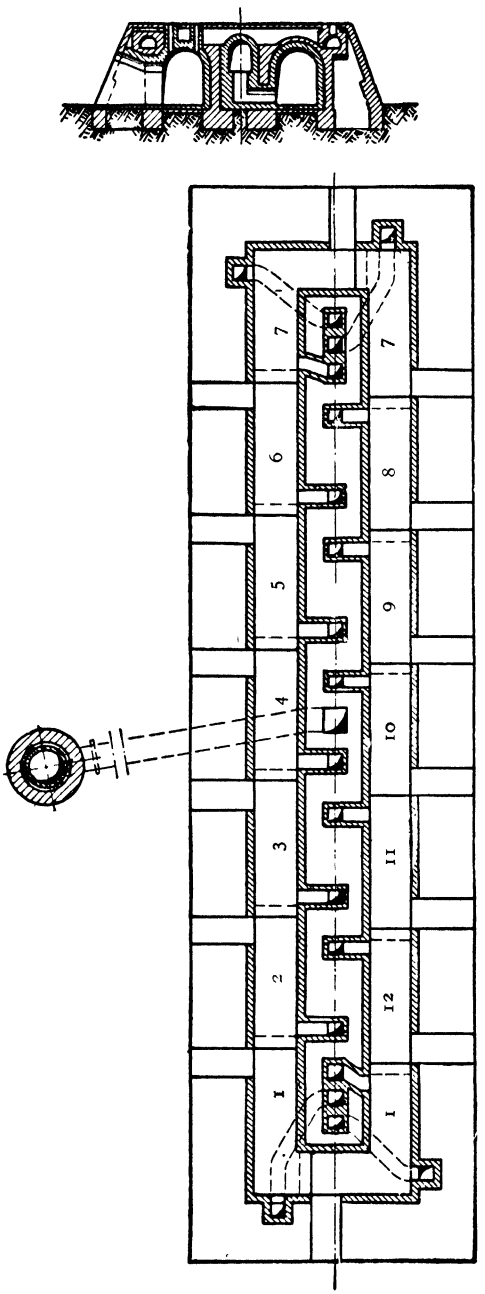


Fig. 4.—Annular Furnace for a Yearly Output of 1 million Bricks

Building bricks and firebricks are burnt in annular furnaces. Three processes are necessary to complete their manufacture: (1) removal of the water, (2) baking, (3) burning. The first is effected either by passing hot air above the annular furnace or by heating the pressed clay in a special drying compartment, heated by the waste gases of the furnace or by warm air which is heated by these gases or by the exhaust steam from the steam-engine of the manufactory. The baking takes place in the oven itself, and is used for expelling any remaining water. The expelled water must not, of course, be deposited on other cold forms. Hence when, as is described in the lime kiln, the exhaust gases are led from the burning compartment they pass over the adjacent unburnt bricks in a separate thin-walled passage, drying the bricks without coming into contact with them and the evaporated moisture. In the burning process, a reducing or oxidizing flame is utilized according to the kind of clay. House bricks are burnt at about 1800° F. (Seger cone, 05) and fireproof bricks at 3300° F. (Seger cone, 38). The amount of coal required for the burning process only, and excluding the manufacturing machinery, for 1000 bricks weighing 7000 lb., is 330 to 380 lb. of coal of calorific value 12,500 B.Th.U. per pound. With a suitable kind of clay very inferior coal and coke can be used. Fireproof brick requires 400 to 450 lb. of coal for 6000 lb. weight of brick.

Plaster of Paris, $CaSO_4 2H_2O$, is also burnt in furnaces similar to lime kilns. At 250° F., $CaSO_4 \frac{1}{2}H_2O$ is obtained, and at 340° F., $CaSO_4$. Burning at 1000° F. produces a material which does not harden with water, and it is only at 1800° F. to 2300° F. that a useful product can be obtained. The coal consumption for 1 lb. of plaster of Paris at 340° F. is about 0·05 lb., and for 1 lb. of plaster of Paris burnt to 2000° F. it is 0·10 lb. of coal. The mineral is first burnt in the vertical furnaces, and the burning process completed in muffles or on plates over a fire. During the second process, the material should be repeatedly stirred.

Of the many other important chemical processes which utilize heat in the combustion of coal without a grate, or which give off heat in chemical transformations, the preparation of superphosphate may be mentioned. Tricalcium phosphate and sulphuric acid produce monocalcium phosphate, plaster of Paris, and water when subjected to a powerful application of heat.

2. COMBUSTION OF SOLID FUELS, FURNACES WITH GRATES

The grate, on which the fuel lies, consists of plates or bars. The whole or part of the air required for combustion flows in between these bars or through holes in the plates. When bars are used, the grate is usually so constructed that all or part of the combustion residue can fall through, while the fuel can remain on the grate for the purpose of combustion. The above conditions cannot both be completely fulfilled. As anything approaching a pure combustion is only possible with coke, all other fuels giving off

certain gases in the combustion, a case of pure combustion rarely occurs, but we have instead gasification and combustion combined. The study of combustion conditions from Nusselt's formula, which applies only to pure combustion without the development of gas, produces interesting results. The formula is:

$$W = K(vT)^{0.214} V^{0.786} d^{-0.732} nt^{0.946} O_2,$$

where W is the fuel consumption, K a constant, v the diffusion velocity, T the absolute temperature, V the gas volume, d the diameter of the gas passages, n the number of passages, t the depth of the charges, and O_2 the oxygen concentration. The formula specifically applies to pure carbon permeated with vertical passages.

The higher the temperature T the greater is the rate of burning, while at the same time the influence of dissociation increases. In industrial furnaces, T is not allowed to exceed 3250° F. owing to the effect on the materials of construction. If T tends to exceed this amount, the radiated and conducted heat must be led off in order not to endanger the stability of the parts of the furnace exposed. The diffusion velocity v is increased by increase of V and the introduction of eddies. An increase of draught is equivalent to an increase of V, while n and d are functions of the shape and size of the fuel particles, increasing with the amount of exposed surface. With small fuel, the exposed surface is greater than with larger fuel. The oxygen concentration O_2 is affected by the depth of the layer t. The volatile constituents in fuels other than coke result in more rapid combustion. The mixture of the inflammable gas with oxygen is effected by diffusion. With a solid fuel, the oxygen must always be brought into contact with the surface of the fuel, and the carbon dioxide must be removed so that fresh oxygen can take its place. The oxygen must be able to penetrate through burning charge to unburnt charge, and the charge should accordingly be less deep than in the case of a fuel containing gas.

Fuels rich in gas must be arranged in layers the height of which varies according to the gas contents. The temperature of the charge remains proportionally low, as a part of the combustion heat is at first latent and only becomes free in the flame zone. Inferior fuels require a charge of greater depth, the temperature being low in this case owing to the mixture of inert matter. The vaporization of the volatile constituents is accelerated by an increase of T, but subsequent decomposition of these volatile constituents tends to check such increase. The vapours and gases permeate the rising air by means of diffusion and mechanical mixing, and burn in the flame zone. The combustion process in the flame zone is therefore dependent both on the character of the distillation process and on the mixing of the vapours with air or oxygen. An excess of oxygen shortens the combustion while a deficiency of oxygen lengthens it. A high gas velocity, which usually includes a strong eddying formation, brings about a more rapid and intensive mixture of the gas and oxygen, and thus shortens the combustion process and the flame zone. A low gas velocity with a weak eddying formation

lengthens the combustion process and the flame zone. A short or long flame can thus be obtained according to the manner in which the various factors are controlled. The former is mostly used in boiler plants where the flame must not exceed certain dimensions. The long flame is present in metallurgical and ceramic processes.

Non-combustible admixtures lower the temperature T of the fuel charge, as on the one hand they considerably retard the complete burning of the carbon by enveloping it, and on the other hand they retain to a certain extent the heat required for vaporizing the volatile constituents. The calorific value per pound of fuel containing the admixture is also less than that of a purer fuel. Thus, the calorific value of anthracite is 15,000 B.Th.U. per pound, while brown coal varies between 8000 and 11,000 B.Th.U. per pound. The quality of the fuel necessarily affects the grate loading. If high-percentage non-bituminous coal is burnt, a rapid heat withdrawal and a large excess of air are necessary for a high output of the grate. With a slow heat withdrawal and consequently a small excess of air, a small grate loading is necessary to obtain the maximum efficiency. If coal rich in gas is burnt, a normal loading is most favourable, while if low-percentage coal is burnt a high grate loading is required.

If the heat supplied to a furnace due to the calorific value of the fuel and chemical reactions is Q_1, while Q_2 is the heat lost from all causes, the efficiency of the furnace may be represented by

$$\eta = \frac{Q_1 - Q_2}{Q_1} = 1 - \frac{Q_2}{Q_1}.$$

The heat loss Q_2 is made up of the following factors: (*a*) Loss due to unburnt fuel which falls through the grate. This is small in amount, and, with flat grates, should be only from 1 to 3 per cent after the furnace has been burning for a short time. With flat travelling grates and chain grates it may amount to 12 or 20 per cent if fine coal or coal dust is used. A portion of the lost fuel may be recovered from the ashes, but 4 or 5 per cent is irrecoverable. The fuel and ash dropped in the first half-hour or so can be simply replaced in the furnace. (*b*) Loss in the residue finally left on the grate. This is determined by the composition of the fuel, the care exercised during burning, and the type and loading of the grate. With a normal loading the loss is from 2 to 3 per cent, while with a heavier loading it may be from 6 to 8 per cent. With a fuel not suitable for the grate in which it is used it may be as much as 20 per cent. This loss is influenced very greatly by the temperature of the charge. If the slag is difficult to melt, the value is low, but if it is easily melted it encloses particles of coal which then cannot burn and the loss rises. If the temperature of the charge is too low, more coal is left unburnt to form the residue. (*c*) Loss by the mechanical carrying of coke or coal dust into the flues by a too-high air velocity. This is most prevalent with low-percentage fuels, and stoking by means of shovels causes an increase in its amount. In unfavourable cases it may be as much as 12 per cent. (*d*) Loss through incomplete combustion of the distillation pro-

ducts through an insufficient oxygen concentration. Its value can also reach 12 per cent. (*e*) Loss by unburnt soot. The deposition of soot is due to the decomposition of the hydrocarbons and insufficiency of the oxygen supply for the combustion of the carbon thus set free. The loss may be 2 per cent. (*f*) Loss by incomplete combustion, carbon monoxide being formed instead of carbon dioxide. This effect may be considerable, and its presence can be tested by analysis of the exhaust gases. (*g*) Loss by radiation and conduction, which may be between 3 and 10 per cent. The amount is very difficult to determine accurately, and is dependent upon the heat density, the character of the surfaces, and the air velocity. (*h*) Heat loss from the exhaust gases after the abstraction of such heat as can be usefully employed. By suitable surfaces and temperature gradient the gases can be reduced to 200° F., and the lost heat limited to from 6 to 9 per cent.

Under the most favourable conditions it is possible for the total losses to be as low as 19 per cent, the efficiency being then 81 per cent.

The air required for combustion must pass through the openings in the grate, and if this is not sufficient, a secondary air supply must be provided from above or at the side. In most types of boiler furnaces the air required for combustion is not pre-heated. With industrial furnaces, however, particularly when they work at high temperatures, pre-heating takes place. Pre-heating must always be provided when secondary air has to be supplied to the gasification and combustion products. Excessive cooling would otherwise take place and would prevent the ignition of the gasification products. Let the air velocity in the grate openings be 3 ft. per second with a natural draught. The free grate surface, through which the air enters, is x times the total grate surface. Therefore the air flowing per hour through the free grate surface x per square foot of total grate surface, is $10{,}800x$ c. ft. If 1 lb. of fuel requires V c. ft. of air for its combustion, the amount being that actually (not theoretically) required, then on 1 sq. ft. of total grate surface $10{,}800x/V$ lb. of fuel are burnt in 1 hr. The following table gives the fuel consumption in pounds per square foot of total grate surface, and also the draught in inches of water measured over the layer of fuel with normal loading.

	Lumps.		Small Coal.		Dust.	
	Cons.	Draught.	Cons.	Draught.	Cons.	Draught.
Anthracite and non-bituminous coal ..	14	0·45	12	0·65	12	0·85
Semi-bituminous forge coal	16	0·4	18	0·45	18	0·65
Bituminous coal ..	20	0·3	22	0·35	19	0·4
Gas coal	20	0·2	20	0·25	24	0·35
Crude brown coal ..	40	0·2	46	0·3	52	0·3
Earthy brown coal ..	40	0·2	40	0·3	40	0·3
Mineral coal coke ..	15	0·35	16	0·45	20	0·65
Brown coal briquettes	30	0·25	35	0·35	38	0·55

The loading of the boiler furnace grate which has been designated normal

is that which produces the highest efficiency. A deficient or a too-heavy loading reduces the efficiency. The actual loading varies with the boiler requirements, but the change of efficiency due to erroneous loading is approximately the same for various types of boiler grates. The following figures give the experimental results of a test on this effect:

Loading	..	..	$\frac{1}{4}$	$\frac{1}{2}$	$\frac{3}{4}$	normal	$\frac{5}{4}$
Efficiency (per cent)	..	..	35	60	72	81	75

In order to make the grate durable, iron alloys capable of resisting high temperatures are used, while the grate bars are subject to internal and external cooling. The grates themselves are constructed in all possible forms, such as flat, inclined, step, and travelling grates. They can, according to their design, be applied to any fuels from high-percentage to low-percentage fuels. The loading takes place either by hand or mechanically. In the former case the furnace door has always to be opened, so that cold combustion air enters, while in the latter case this is avoided. Two types of loading can be differentiated, the fuel being either distributed over the entire grate surface or fed to the front of the grate only. A uniform distribution of the fuel is most suitable for high-percentage coal, while with low-percentage coal it is always necessary to load the front of the grate. By this means the fuel is gradually dried and distilled, while the distillation products are conducted over the burning charge. The products themselves are thus brought to the ignition temperature. Care should, however, be taken that there is a sufficient amount of hot combustion air for these gases, either by means of a free grate surface or by a secondary supply.

Another important factor is the effective removal of ash and slag either mechanically or by hand. When effected by hand the slag is removed from the front of the grate through the furnace door, or it can be done mechanically or by hand at the end of the grate. Care should always be taken that this removal is done efficiently. Ash and slag that falls through the grate may be allowed to fall into bins which are removed periodically, or the removal may be effected continuously by allowing it to fall on a moving conveyor. It may also be removed by mechanical or compressed-air scavenging. Fine ash in boilers is also removed in some plants by suction or compressed air in place of manual methods.

Up to now very little use has been found for the heat contained in the slag. With the common fuels in use the slag amounts to only a small percentage of the fuel used. Conditions are different in the oil-shale furnaces which are gradually coming into use in different parts of Germany. Only about 10 to 15 per cent of the fuel is used when the oil and coke are in shale form, and the residue can be used for the manufacture of bricks or Roman cement without being cooled.

An essential requisite for efficient combustion is correct strength of the draught. As stated, a steady current of air must flow through or over the fuel during the combustion or the gasification and combustion. This flow is produced by means either of a flue or of a ventilator inserted in front

of or behind the furnace after the waste gases have given up their heat. A ventilator and flue can also be combined, in which case the flue must naturally be of much smaller dimensions than with a natural draught. Care should be taken that the flow is not hindered by contractions or expansions in the pipes, corners, unsteady temperature conditions, or other disturbances.

If a furnace has been constructed for high-percentage fuels, and if, as a result of the use of low-percentage fuels, it is necessary to have a considerably larger amount of fuel on the grate, the use of an under-draught or a steam blower, or both combined, must be introduced. An enclosed chamber is formed under the grate. The air and steam are introduced into this part and are compressed to a pressure above atmospheric, depending upon the grate openings and the depth of fuel charge. In this case a flue would be necessary. With step grates and self-feeding furnaces the use of steam blowers and under-draught can often be avoided.

The draught is measured as pressure above atmospheric in inches of water head. If h is the flue draught in inches of water, v the gas velocity in feet per second, w_1 the weight of 1 c. ft. of waste gas in pounds, K a coefficient, then with $g = 32{\cdot}2$ ft. per second per second,

$$\frac{62{\cdot}3}{12}h = \mathrm{K}\frac{v^2}{2g}w_1.$$

The value w_1 is dependent on the temperature. If w_0 and w_1 are the weights per cubic foot, V_1 and V_0 the volumes per pound, and t_1 and t_0 the corresponding temperatures in degrees Fahrenheit, then

$$w_1 = w_0\frac{460 + t_0}{460 + t_1},$$
$$\mathrm{V}_1 = \mathrm{V}_0\frac{460 + t_1}{460 + t_0}.$$

If CO_2, O_2, and N_2 represent the volume percentages of carbon dioxide, oxygen, and nitrogen with a complete combustion, the weight in pounds of 1 c. ft. of waste gas is given by

$$w_0 = \frac{0{\cdot}123\mathrm{CO}_2 + 0{\cdot}090\mathrm{O}_2 + 0{\cdot}078\mathrm{N}_2}{100}.$$

Furthermore, $CO_2 + O_2 = 21$, and $N_2 = 79$.

The total draught from the grate to the opening of the flue may be divided into the following sections:

h_1 = draught underneath the grate bars.
h_2 = draught from underneath the grate bars to the grate openings at the upper edge.
h_3 = draught from upper edge of grate to the upper surface of the fuel.
h_4 = draught from the fuel to the foot of the flue.
h_5 = draught from the foot to the top of the flue.

Experiment shows that the coefficient K introduced into the formula is approximately 2 for h_1, h_2, and h_3, but a different value holds for h_4 and h_5. The required draught h_2 is calculated for an average air velocity v of 3 ft. per second and air temperature 120° F., when using natural draught. With forced draught the velocity may be between 6 and 10 ft. per second. The head h_3 is determined for a velocity of the burnt gases of about 12 ft. per second, and a temperature between 1800° F. and 2000° F. The pressure difference h_4 is the resistance encountered by the gases from the upper edge of the fuel to the foot of the flue, and is the sum of the differential effects obtained from the cooling of the gases through heat absorption and the friction from the walls. With an average gas velocity of 10 to 12 ft. per second, and temperature of the waste gases between 300° F. and 750° F., according to the plant, the coefficient K in the equation for h_4 is greater than 2 and may be as much as 10. The value of the flue draught h_5 may be calculated as follows: If t_1 is the mean temperature of the gases inside the flue, and t_2 that of the air outside, the corresponding weights of waste gases and air per cubic foot being w_1 and w_2, then if H is the height of the flue in feet,

$$\frac{62{\cdot}3}{12}h_5 = \mathrm{H}(w_1 - w_2).$$

Using the value of h_5 thus calculated and a velocity of the flue gases between 10 and 14 ft. per second, the value of K in the equation for h_5 lies between 1·7 and 2. If the waste gases of several furnaces are led into the same flue, the value of h_5 due to any one flow will be small owing to the larger diameter of the flue, but the total h_5 for all the flows should be of the same order as for the smaller flue of a single furnace. The total draught required, h_1, is the sum of all the resistances:

$$h_1 = h_2 + h_3 + h_4 + h_5.$$

This total draught, i.e. the pressure under the grate, is thus absorbed in the passage of the air through the grate openings, the fuel, and the flues. The velocities equivalent to various draughts for a boiler furnace are given in the following table.

h_1 =	0·02 in. of water.	v =	7 ft. per second.
=	0·1 ,,	=	15 ,,
=	0·5 ,,	=	35 ,,
=	1·0 ,,	=	50 ,,
=	1·5 ,,	=	61 ,,
=	2·0 ,,	=	70 ,,

Other furnaces work with much higher pressure differences. The ordinary flue draught is rarely sufficient for moving the gas against the resistance of a regenerator or recuperator. Pressures up to 12 in. of water head and more are used and are produced by a suitable blower.

3. TYPES OF GRATE FURNACES

Grate furnaces may be classified according to the position of the furnace with respect to the apparatus to be heated, falling into three classes: (1) internal furnaces, (2) furnaces under the boiler, oven, or retort to be heated, and (3) furnaces in front of the boiler, oven, or retort. The grates may also be classified according to the method of stoking (1) by shovel, (2) frontal stoking, and (3) under-feed stoking.

Internal furnaces, as the name implies, are within the apparatus to be heated, and their dimensions are limited. The combustion chamber and the flame zone must be small, so that non-bituminous high-percentage coal such as anthracite, mineral coal poor in gas, and coke are the best fuels. Cornish boilers and locomotive boilers belong to this class, also industrial furnaces for high temperatures and large quantities of heat, such as ceramic furnaces and crucible furnaces for metals. In the boilers heat conduction, and in the other furnaces heat radiation, produces a high efficiency.

The dimensions of underneath furnaces can vary within wide limits. They are used with water-tube and smoke-tube boilers, with brewing coppers, salt-extracting boilers, and annealing pans, and in other cases where vaporization, drying, or roasting is required, and in the ceramic industry. Between the grate and the surface where the heat is withdrawn, there must be a large combustion space so that the gases can be burnt before they meet the surface, which may often be very cool. In this connexion the surrounding brickwork often exerts a considerable influence through radiation. Fuels of medium quality with gaseous constituents are most suitable for these furnaces. As low-percentage fuels are used to a large extent with underneath furnaces, care should be taken in the construction of the furnaces to provide sufficiently large grate surfaces. Furnaces for heating ovens and boilers for central heating are often partly internal and partly underneath.

In frontal furnaces, the combustion chamber is quite separate from the surfaces to be heated. The grate is enclosed by fireproof brickwork, which should be protected from heat conduction and radiation to the outside. The walls of the brickwork, radiating inwards, assist in heating such gases as are still unburnt to the ignition temperature. The air required for combustion of these gases is conducted either into the furnace or into the space in which the heat withdrawal takes place. This air should be pre-heated. It is possible with these furnaces to have the combustion space and the flames very long. The most suitable fuels are low-percentage fuels and those rich in gas. With low-percentage fuels poor in gas this type of furnace is used for boilers. With low-percentage fuels rich in gas it is sometimes used for metallurgical, ceramic, and chemical purposes. The furnace itself can have an oxidizing or reducing effect according to the amount of oxygen supplied. To this group belong the Siemens-Martin furnaces, coke furnaces, furnaces for manufacturing illuminating gas, furnaces for the porcelain, clay, brown-ware, and glass industries, muffle furnaces for the manufacture of sulphate, zinc, &c.

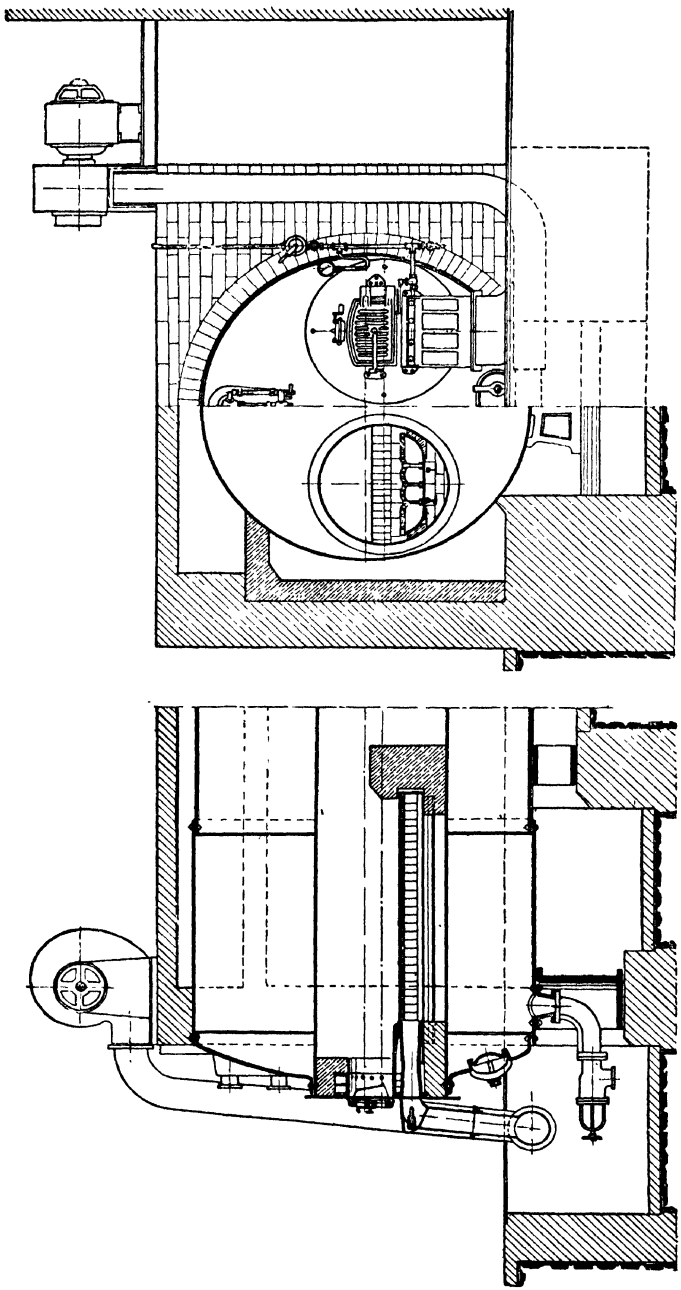


Fig. 5.—Internal Furnace of a Lancashire Boiler with Nozzle Grate

4. INTERNAL FURNACES

In figs. 5, 6 and 7 the arrangements of the furnace within a Lancashire boiler are shown with different kinds of grates. Fig. 5 shows a nozzle grate with an undergrate blast of air for crude brown coal and a steam supply for

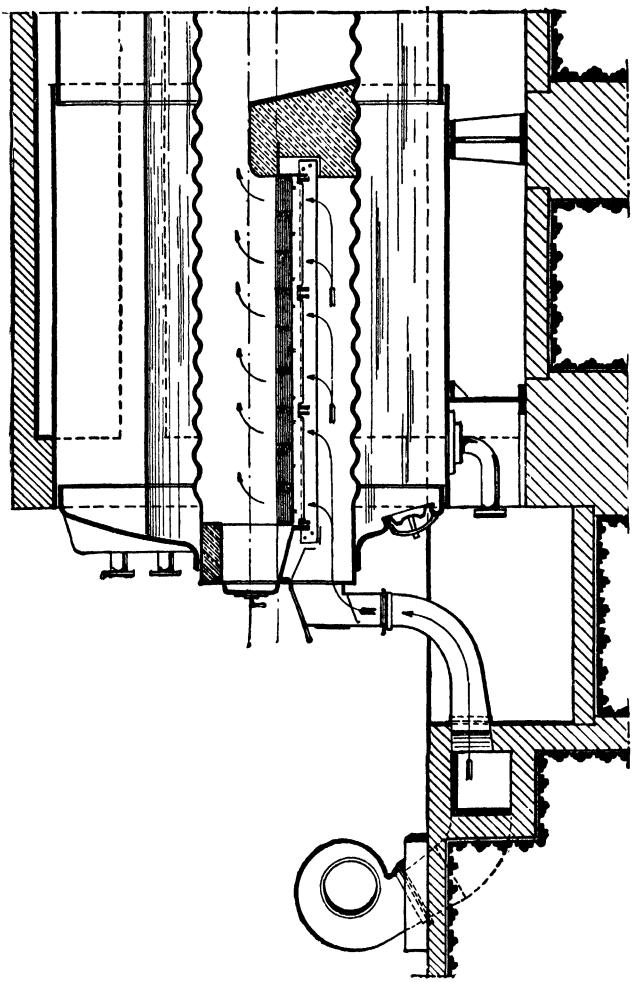


Fig. 6.—Internal Furnace of a Lancashire Boiler with Step Grate

coke. This grate is only suitable for a fuel with little ash and light ash at that. The light ash that passes over the fire bridge is extracted by suction (see fig. 7). Fig. 6 shows the construction of the step grate. With this arrangement, the unburnt fuel dropping through the grate can be reduced to a minimum, while at the same time the bars of the grate are well cooled, so

that all types of fuel can be burnt on it. The passage of the injected air should cause a minimum movement amongst the fuel particles.

In burning low-grade fuels with a low ash content like lignite, brown

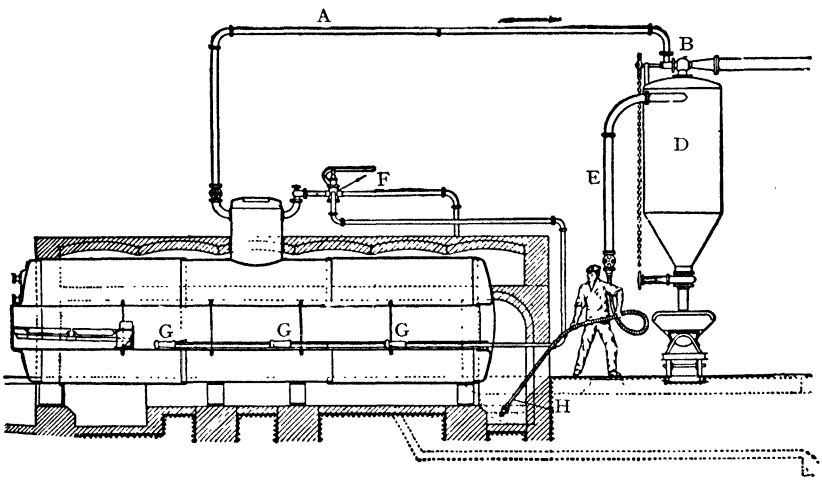


Fig. 7.—Arrangement for Extraction of Light Ash

A, Live steam pipe. B, Injector. C, Steam outlet pipe. D, Extractor for settling light ash. E, Suction pipe for ash extraction. F, Distributor valve for ash disturber. G, Light ash disturber. H, Suction nozzle.

coal, &c., a large volume of light ash is apt to be formed in spite of the fact that the actual *weight* of the ash is small. This is especially the case if the fuel contains a large amount of dust. It is therefore necessary to have an

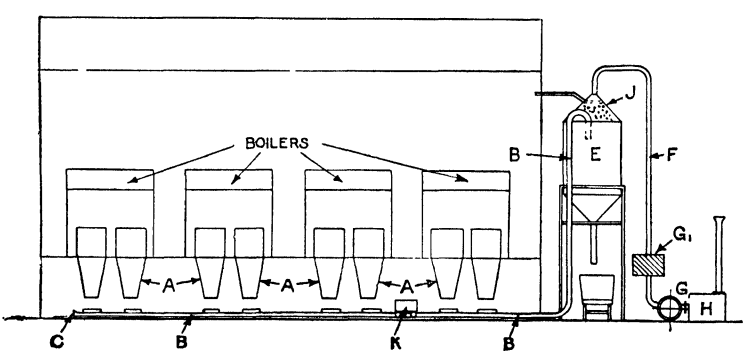


Fig. 8.—Diagrammatic Arrangement of Vacuum Ash System

apparatus for getting rid of this light ash in order to save labour and increase the boiler output; fig. 7 gives an example of an arrangement of this kind employed by many firms.

A diagrammatic arrangement of a vacuum ash system is shown in fig. 8,

in which the ash hoppers below the boilers are indicated at A. The ash pipes B are provided with openings under each boiler ash hopper to receive the ashes. These openings are provided with lids to close them when not in use. The end of the ash pipe is B left open to the atmosphere at C, and the main supply of air for conveying the ashes enters the system at this point. The ash pipe B is led to an ash receiving bunker E, where the ashes are delivered and quenched by water sprays J. The ash receiver is provided with an ash discharge valve at the bottom, through which the ashes are periodically emptied into railway wagons. The air for conveying the ashes to the ash-receiving bunker is drawn through the system by a motor-driven exhauster G, by way of pipe F, and through the dust extractor G_1. The exhaust from the exhauster G is discharged through a silencer and atmospheric pipe H.

With certain coals the ash is delivered from the ash hoppers under the boilers in relatively large slabs, and

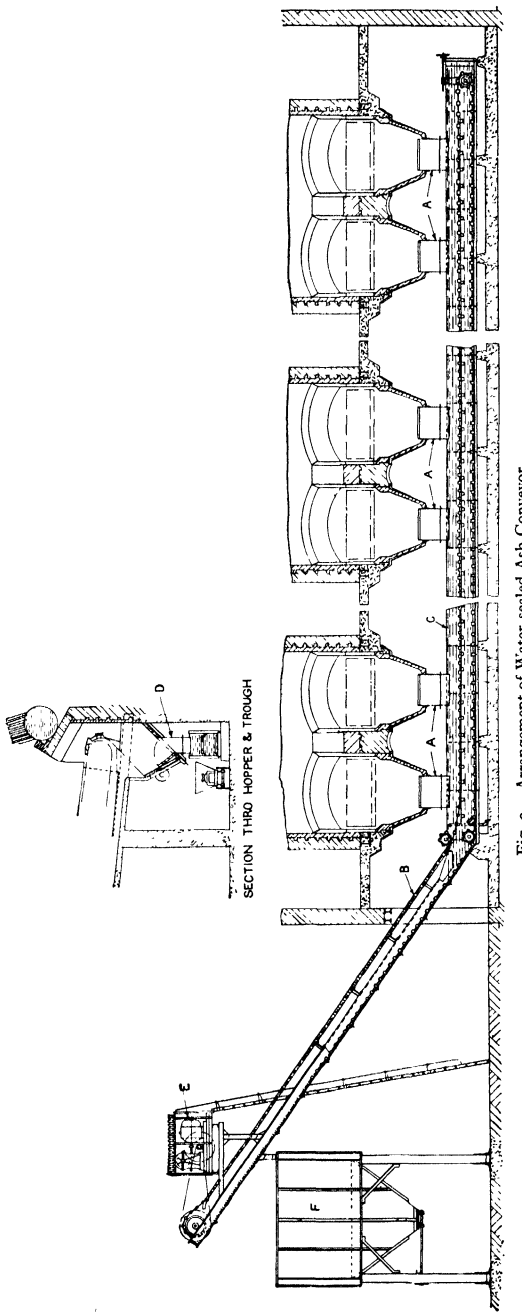


Fig. 9.—Arrangement of Water-sealed Ash Conveyor

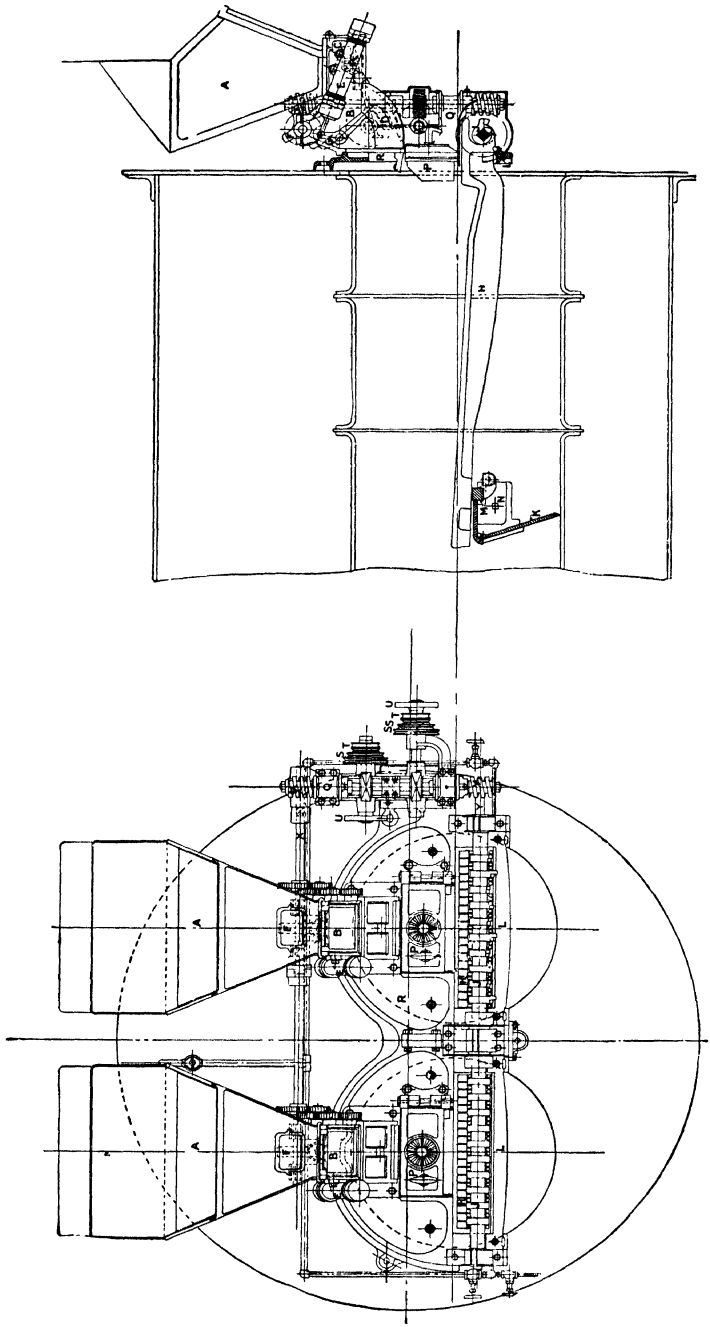


Fig. 10.—Sprinkler Type of Mechanical Stoker ("Triumph")

it is necessary that these should be broken before being admitted into the ash pipe. For this purpose a travelling motor-driven 4-roll ash breaker is provided on rails below the ash hopper doors. The crusher is indicated at K.

Fig. 9 shows another special type of ash-handling plant. The water-sealed shoots from the boiler grates are shown at A, while the conveyor and trough are shown at B and C. The ash shoots are shown in section at D, from which it will be seen that an emergency ash door is provided for the extraction of the ashes in the event of the conveyor being out of commission. The conveyor chain is driven by motor and gearing at E, and the ashes are discharged in a thoroughly quenched condition into an ash hopper F. The trough under the water-sealed shoots may be made of cast-iron or concrete, and in either case the bottom of the trough should be provided with loose cast-iron wearing plates for the chain to run on. A continuous flow of water is maintained in the trough in order to keep the temperature of the water low enough to prevent steaming.

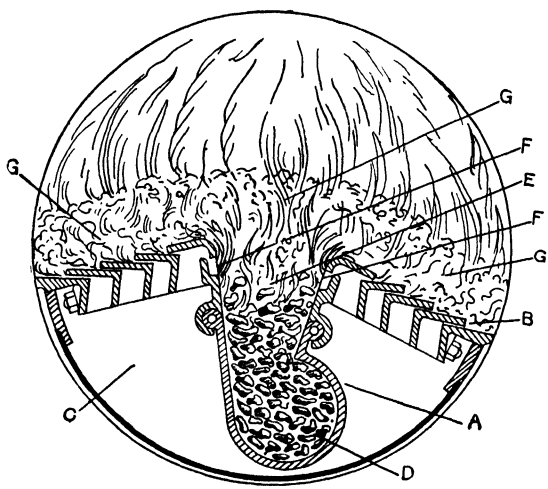


Fig. 11.—Principle of Underfeed Stoker

Arrangements for the mechanical stoking of the furnaces of Lancashire boilers are shown in figs. 10, 11, and 12. The stoker shown in fig. 10 is of the sprinkler type. Power is supplied from a suitable countershaft to two pulleys S and SS, which transmit a slow rotary motion independently to two shafts X and Y by means of worm gearing. These two shafts actuate the feed and the firebar motion. The coal descends from a hopper A into a feed box in which an adjustable coal pusher C is situated. This coal pusher receives a reciprocating motion from the shaft X through a train of spur wheels. At regular intervals the pusher feeds a small charge of coal to the front of the shovel D, which is held by forged steel arms keyed on to a shaft working in bearings on either side of the feed box. On the centre of this shaft is keyed a steel trigger G. The shaft X carries a four-armed cam F, the projections of which severally engage with the trigger G on the shovel axle, thus raising the shovel arm into the backward position. A prolongation on the shovel arm is connected to a powerful spring E, which is compressed when the shovel is in the backward position. As the cam F rotates and releases the trigger on the shovel arm, the spring drives the shovel smartly forward,

scattering the charge of coal over the fire. The four arms of the cam, being of different lengths, vary the throw of the shovel, thus distributing the fuel uniformly over the grate. The firebars are actuated by cams J, threaded loose on the square shaft V. A separate cam for each firebar engages on a renewable nosepiece on the bar and draws each bar forward 2 in., one after the other. This separates it from clinkers and frees the air space. When all the bars have been brought forward, there is a slight pause, and two side projections on the cams then push all the bars simultaneously back into

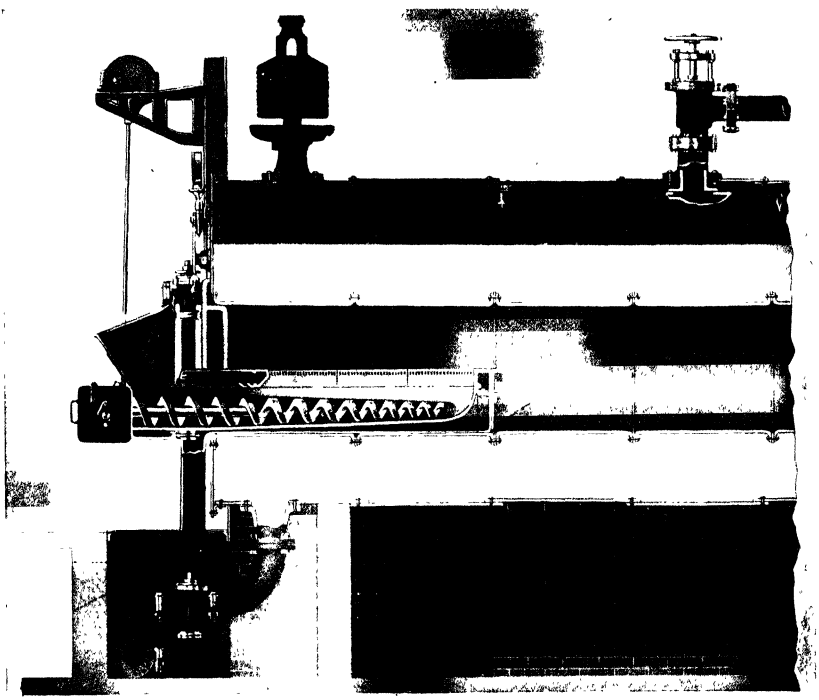


Fig. 12.—Application of Underfeed Stoker to the Furnace of a Lancashire Boiler

their original position, carrying back with them the whole fire. In this way the clinker and ash travel towards the back of the furnace and finally fall into the ashpit.

The principle of the underfeed stoker as applied to the furnace tubes of a Lancashire boiler is shown in figs. 11 and 12. A retort or fuel magazine, A, is fitted along the bottom of the furnace tube. In the lower portion of the magazine a taper feeding worm conveys the coal along, and gradually pushes it upwards on to the terraced grates B. As the coal gradually rises it commences to burn at the point E, where incoming air is supplied through the tuyeres F from the wind box C. The coal is thus first coked, the volatile hydrocarbons mixed with air from the air inlets of the fuel magazine rising

through the incandescent mass above, where they are more or less completely burnt, combustion of the solid fuel being completed by the introduction of air through the apertures in the sides of the terraced grates B. By this means the makers claim that as combustion is practically complete

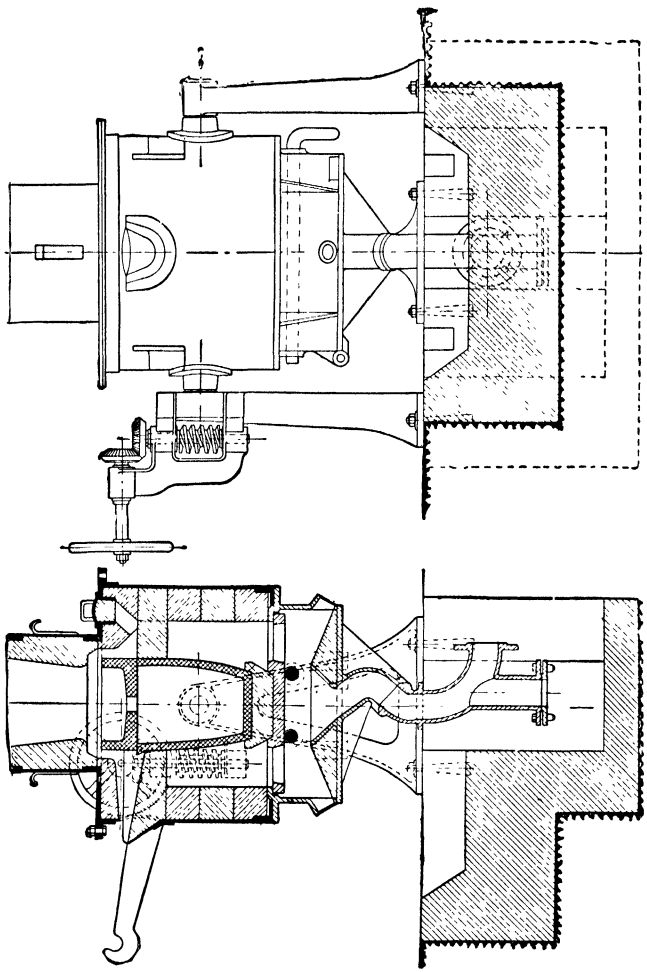


Fig. 13.—Filtering Crucible Furnace for Coke Firing (*Badischen Maschinenfabrik*)

by the time the gases have passed through the superincumbent layer of incandescent coal, a clear, short flaming fire results, and the cooling effect of the plates of the boiler does not prejudice the perfect burning of the products of combustion. This arrangement, which ensures surface combustion, is generally acknowledged to produce a good fire, but there is a liability to jamming with the worm feed. The stoker itself consists of a hopper situated at the front of the boiler, through which the fuel is fed into a com-

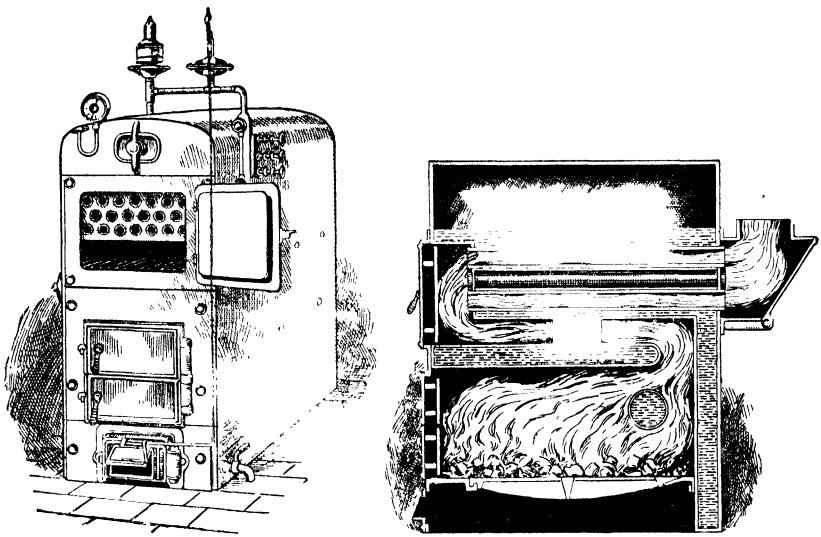


Fig. 14.—Dromana Tubular Boiler

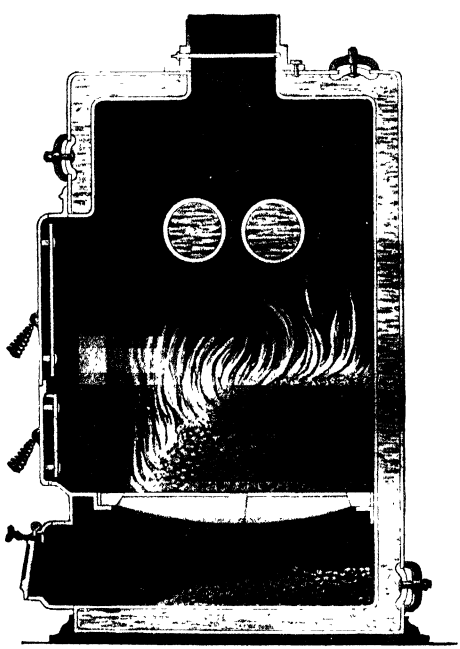


Fig. 15.—Royal Goliath Boiler

bustion retort which runs along the bottom furnace tube. The worm conveyor feeds the coal along this retort, at the same time pushing it upwards as already explained. This worm is driven from an eccentric on the main shaft through a ratchet wheel and pawl. The feed may be varied by the adjustment of a shield at the side of the ratchet wheel, which determines the number of teeth fed by the pawl for each stroke of the eccentric. The air chamber is supplied with air through a cast-iron box situated at the back of the hopper, the air supply being regulated by means of a blast gate. In order to prevent injury to the mechanism in the event of a jam in the driving worm, a hook-shaped piece of cast iron is inserted between the ratchet wheel on the

driving shaft and the shaft which it drives. This breaking link can be inexpensively and easily renewed without interruption to the boiler.

An example of a crucible furnace is given in fig. 13. Although the crucible to be heated is over the furnace, the heating of the surrounding fireclay walls, which subsequently have a radiating effect on furnace and crucible, is so essential a feature that the furnace may be classed as internal.

For the heating of rooms either by hot water or by steam, boilers with

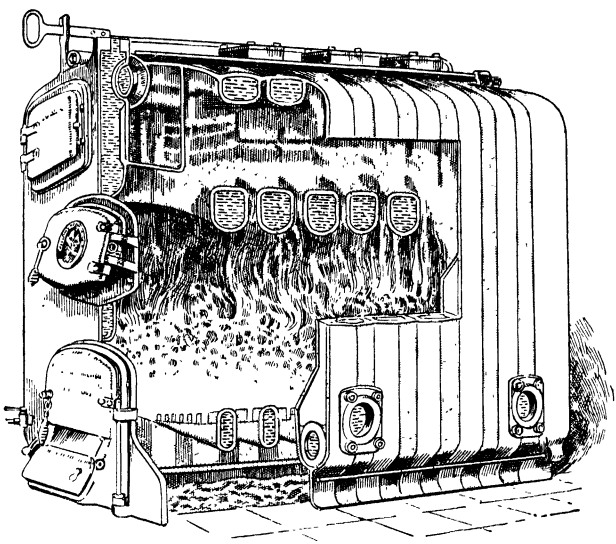


Fig. 16.—Britannia Boiler

internal furnaces are usual in modern plants. Many types of small boilers for heating water for domestic purposes consist of a fire grate nearly surrounded by a water tank. In figs. 14, 15, and 16 are shown various types of boilers for central heating of rooms. Burning takes place usually from the bottom upwards. A boiler designed to have a large heating surface is shown in fig. 16.

5. UNDERNEATH FURNACES

In fig. 17 is shown a water-tube boiler with furnace wholly underneath. The combustion is promoted by an under-grate air blast. Such furnaces are particularly suitable for low-percentage fuels. Care should be taken that the air velocity after passing through the fuel is not too great, as otherwise dust will be blown through the flues in the form of fine ash. Owing to the unequal burning of the material the layer of fuel becomes thinner at some points than at others. Too much air or steam then flows through these points. This evil is overcome by means of box-like divisions underneath the grate. Each box contains a specially adjustable air or steam pipe which can be partly closed by a disc or valve. By this means the stoker is able to decrease

the excess supply of air or steam at the points where the burning is unequal.

The formation of fine ash can be partly or entirely avoided, in particular with coke, dross, and crude brown coal, by the insertion of a so-called fire stop. For example, a fireclay ring may be placed over one end of the furnace and the fine ash is ignited on the red-hot fireclay. A method which reverses that of the adjustable draughts for unequal burning is that of the fire bridge,

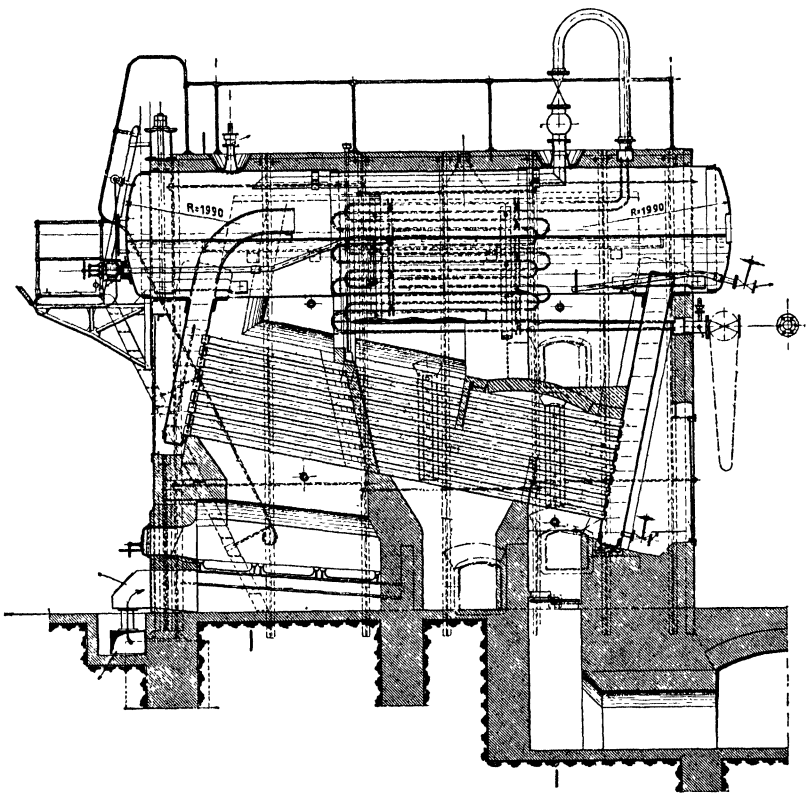


Fig. 17.—Borsig Water-tube Boiler with Under-grate Blas

which is lowered into the furnace, and those parts of the fuel which are still unburnt are consumed by the forcing downwards of the flame and the mixing of unburnt gases and air. A continuation of the bridge acts in the same way as the ring. By means of this arrangement the efficiency of the boiler can be increased from between 45 and 55 per cent to between 65 and 75 per cent with coke or crude brown coal and flat grates. The fire bridge with continuation in a horizontal direction known as the fire bridge arch is said to give an increase of boiler efficiency of 10 per cent, due to the eddying currents produced which increase the heat transfer to the boiler surface.

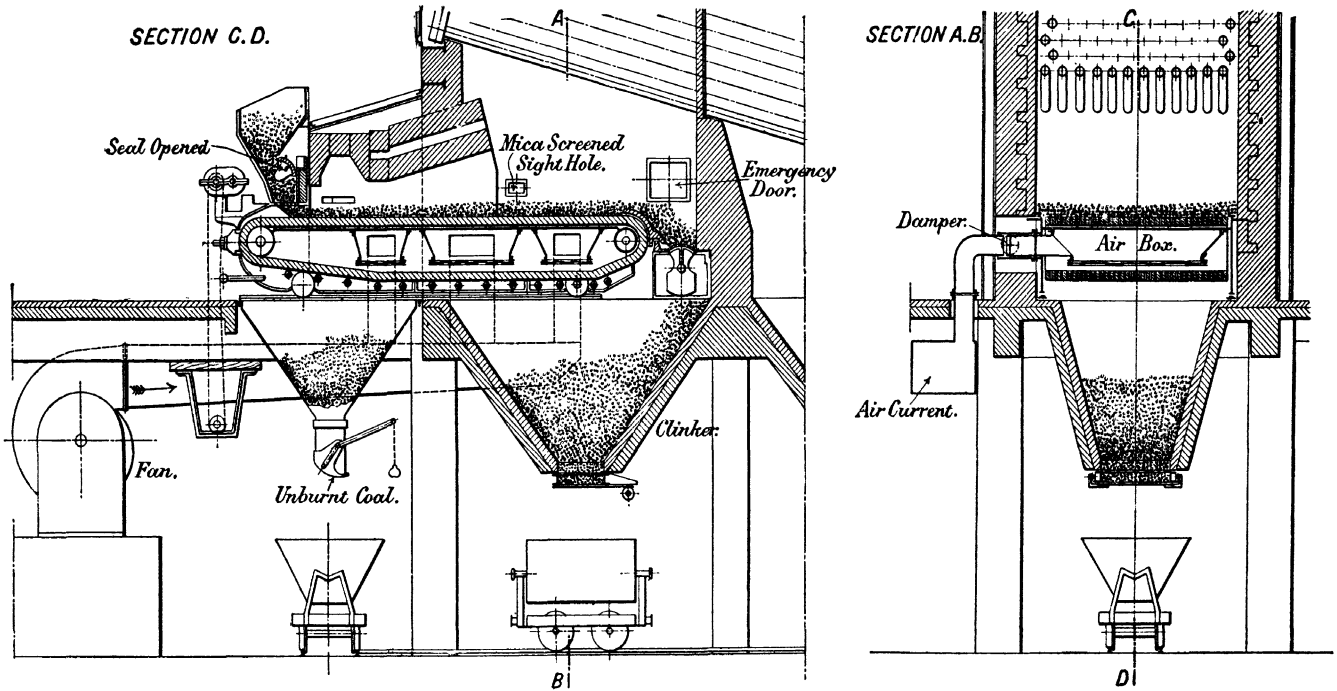


Fig. 18.—Application of a Forced Draught Mechanical Stoker to Babcock & Wilcox Water-tube Boiler

From *Engineering*

Other methods of burning the fine dust are the insertion at intervals of segments of firebrick which act in the same way as the ring, and the insertion of spiral vanes for the thorough mixing of the coal dust with air.

In recent times pendulum grates have been introduced, following on the utilization of crude brown coal containing a large proportion of sand. These grates are arranged so that the bars of the step grates revolve about an axis. By this means the hard baking of the slag is avoided, while the position of the material is continually changed. At the end of the step grate there is in most cases a slag grate. The second burning of the slag reduces it to a mass that can be removed separately. The gear for moving the grate bars is worked by an electric motor or other drive. Reference may also be made to the method of covering the grate bars with aluminium, as this prevents the slag from baking hard, and the grate bars from burning.

Coal washings may be burnt in a grate with an under-grate blast of air, and coke dross when both air blast and steam are supplied. In other grates it is necessary to first press the washings or dross into bricks. The coal washings contain 20 to 30 per cent of water, the coke dross usually only a small percentage, while the non-combustible parts of both kinds of fuel amount to between 10 and 22 per cent. The grate itself must have very fine holes so that the fuel does not fall through. Air pressures up to 2 in. of water are required, and the blast also serves to prevent particles of fuel from falling through. The high air pressure necessitates a high fuel charge. Damp coal washings and coke which is difficult to ignite should be heated to the ignition temperature by means of a radiating arch. In addition to fixed grates, which are only used for small amounts of fuel, travelling grates are used which fulfil all the above conditions, and are particularly suited for this character of fuel. Grate loadings up to 35 lb. per squre foot are possible.

Fig. 18 is a diagram of a travelling grate used for a furnace underneath a water-tube boiler. The grate is attached to a continuous chain which passes round two shafts. The front shaft is driven by a cog wheel, and its speed is capable of adjustment. At the end of the grate is a slag box into which the slag from the grate is carried. As the slag box contains a considerable proportion of unburnt fuel, particularly with a high loading of the grate, plants have been constructed in which the slag falls into a heated shaft which gasifies such volatile parts as are still unburnt, and these are then added to the furnace gases and consumed. Automatic stoking is used with large grates. In one method, the fuel is thoroughly mixed and disintegrated on the grate by means of a serrated bent tube which is water-cooled and which moves backwards and forwards. In other designs the bars of the travelling grate are raised at certain points by means of rollers lying underneath the grate. Tests on a Garbe boiler plant revealed an efficiency of 83 per cent with these arrangements.

Periodic furnaces in the ceramic industry are largely of this class. The fuel is burnt on a grate, the gases rising straight upwards and coming into direct contact with the material to be burnt.

6. FRONTAL FURNACES

In fig. 19 is shown a frontal furnace for heating a water-tube boiler, with manual stoking. The fuel is brought up by wheel-barrows or con-

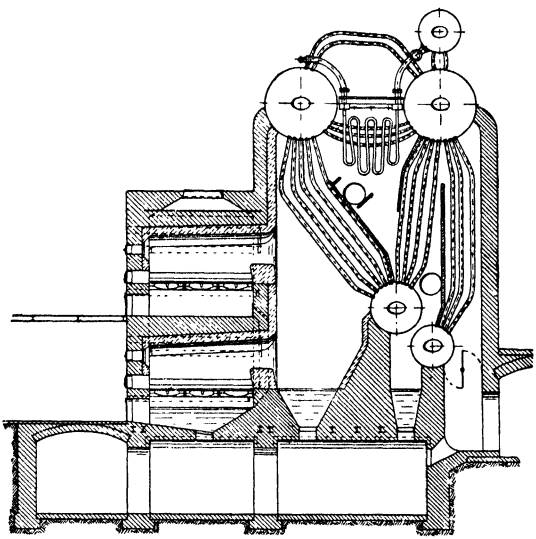


Fig. 19.—Two-storied Frontal Furnace for a Water-tube Boiler

veyor, and supplied to the furnace through horizontal doors which open downwards. The grate is a box grate, which type is suitable for low-per-

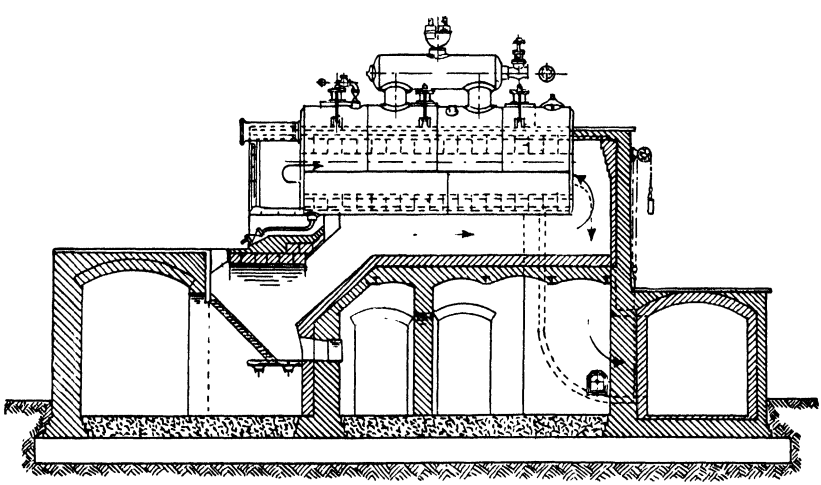


Fig. 20.—Special Multi-tubular Boiler with Frontal Furnace for burning Bagasse or Refuse Fuel

centage fuels rich in gas. The stoking must be manipulated so that there is not excessive combustion at the sides of the sections. A frontal furnace for heating a multi-tubular boiler is shown in fig. 20. The purpose of these

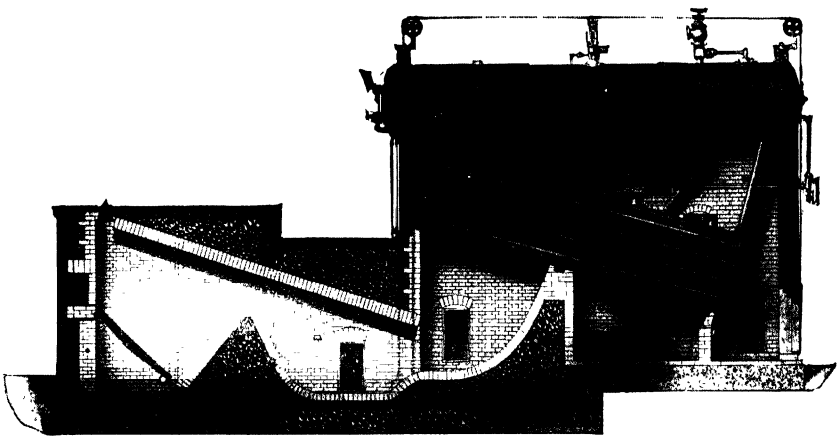


Fig. 21.—Babcock & Wilcox Boiler fitted with a Frontal Furnace for burning Bagasse with Natural Draught

external furnaces is to provide very large combustion chambers that are needed when low-grade fuels are burnt; e.g. brown coal, tan-bark waste, and wood chippings, &c. They are commonly used for burning green bagasse, the waste product from the manufacture of sugar from cane. Heat from the fireclay brickwork assists in the distillation of the volatile constituents of the fuel. These mix with the secondary supply of air which has been heated to a high

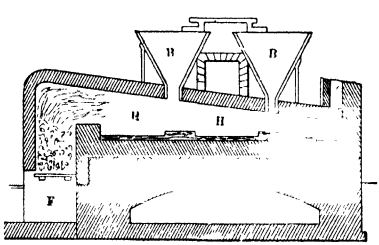


Fig. 22.—Reverberatory Furnace for Calcining Copper Ores

HH, Hearth. F, Fire-place. BB, Hoppers

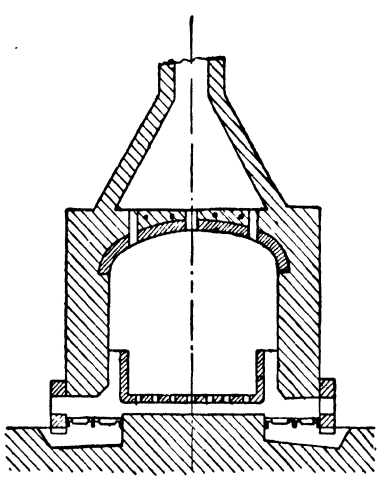


Fig. 23.—Reverberatory Furnace with Flames rising upwards

temperature, burn, and pass to the boiler with the other combustion products. Another example of this kind of furnace used in conjunction with a water-tube boiler is shown in fig. 21.

Reverberatory furnaces can be classified with frontal furnaces, the essential feature being that combustion takes place in a chamber separate from the body to be heated, so that the latter does not come in contact with the fuel. Fig. 22 shows the principle of the reverberatory furnace, while in figs. 23, 24, and 25 are shown the different constructions designed to produce flames of desired character. Fig. 23 represents a furnace with flames rising upward for the manufacture of pottery, fig. 24 a similar furnace with downward flames, and fig. 25 one with horizontal flames. In the first design, the hot gases move upwards from below; in the second they have a circular movement in the furnace, and are withdrawn into the smoke stack from the floor of the furnace; while in the third they flow in a horizontal direction through the substance to be heated.

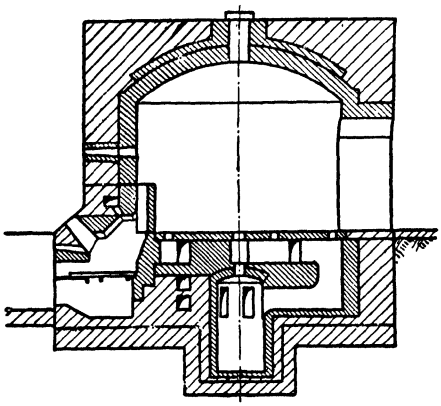


Fig. 24.—Reverberatory Furnace with Downward Flames

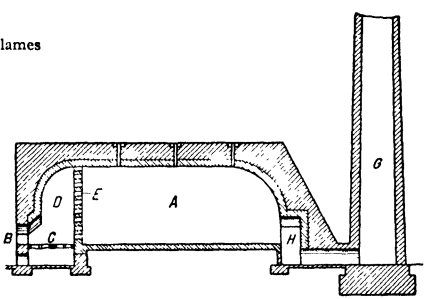


Fig. 25.—Reverberatory Furnace with Horizontal Flames

7. GRATES

Some examples of various kinds of grates have already been shown in the preceding sections. Shovel furnaces, either manual or mechanical, are used with flat or slightly inclined grates. The newly added fuel, being fed to the top of the burning charge, is dried and distilled before combustion takes place. On grates with frontal stoking, the fuel is pushed forward through the length of the grate, and is introduced gradually first into the distillation, then the combustion zone, and finally into the ash zone. With all large grates, the fuel is injected and periodically moved during the combustion by mechanical means. An example of a plant with a step grate fed from above at the front is shown in fig. 26. Each of the bar sections of the grate passes through a plate, and possesses underneath a projection in contact with guides on a frame that moves backwards and forwards. By adjusting the guides the bar sections can undergo a large or small, horizontal, backward and forward displacement. The idea behind the design of the

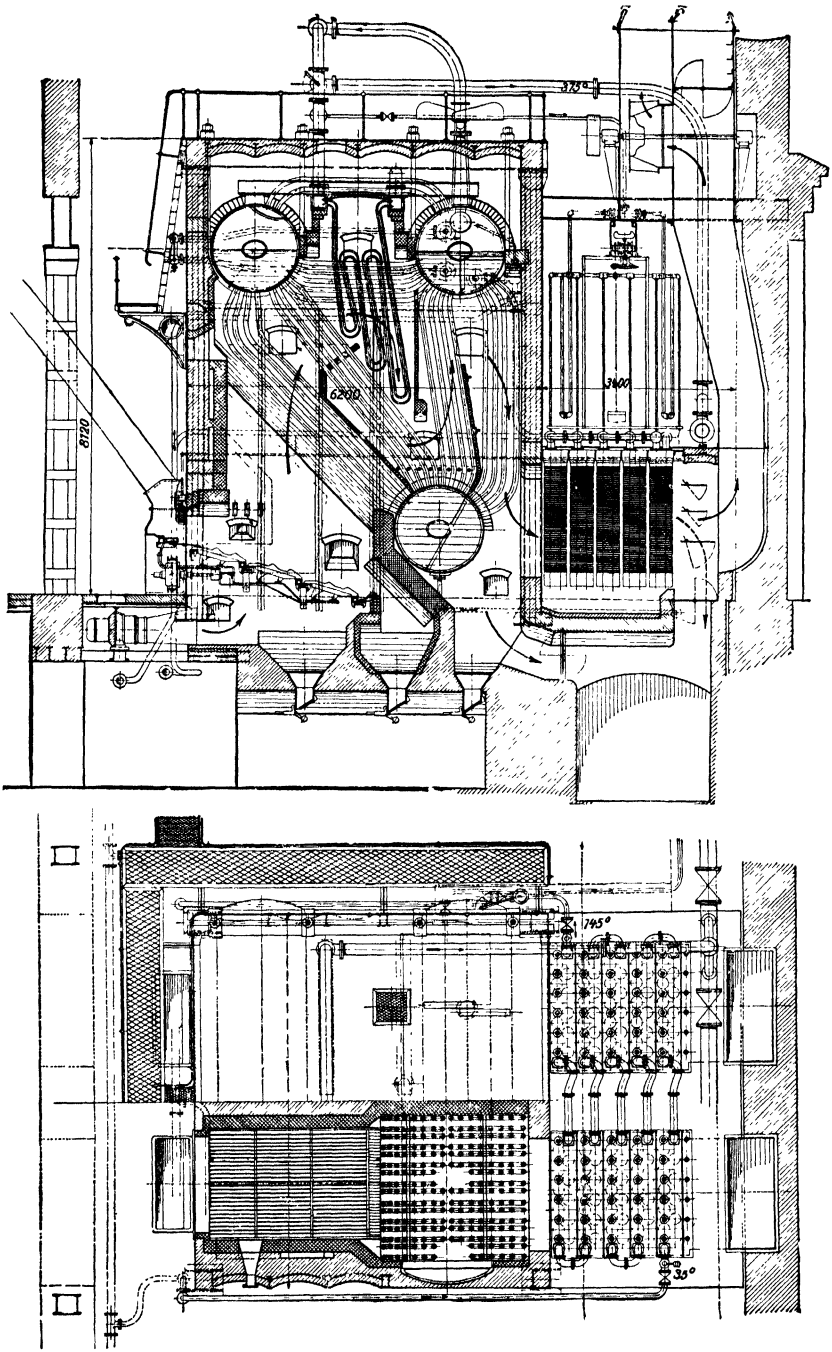
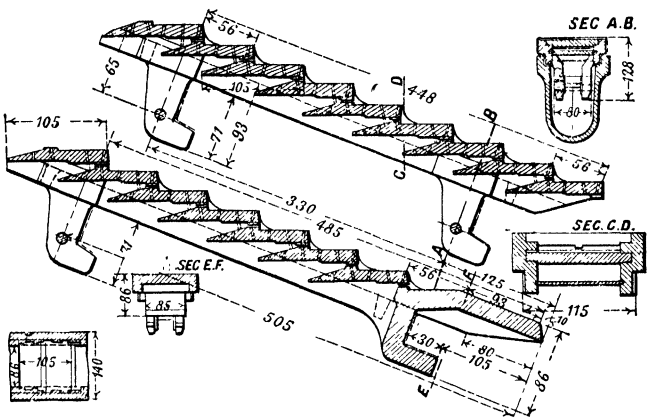


Fig. 26.—Boiler Plant with Over-feed Step Grate and Economizer (*Kablitz*)

furnace is that the fresh layers of coal will roll over the charge that is already burning. This is effected by means of the boss-like projections on the grate bars, the bosses jerking backwards and forwards with the bars. When the bars move forward the fuel is ventilated and pushed forward, while with the moving back of the bars the front bosses, which are turned towards the centre of the furnace, draw back the lower layers of coal so that the upper layers move over them. In practice it has been found that this furnace uses 65 lb. of good coal per square foot per hour, 55 lb. of anthracite, or 198 lb. of crude brown coal.

The details of a step grate can be seen from fig. 27, which shows a portion of the " Pluto " grate. The inclination of such grates is dependent on the



From *Engineering*.

Fig. 27.—The Fire-bars for a " Pluto " Mechanical Stoker

size of the lumps of fuel as well as on the roughness and hardness. With fine coal the inclination varies from 26° to 29°, and with larger coal from 30° to 33°. A change in the fuel supply should, therefore, be accompanied by an adjustment of the angle of the grate. The fuel is supplied through funnels at the front, and the adjustment of the height of the fuel charge itself is effected by means of a regulating valve. Bridges are also inserted in the delivery pipe in wide grates. These regulate the fuel distribution, as the fuel lying nearest the brickwork burns more quickly than that in the middle of the grate. Step grates often have one or two horizontal or slightly inclined flat grates which can be drawn out or folded down, attached to the end for removing the ashes. They often work without an under-draught of air, though it is more usual for them to have an under-draught either with or without a steam jet according to the type of fuel. In order to obtain drying, distillation, and a good ignition of fuels that are difficult to ignite, these grates are provided with ignition arches. These are necessary with fuels rich in gas, as was seen in the section on combustion. Any failure to ignite brings about a large excess of air, delaying the distillation and subse-

quent combustion of the fuel. The step grate is suitable for brown coal briquettes, crude brown coal, and peat. The fuel consumption may reach 60 to 80 lb. per square foot per hour. In some designs of step grate for a high-performance furnace using crude brown coal, before the coal falls on to the grate it passes through a preliminary drying and gasification grate, so that space is saved in the actual step grate. A vertical shaft enables the coal to fall uniformly on the first grate. The escaping moisture and distillation products pass through a chamber attached to one side of the vertical shaft over the step grate, on which a simple combustion of coal can then take place.

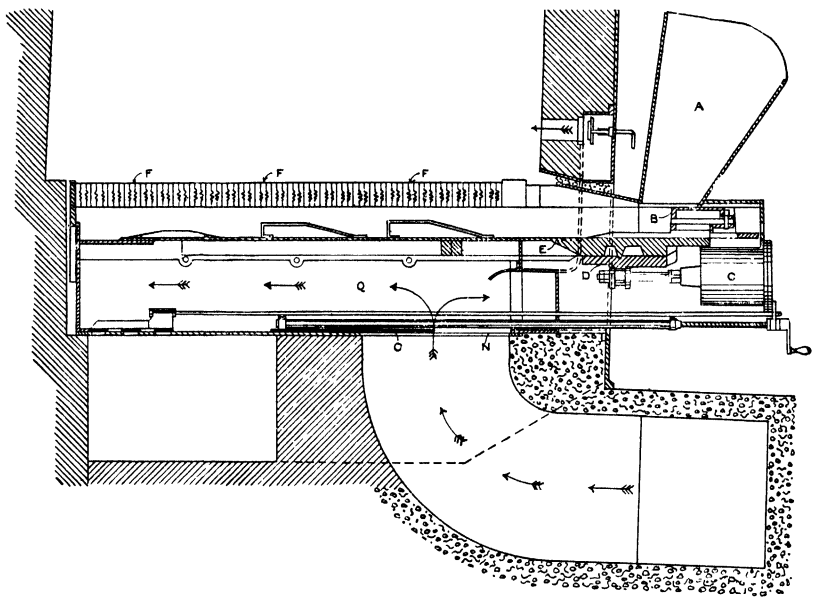


Fig. 28.—Longitudinal Section of an Under-feed Mechanical Stoker

A, Hopper. B, Coal feeding block. C, Steam cylinder. D, Crosshead. E, Reciprocating sliding bottom. F, Firebars. K, Dump plate for clinker. N, Air inlet. O, Adjustable air gate. Q, Wind box. S, Hot air discharge. T, Hot air chamber (see fig. 29).

The box grate requires a uniform, fine-grained fuel containing gas. It is particularly suitable for fine coal.

In the case of travelling grates, the material is always laid on the grate at a uniform height direct from the funnel, and the grate movement depends upon the rate of fuel consumption. The combustion of the gas and fuel is assisted by the ignition arch. Fuels poor in gas, such as coke, can be burned, although very low ignition arches are required. Crude brown coal gives an unfavourable combustion when placed straight on a travelling grate. A step grate is therefore inserted so that the crude coal is dried and distilled before it reaches the travelling grate, which then has only to burn fuel which is already red hot. With this arrangement an average of 60 lb. per square foot per hour of crude brown coal is burnt. If crude brown coal is mixed with

high-percentage coal (up to $\frac{2}{3}$ crude brown coal), without this preliminary heating, coal consumption up to about 25 lb. can be reached. If the crude brown coal is dried, though not on the grate itself but by means of the ignition arch, the consumption can be increased to about 35 lb. In the former case a special heating of the ignition arch has proved advantageous.

The arrangement of an underfeed grate stoker of recent design, and on a different plan to those of figs. 10 and 12, is shown in figs. 28 and 29. The mechanism is actuated by steam pressure on the piston of the cylinder under the feeding funnel. A reciprocating motion is given to the long plate carrying the fuel under the grate bars, and to the push rod under the mouth of the funnel. At each stroke a quantity of fuel is pushed forward on to the plate, and this is unable to return when the rod moves back. As the

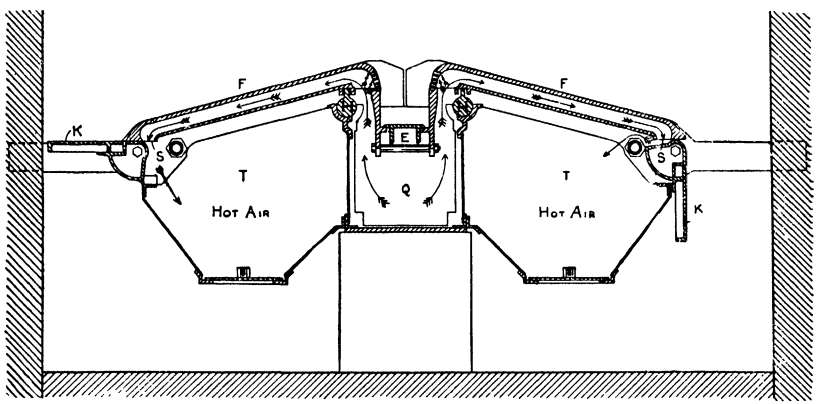


Fig. 29.—Cross Section of an Underfeed Mechanical Stoker

plate moves back, the sloping abutments on the top cause the fuel to rise on to the grate bars and under the burning charge, and gasification takes place during motion through it. Radiation from the burning charge downwards also assists the distillation from the new charge beneath. The gases and combustion air, which is also supplied from underneath, combine immediately over the solid fuel. The ash is removed at the side of the grate. Owing to the design of the grate, the largest quantity of air is caused to enter at that point at which the fuel is supplied to the grate after distillation. The underfeed grate is not suitable for fuels containing water; it can, however, use low-percentage fuels and those containing ash, as well as high-percentage fuels. The fuel must be uniform in size. The consumption with coal nuts is 20 to 26 lb. per square foot per hour, and with small coal and coal-dust up to 36 lb. per square foot.

The grate openings, i.e. the free grate surface, should be so arranged as to allow more air to enter at points where there is a high layer of fuel than where it is lower. The ratio of free grate surface to total grate surface should, therefore, decrease from the furnace door to the fire bridge. The air supply will thus be adapted to the amount of fuel at disposal.

8. WATER AND ASH

The water contained in the fuel has usually no influence on the combustion of the other constituents. The moisture is driven out by the heating of the coal before and during gasification. In most cases there is no water present when hydrogen, coal, and hydrocarbons are burnt. Steam rarely comes into contact with the burning layer of carbon, so that there is no formation of water gas, which would give a heat of combination of 4300 B.Th.U. per pound. The water in the exhaust gases carries with it a considerable amount of heat in the form of latent heat of vaporization.

Ash is to a great extent removed in the preparation of the fuels. The calorimetric value increases in proportion to the increase in percentage of combustible substance, while the efficiency increases in a higher proportion as the conditions for complete combustion are more favourable. In particular, the requisite amount of air for combustion is able to come into contact with the fuel more easily in the absence of inert mixture. The ash contains a mineralogical and an organic part. The former has a high melting-point and becomes part of the residue in an almost unchanged state. The second part arises from the pith and fibrous substance and is thoroughly mixed with the coal. Its melting-point is as low as 1800° F. to 2000° F., as it consists of potash, iron, sulphur, and silicic acid, while the mineral ash is mostly alumina. This organic part is spherical and blistered, while the mineral portion when seen in the residue is laminated, with sharp edges. In the preparation of the fuel the mineralogical part only can be decreased. If, therefore, a fuel contains a great deal of ash, it must be burnt with a high charge so that the oxygen in the inlet air can find sufficient points of combination. As the high fuel charge offers a high resistance, the draught must be increased either by raising the chimney or by means of a suction ventilator or an undercurrent of air. The waste ash of these fuels contains a large amount of sensible heat, but a much larger possible loss is due to the presence of unburnt substances, their character being such that the furnace conditions are not sufficiently intense to cause their combustion. A considerable amount of fuel thus remains in the ash, and this can be recovered by treating the ashes. There are two modes of treatment—the wet and the dry method.

The dry or magnetic treatment consists of moving the ash over a rotating roller in the centre of which a powerful magnet is placed. As nearly all coal contains iron, this iron being present in the ash after the combustion, it is held by magnetic attraction to the surface of the roller, while the combustible parts fly off in a tangential direction.

The wet treatment can be applied in two ways. One way consists of adding the ash to a liquid medium of which the specific gravity is greater than that of coke. This is made of water with an admixture of loam, plaster of Paris, &c. The ash residue, which is divided into fine ash and coarse-grained slag by means of a revolving screen, is separated further by centrifugal action in a rotating drum into coarse and medium material, the former

being then deposited in the separator containing the liquid. The separator contains an upper and lower screw. The upper screw enables the light, floating particles of coke to be removed, while the lower screw is for removing the heavier particles that fall to the bottom. This process requires about 2 h.p. per 100 c. ft. of crude slag per hour.

In another form of wet treatment, the separation is made by running water. In a plant for the treatment of 80 tons of residue in 12 hr. and requiring 12 h.p. and 700 c. ft. of water per hour, 24 per cent of large coke was recovered while 14 to 17 per cent was large slag. The mechanical treatment of the lighter portions separated it into 14 to 16 per cent small coke, 18 to 22 per cent small slag, and 22 to 25 per cent fine ash. Increase in the flue draught increased the proportion of fine ash owing to the improved combustion. The use of the fire stop and arch for reducing the amount of fine fuel which remains unburnt has already been described.

With fuels containing a large quantity of ash great care has to be taken in the removal of the slag. The use of a steam jet and air causes the slag to blister, so that it does not bake together in large compact masses. The cleansing of the grate of the large amounts of slag, which are often melted, increases the duration of the opening of the furnace doors and the supply of cold surplus air. For this reason alone, an automatic steady charging of the grate is much to be preferred to stoking by hand, as the latter necessitates the opening of the doors.

The consumption of fuels containing a large quantity of ash is of the order of 45 lb. per square foot per hour, while that of fuels containing considerable water may be 65 or even 100 lb. per square foot per hour. Both fuels containing water and fuels containing much ash require an under-draught of air. A small proportion of free grate, about 4 to 6 per cent, and a high pressure improve the combustion in the latter case. The air is introduced through narrow slots or holes, while the grate may be subdivided into sections, the draught through each being separately adjustable. The strength of the under-draught varies according to the type of fuel and the depth of the charge. It often exceeds 1 in. of water head while over the grate there may be $\frac{1}{25}$ in., and the flue draught lies between $\frac{1}{8}$ and $\frac{1}{5}$ in.

The presence of sulphur in the fuel has a deleterious effect on boilers and regenerators. Strongly alkaline ash attacks the brickwork of flues.

9. FURNACES FOR COAL-DUST AND REFUSE

In the coal-dust furnace, a considerable amount of energy is used in preparing and distributing the dust, so that in the furnace itself it is necessary to procure the highest possible efficiency by the use of the minimum amount of air necessary for complete combustion, and to use all mechanical aids effective for the same purpose. The coal-dust is separated by centrifugal force, or is reduced by rollers with suitable sieves, and is carried into the furnace by means of compressed air. The burners for coal-dust are simple,

all that is necessary in most cases being a round opening through which the dust is ejected. The remainder of the air required for combustion is introduced through a tube placed round the burner. Fig. 30 shows a simple arrangement for heating a boiler or industrial furnace. Fig. 31 gives the

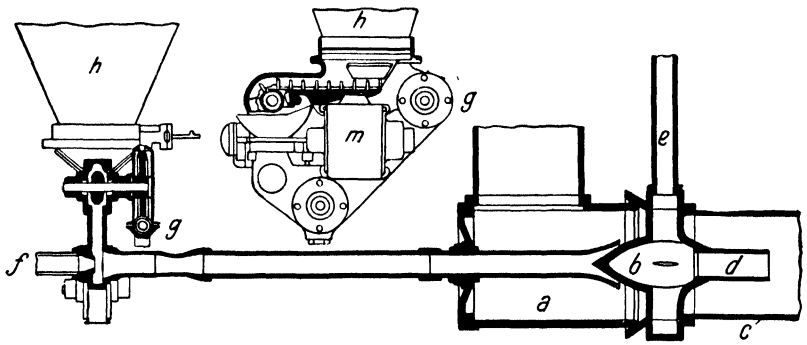


Fig. 30.—Furnace with Compressed Air and Coal-dust Injectors

arrangement of an ingot furnace, and fig. 32 a locomotive furnace. In fig. 30, *h* is the supply vessel containing the coal-dust. The endless conveyor *g* moves up the coal-dust to the nozzle of the blow-pipe *f*. Air at a low pressure issues from *a* and distributes the coal-dust-air mixture over the cone *b*, thus filling the pipe *c* uniformly with coal-dust. If the air velocities are not

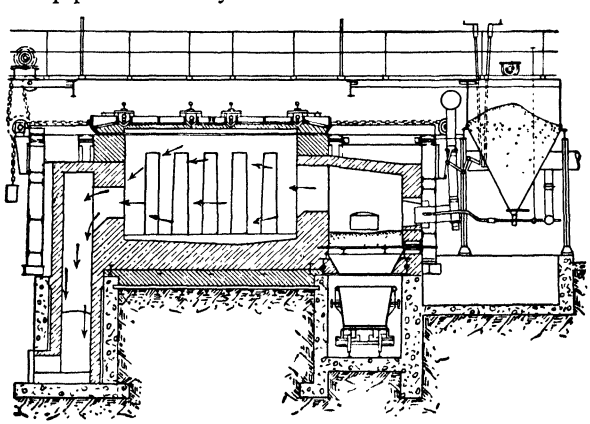


Fig. 31.—Reheating Ingot Oven with Coal-dust Furnace

high enough for the combustion chamber, additional highly compressed air can be supplied through *e*. From the last two figures it can be seen that the combustion requires an ignition and burning arch in which the combustion gases can be produced. The size of a suitable combustion chamber is in general 10 to 14 c. ft. for 1 ton of coal-dust per hour.

Damp coal cannot be used in this manner, a drying process being first

necessary. Drying is effected either by the waste gases or by a special furnace. The latter type of plant is seen in fig. 33, all points of interest being shown on the figure. In most cases up to 15 per cent of crude brown coal

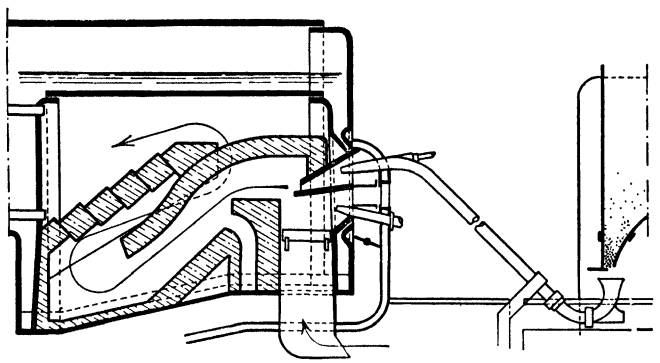


Fig. 32.—Peat-dust Furnace for Locomotive

is required for the drying, and a slightly lower percentage with mineral coal.

Coal-dust furnaces find an important application in cement factories with revolving furnace plants. An example is shown in fig. 34. The cement

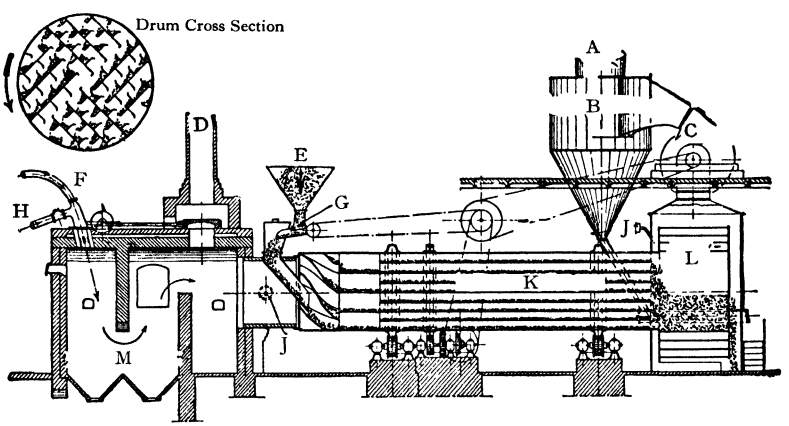


Fig. 33.—Drying Plant for Coal containing Water (*Decker & Co.*)

A, To the dust chamber. B, Dust collector. C, Ventilator. D, Emergency flue. E, Crude coal supply. F, Coal-dust. G, Measuring apparatus. H, Secondary air. J, Thermometer. K, Drying-drum. L, Collecting chamber. M, Fire chamber.

that is to be burnt is sintered in a rotating drum by means of the combustion gases of the coal-dust, which is injected at one end.

An important advantage of the dust furnace of modern design is the high temperatures which can be reached as a result of the small excess of air. With older boilers, the temperatures reached are less owing to the large

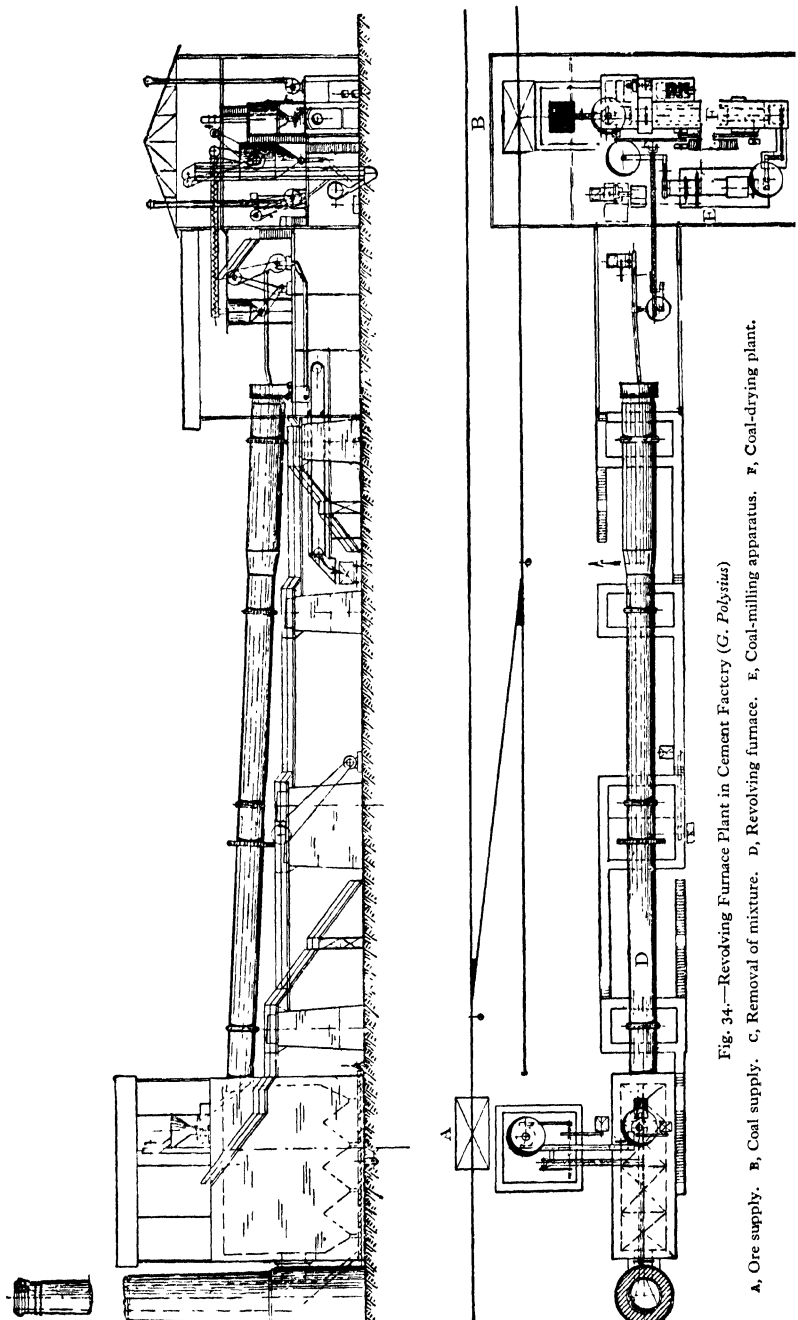


Fig. 34.—Revolving Furnace Plant in Cement Factory (*G. Polysius*)

A, Ore supply. B, Coal supply. C, Removal of mixture. D, Revolving furnace. E, Coal-milling apparatus. F, Coal-drying plant.

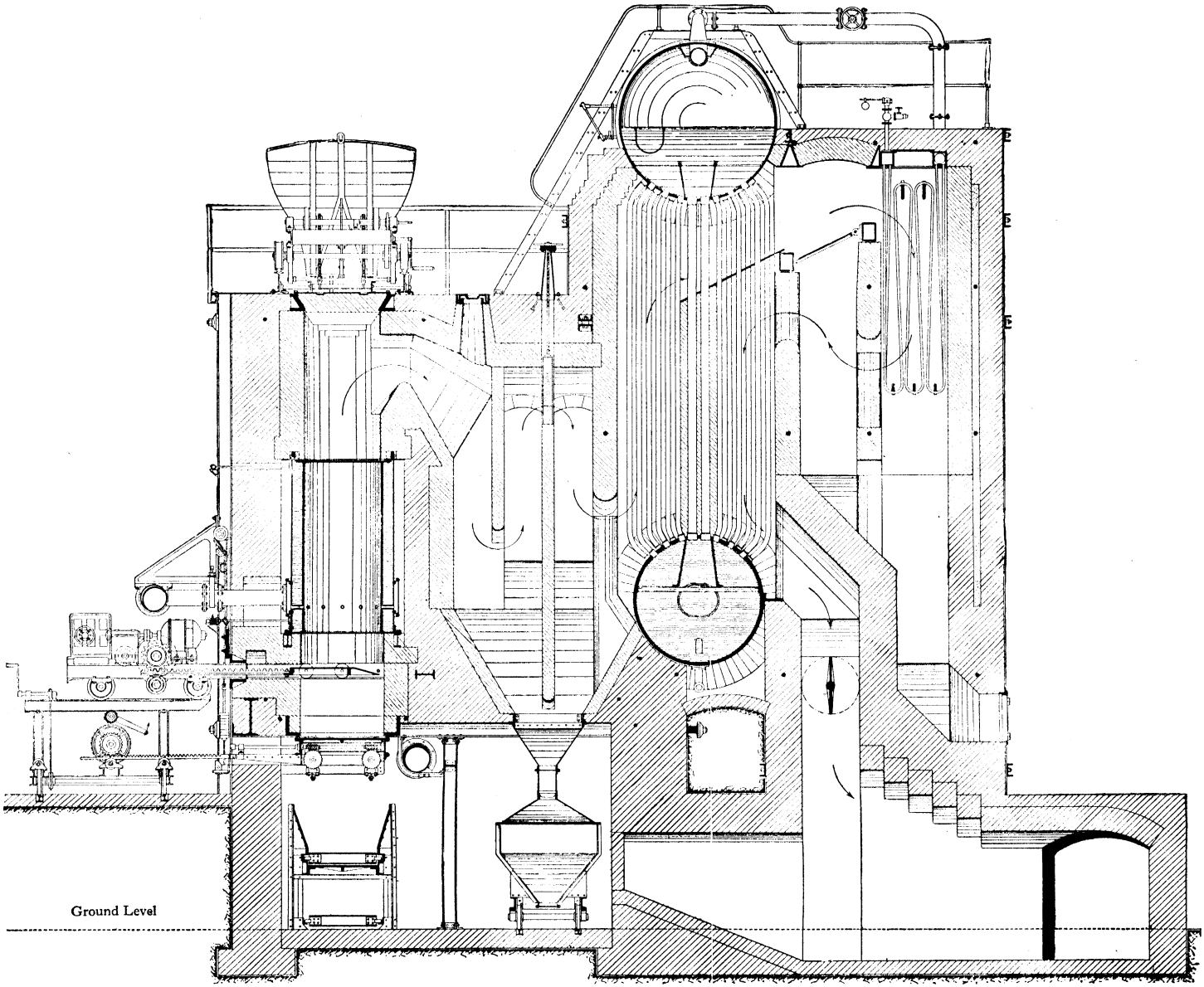


Fig. 35.—Refuse-burning Furnace

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amount of metallic surface in contact with the flames. Another advantage of the coal-dust furnace is that it can be easily adjusted by reducing or increasing the supply of fuel. It can thus be adapted to very varying conditions.

Among plants for burning waste substances, the most important are those for refuse from houses and manufactories. The average amount is estimated at 1 lb. for each individual per day. The deposit of unburnt coal varies for each district, e.g. in London it is 18 per cent and in Berlin 1 per cent. It also depends on the furnace system and the kind of manufactories. The refuse can be used to produce steam or it can be made into bricks by burning it with an addition of lime.

The arrangement of a refuse-burning plant for producing steam is seen in fig. 35. The furnace is set up for burning 25 tons of refuse per day in the city of Wiesbaden. This plant produces 0·4 to 0·8 lb. of steam per pound of refuse burnt. If the plant is used to operate an electric generator, 55 to 75 lb. of refuse produce 1 kw.-hr., the internal consumption of the plant being 15 to 25 per cent of the electrical energy generated.

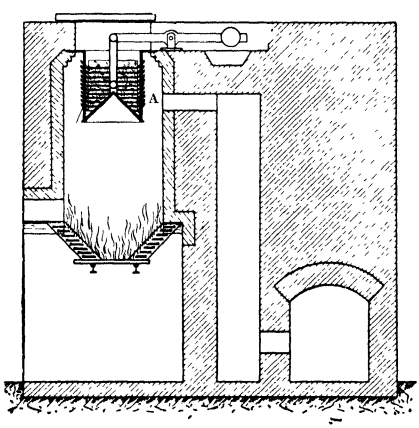


Fig. 36.—Refuse-burning Furnace
A, Funnel for preliminary drying

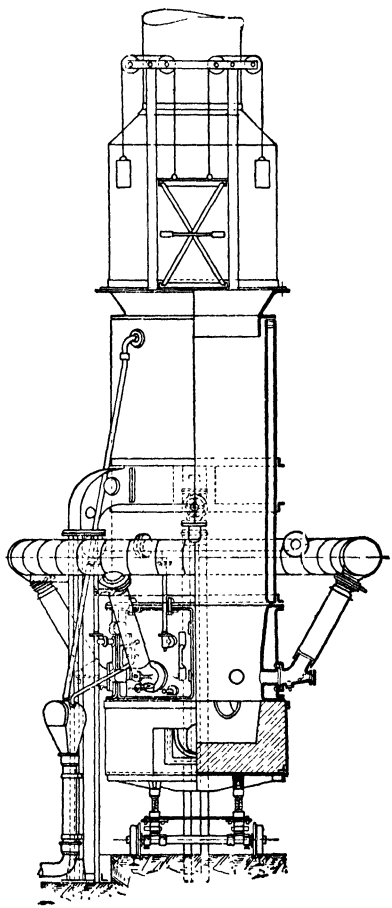


Fig. 37.—Water-jacket Furnace

Figs. 36 and 37 show a plant for the production of building blocks from slag. The refuse is sifted and the coarser part is burnt in the furnace where the bricks made with the finer parts are dried, fig. 36, the hot gases passing out for utilization elsewhere. The finer parts are made into briquettes by

adding the ash of the refuse, 10 per cent of coke, and some lime, according to the coal content of the refuse. The blocks are dried and then burnt in the furnace, fig. 37. In a plant consisting of six such units, 50 tons of refuse are treated in 24 hr. The sieve droppings amount to about one-third.

The temperature reached in burning the refuse is about 1100° F. One means of obtaining higher heat values is by sifting the refuse. The calorific value of the various grades can be seen by the following table.

Percentage Removed.		Calorific Value, B.Th.U. per Lb.
0	....	1500
20	....	1750
40	....	2100
60	....	1800
70	....	1500

Another means is the utilization of hot air, which enables a larger proportion of the refuse to be burnt. The heating of the air to approximately 200° F., 400° F., and 600° F., gives an increased utilization of the refuse of 8, 20, and 30 per cent in the respective cases.

10. COMBUSTION OF LIQUID FUELS

The combustion of liquid fuels is becoming more and more general, in the first place because there is a plentiful supply of natural and artificial liquid fuels, and secondly because the combustion of liquid substances can

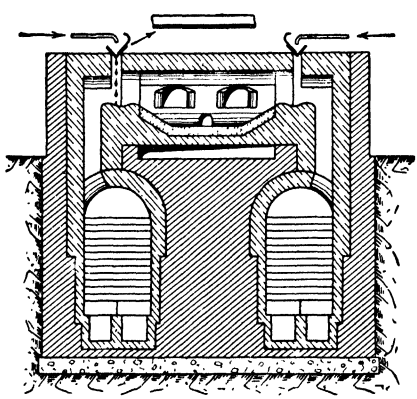


Fig. 38.—Siemens-Martin Oil Furnace with Drip Feed

be carried through with a high efficiency. Further, the calorific value of 1 lb. of liquid fuel is very high, being 16,000 to 20,000 B.Th.U., while the supply and regulation of liquid fuel can be arranged effectively.

Furnaces for liquid fuels are found in nearly all branches of industry. There are two possible ways of supplying oil to the furnace. The first way

is to allow the oil to drip into the fire box at atmospheric pressure, part of the oil being gasified before burning and part burning as liquid drops. A variant of the same method is to allow the oil to first fall on to a hot plate where it is vaporized, and the combustion of the vapour may take place

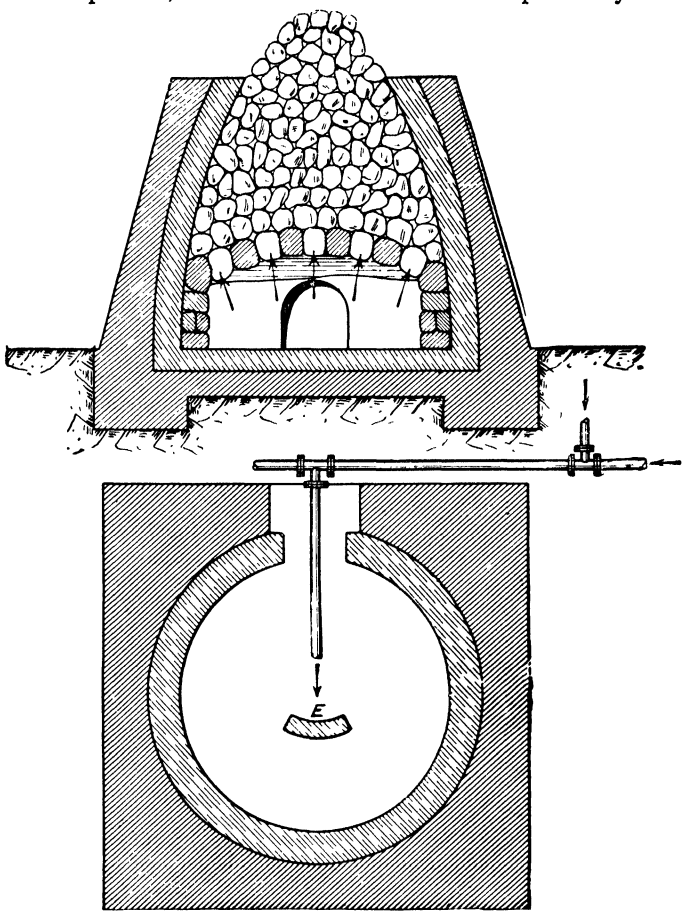


Fig. 39.—Baker Lime Kiln using Compressed Oil

immediately over the plate or in an adjacent combustion chamber to which the vapour is led.

Fig. 38 shows a Siemens-Martin oil furnace, in which there is a mixed combustion of liquid and vapour. Oil furnaces with liquid and vapour combustion are also used for forges, oven heating, and muffle furnaces. The starting is the difficult factor in these furnaces. The furnace is brought to a glowing heat by means of coal, blast-furnace, or generator gas, the gas supply is then cut off, and subsequent firing is effected by oil. An alter-

native method of starting is to raise one portion of the plate to the ignition temperature of the oil by means of a small supply of fuel of lower ignition temperature. Hydrogen or carbide blow-pipes can also be used for raising

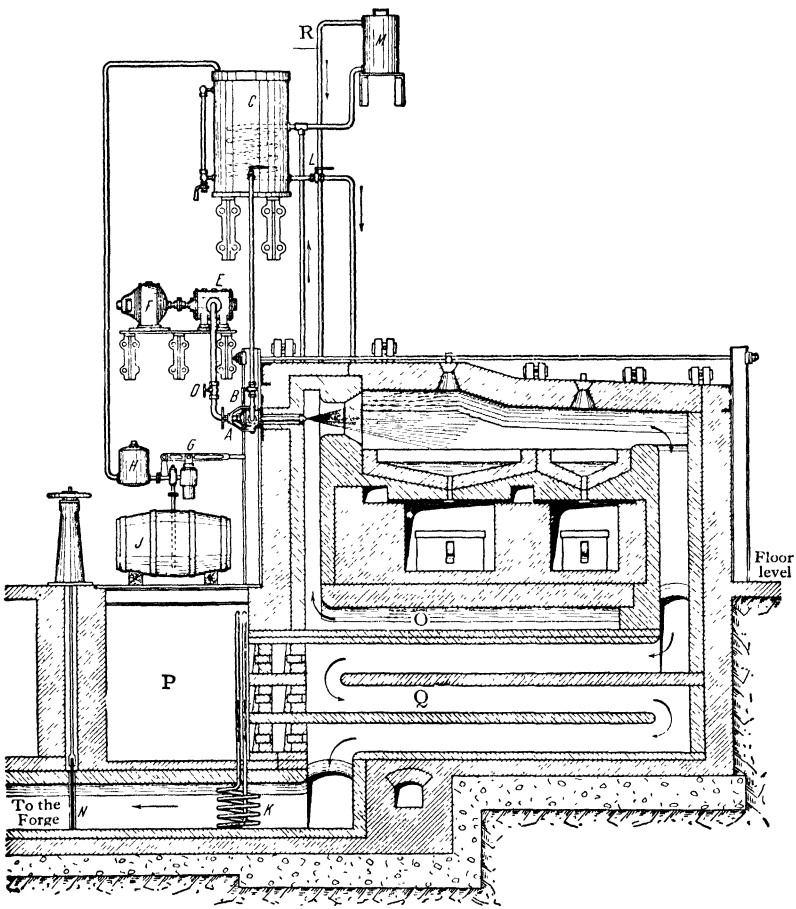


Fig. 40.—Enamel Smelting Oven with Oil Furnace

A, Atomizer. B, Oil Stop-cock. C, Oil tank. D, Air valve. E, Rotary blower. F, Electric motor. G, Hand-pump. H, Air-drum. J, Oil vessel. K, Warm water heating coil. L, Warm water stop-cock. M, Expansion chamber. N, Fuel gas regulating slide. O, Air. P, Cleaning tank. Q, Recuperator. R, Overflow pipe.

the temperature of the vaporizing plate from a cold state to a glowing heat for the purpose of oil firing.

The combustion itself takes place fairly slowly as the air, the drops of oil, and the oil vapour are mixed together slowly with a small natural draught. In industrial furnaces the air must be pre-heated. By changing the direction of the flame an eddying formation and a thorough mixing of oil, oil vapour,

and air is obtained, giving rise to a complete combustion. Practical experiments have shown that a maximum of 80 to 100 lb. of oil per hour can be supplied through a single pipe and complete combustion attained.

In order, therefore, to obtain greater amounts of heat and a more rapid combustion, a second method is used, the oil being burnt in a finely atomized state. This is effected by means of a sprayer. The oil is forced out of the atomizer by means of a pump or by compressed air, and is then burnt directly in a long jet or after striking against an incandescent wall, or the oil is mixed with compressed air or steam in the atomizer, on issuing from which it burns with a long flame. The latter arrangement is analogous to the dust furnace. In fig. 39 is shown a furnace using compressed oil without an air feed in the burner. The figure represents a lime kiln, in which the oil is directed against the fireclay wall E. An enamel smelting furnace is shown in fig. 40, with the entire oil-supply arrangements. Fig. 41 shows the arrangement for a lime-burning furnace, the oil being introduced at A. Fig. 42 shows a calcination furnace.

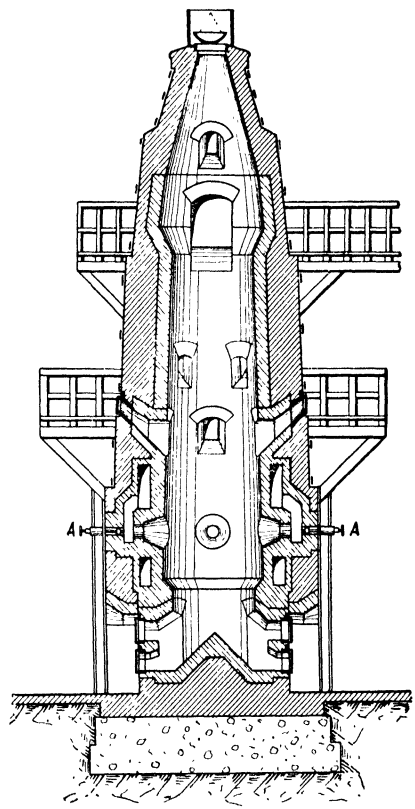


Fig. 41.—Lime Kiln with Oil Furnace

Oil fuel is also used for smelting furnaces. Fig. 44 shows the arrangement of a crucible smelting furnace with which the fuel may be either oil or melted naphthalene. The crucible is shown in fig. 45, the jets of fuel being introduced in a tangential direction through the orifices A. The liquifier for the naphthalene is shown in more detail in fig. 46. In a test of this furnace, five smeltings were completed in 6 hr. 40 min.: (1) 1 ton of steel bronze, (2) 1·2 tons bronze, (3) 1 ton nickel bronze, (4) 1 ton nickel bronze, (5) 1 ton steel bronze. The fuel consumption, including the preheating, was 850 lb. of oil or 7·5 lb. to 100 lb. of charge. Fig. 47 shows a regenerator furnace using oil fuel. Steam boilers heated by oil are now used very widely, a plant of this kind being shown in fig. 48.

Oil fuel is largely used in combination with low-percentage solid fuel. The oil is atomized and distributed over the whole grate, so that the incandescent burnt oil gases come into direct contact with the upper layer of

solid fuel, thus producing more efficient combustion owing to the higher temperature. The addition of the oil fuel enables a greater output of steam

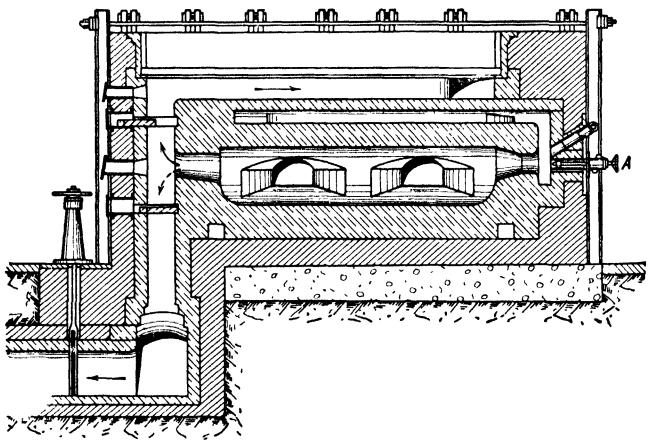


Fig. 42.—Calcination Furnace Oil Fired

to be obtained from the same plant than that which could be obtained from the low-percentage fuel only. Fig. 43 represents the oil furnace of a ship's

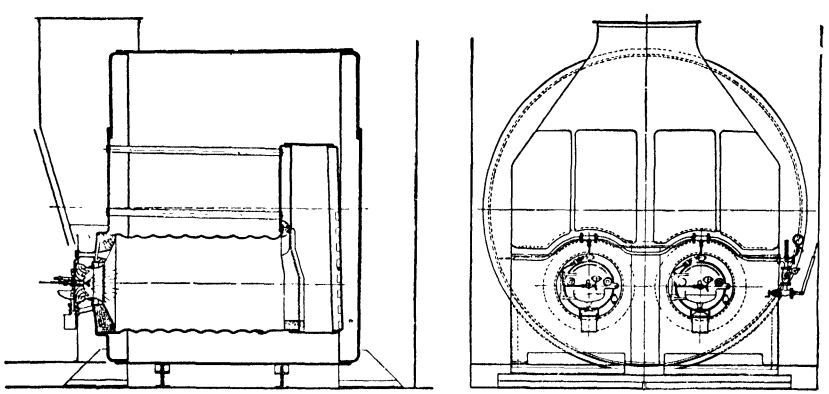


Fig. 43.—Marine (Scotch) Boiler with Interchangeable Coal and Oil Firing

boiler. The oil furnace can easily be removed and a coal furnace substituted when desirable. In many ships, some boilers are heated with oil fuel and some with coal, the former being used when it is required to raise steam rapidly. The higher cost of oil acts as a deterrent to the complete transformation from coal to oil firing in the case of slow ships. Where low running costs are not the deciding factor, e.g. in the case of liners, the ease of re-fuelling and the greater cleanliness of the process as well as the saving in the labour of stoking are sufficient to overcome this disadvantage.

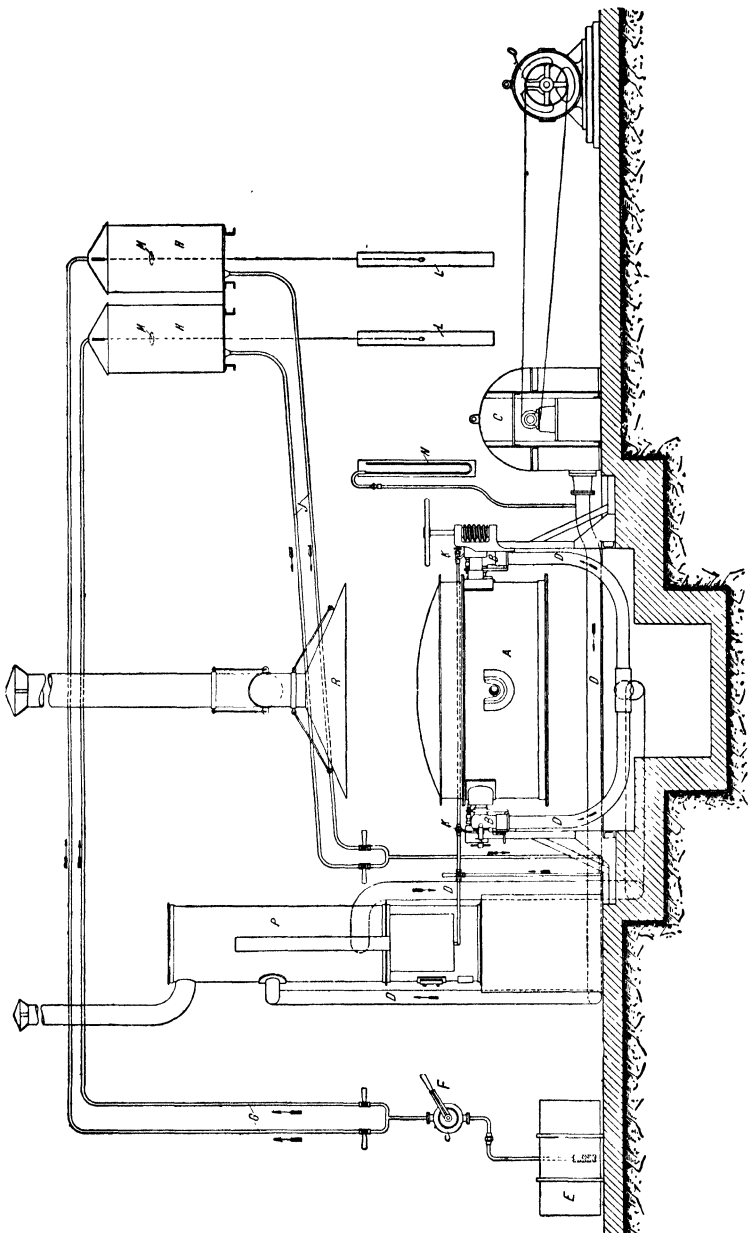


Fig. 44.—Smelting Plant with Oil or Crude Naphthalene Furnace

A, Smelting furnace with a capacity of 1 ton. B, Burner. C, Blower. D, Air pipe. E, Oil tank. F, Hand-pump. G, Oil pipes to the oil containers. H, Oil container. J, Oil pipes to burners. K, Regulating valve. L, Oil level meter. M, Float. N, Air-pressure gauge. O, Motor, 3·5–4 h.p. P, Crude naphthalene liquifier. R, Adjustable hood.

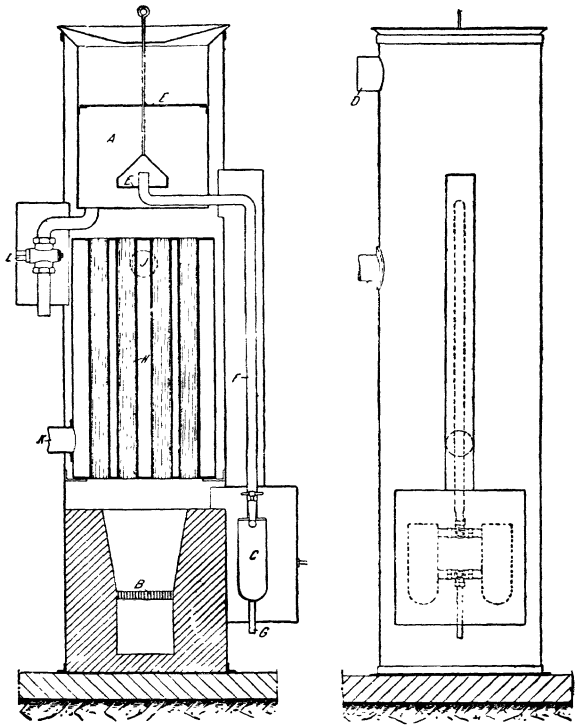


Fig. 46.—Crude Naphthalene Liquifier

A, Receiving tank. B, Grate. C, Adjustable strainer. D, Discharge tube. E, Screen. F, Connecting pipe from tank to strainer. G, Connecting pipe from strainer to the place of utilization. H, Air pre-heater. J, Cold air inlet. K, Hot air outlet. I. Shutting-off valve.

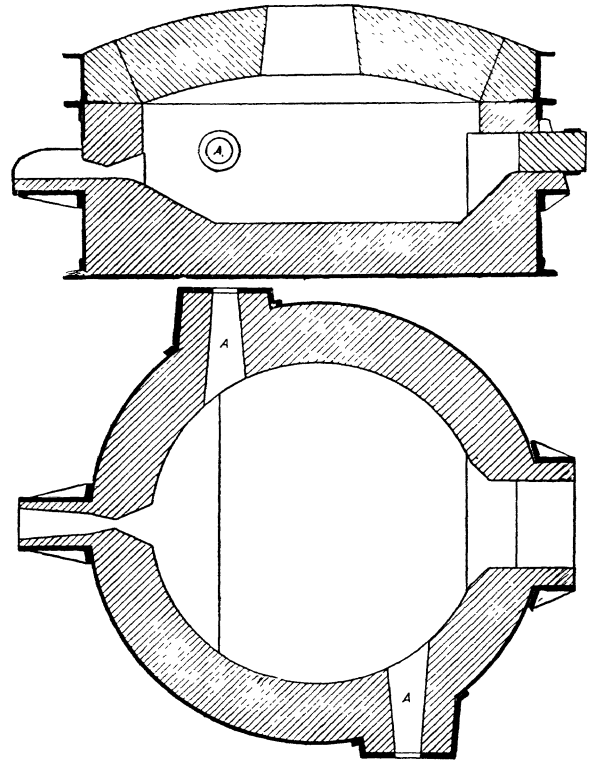
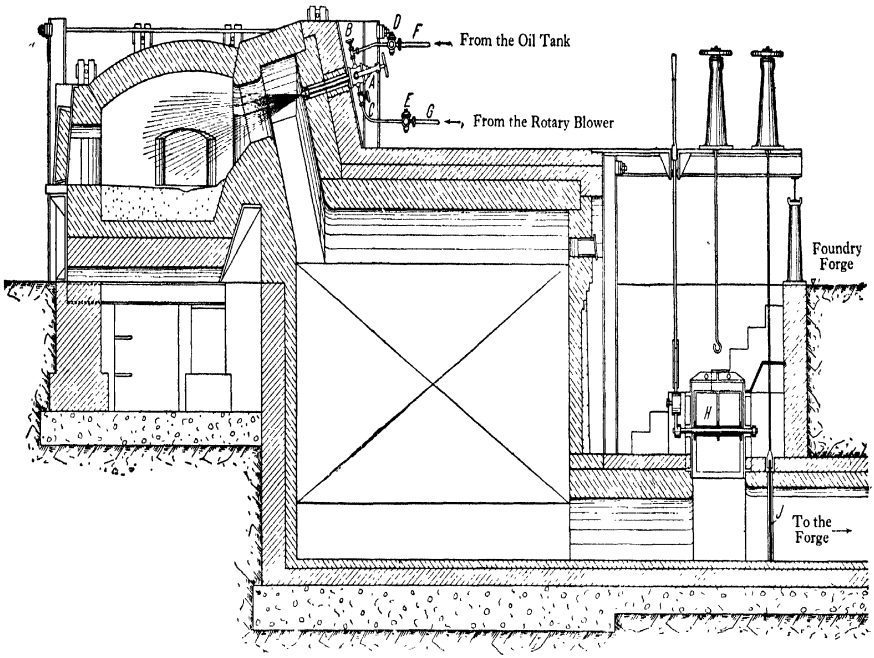
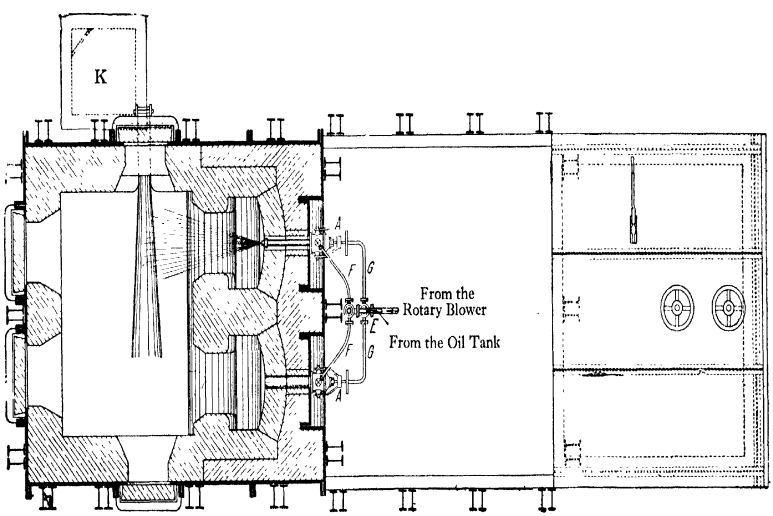


Fig. 45.—Smelting Furnace Crucible



Sectional Elevation



Ground Plan

Fig. 47.—Regenerator Furnace Oil Fired

A, Compressed air atomizer. B, Needle valve. C, Air stop-valve. D, Three-way cock for oil. E, Three-way cock for air. F, Oil pipe. G, Air pipe. H, Air damper. J, Flue gas regulating slide. K, Delivery shaft

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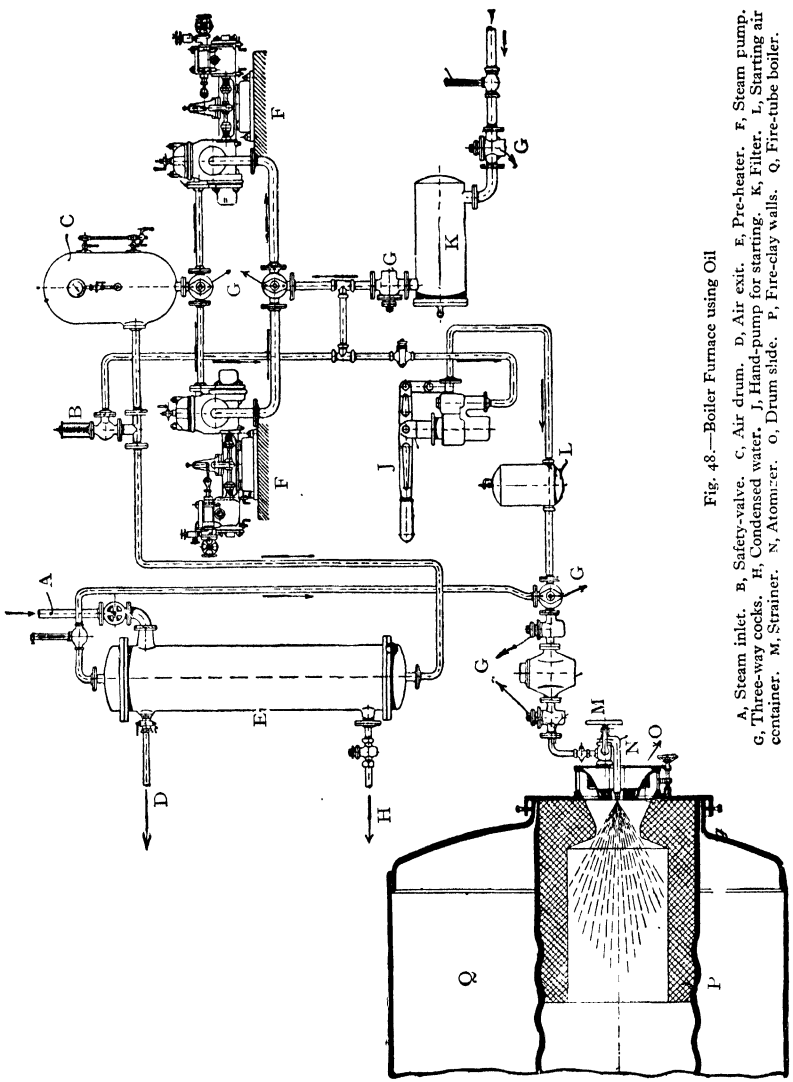


Fig. 48.—Boiler Furnace using Oil

A, Steam inlet. B, Safety-valve. C, Air drum. D, Air exit. E, Pre-heater. F, Steam pump. G, Three-way cocks. H, Condensed water. J, Hand-pump for starting. K, Filter. L, Starting air container. M, Strainer. N, Atomizer. O, Drum slide. P, Fire-clay walls. Q, Fire-tube boiler.

11. OIL ATOMIZERS

The atomizers may be divided into two classes, (*a*) compression atomizers and (*b*) air and vapour atomizers. In the first type the oil is ejected from a

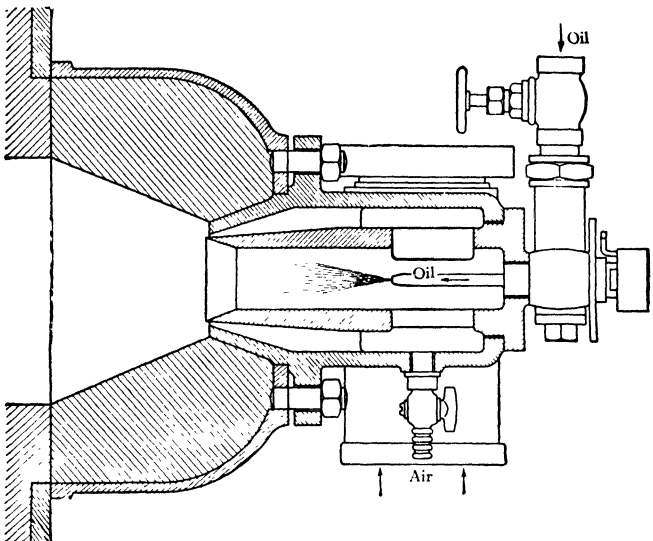


Fig. 49.—Compression Atomizer

jet under considerable pressure created by compressed air or by piston or centrifugal pumps. An atomizer of this type is represented in fig. 49, while fig. 50 shows the heads of atomizers. In the atomizer shown in fig. 50, the pressure is 35 lb. per square inch, and the output per hour 65 lb. of tar oil at 60° F. As seen in fig. 49, the combustion air is supplied through an outer nozzle which is subject to an air pressure of 0·3 to 0·6 lb. per square inch. These atomizers are, however, usually worked with an air flow under natural draught. The disadvantage of this type of injection is that a thorough mixing of the oil and combustion air only takes place in the combustion chamber, and even then is not carried out quite efficiently. In general, compression atomizers are only used for outputs of over 100 lb. of oil per hour. Fig. 51 shows a Körting atomizer plant which is in general use for ships.

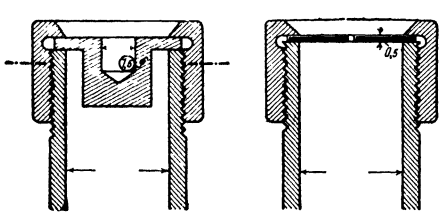


Fig. 50.—Atomizers

In the second type of atomizer, mixing of the oil and air takes place before the combustion process. Atomizers of this type may be divided into low-compression atomizers using air pressures below 0·1 atmosphere, high-

compression atomizers with air pressures above 0·3 atmospheres, and steam atomizers. Examples are shown in figs. 52 and 53. In the atomizer shown in fig. 52, the oil flows into a current of air, mixes with it, and flows out through the nozzle where it unites with more air from the concentric jet. In fig. 53, compressed air or air and vapour flows direct to the outlet nozzle and forms a spiral round the outflowing jet of oil. By this means a breaking up of the jet and a thorough mixing of oil and air is obtained. The vapour for the oil atomization must be subjected to a strong superheating and a pressure of about 100 lb. per square inch.

The work of atomization in the burners can be effected by means either

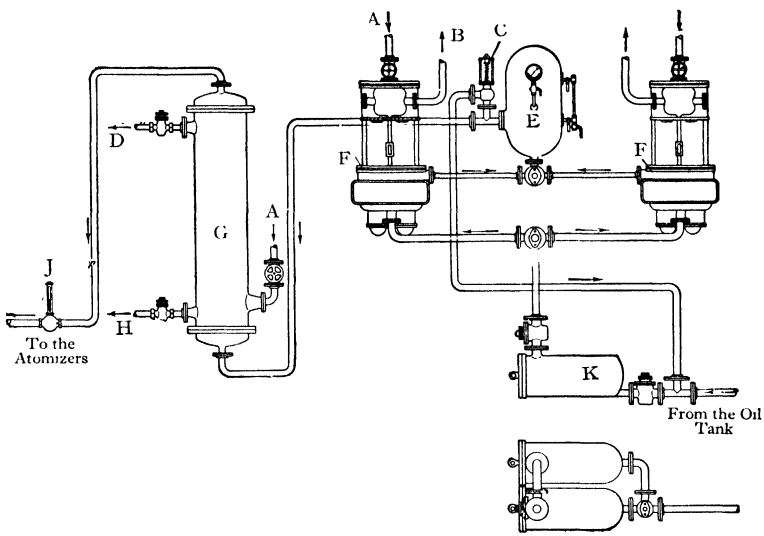


Fig. 51.—Korting Compression Atomizer Plant

A, Steam inlet. B, Exhaust. C, Safety overflow valve. D, Air exit. E, Air drum. F, Oil pump. G, Pressure pre-heater. H, Water exit. J, Thermometer. K, Filtering system.

of a large quantity of air at low pressure or of a small quantity of air at high pressure. The burner is adjusted by throttling the air, this being possible within narrow limits with low-pressure burners, and within wide limits with high-pressure burners. If p_1 represents the air pressure available and p_2 the lowest air pressure at which the burner works, then since the output is approximately proportional to the velocity, the adjustability

$$x = \sqrt{\frac{p_1}{p_2}}.$$

The work of atomization is performed by the pressure of the oil forced out of the nozzle in compression atomizers, and by the air or the vapour in the second type of atomizers. In the former, with an oil pressure of 70 lb. per square inch, the work expended in the atomization of 1 c. ft. of oil is 10,080 ft.-lb. With low-pressure air atomization at $1\frac{1}{2}$ lb. per square inch,

50 per cent of the air required for the combustion of 1 c. ft. of oil amounting to 10,000 c. ft. is supplied by the atomizing air. Thus the work expended on the atomization of 1 c. ft. of oil is 2×10^6 ft.-lb. An output of 1 c. ft. per hour thus requires 1 h.p., and 1 gall. per hour a sixth of this amount.

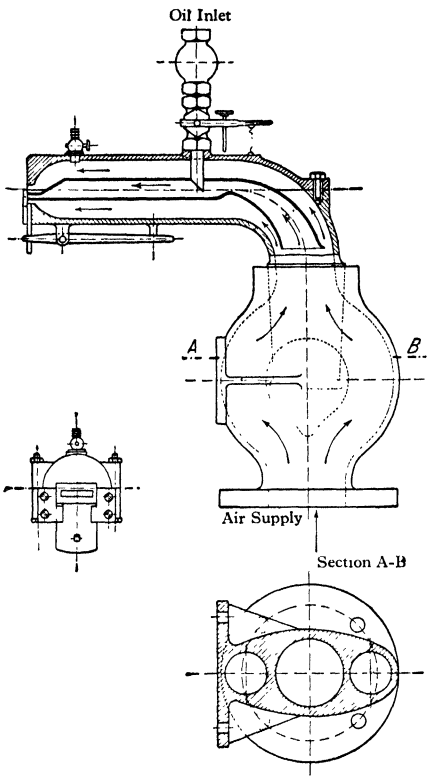


Fig. 52.—Low-pressure Atomizer (*Lochner*)

Low-pressure burners are often used because low-pressure air is more easily obtained. The amount of solid cone-shaped flame is less but they have less adjustability. The latter quality can be increased by the proportion of air supply external to the fuel jet. Where large adjustability is required, high-pressure air atomizers should be used, and the solid flame can be reduced if desired by the use of air supply internal to the jet. Thus for the most efficient burners both low- and high-pressure air is necessary. The burner itself may either pass into the furnace in close contact with the wall or through an opening of larger diameter. In the latter case a portion of the combustion air is drawn into the furnace by injector action, while in the former case all the combustion air must be supplied through

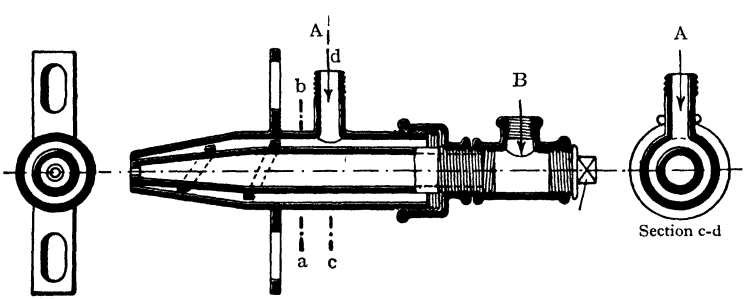


Fig. 53.—Air or Steam Atomizer (*Burgers & Co.*)
A, Air inlet. B, Oil inlet. C, Cleansing device

the atomizer. Care should always be taken to thoroughly clean the burner and particularly the oil pipe.

The action experienced by the oil may be divided into three processes: (*a*) atomization, (*b*) gasification, (*c*) combustion. The completeness of the atomization or the breaking up of the oil into fine particles depends upon the shape of the burner near the exit, the velocity of the oil (which of course is determined by the air pressure on it), and the character of the swirl of air into which the oil issues. The most satisfactory conditions have been found by trial and error.

The gasification is effected by the heat of the flames and the walls surrounding the flames. As soon as the ignition temperature is reached, combustion takes place. The maintenance of a steady flame, however, also depends on the ignition velocity. As the gas jet widens, a point is reached where the ignition velocity is equal to the flow velocity, and here flames begin to form. Fig. 54 illustrates the conditions. In the zone nearest to the exit from the burner there is a partial vaporization of the smallest liquid particles. At the end of this zone, where the ignition velocity becomes equal to the flow velocity, the combustion begins. At the same time the gasification of the remaining drops of oil takes place over a certain space, the length of which is greater according to the incompleteness of the atomization. The light oil is vaporized first and then the heavy oil. In most cases a small residue of carbon remains. If the latter is in suspension and there is sufficient air present it burns. If an oil particle meets the incandescent walls, vaporization follows; the carbon particle, however, remains on the wall, and unless the temperature is sufficiently high may not burn even with a large excess of air. For this reason the atomized flow must be disposed so that no unvaporized oil meets the wall.

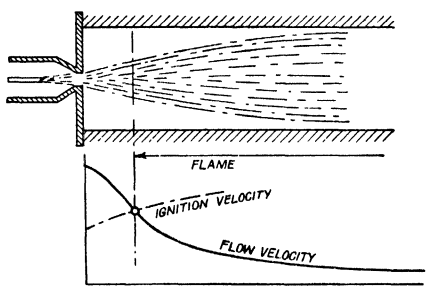


Fig. 54.—Ignition Diagram

If d is the diameter of the oil particle, then if c is a constant, the time of combustion t is:

$$t = c_1 d^2.$$

The path s of the particle of oil moving at a mean velocity v is:

$$s = vt = c_1 v d^2.$$

If f is the cross-section of the combustion chamber, and if c_2 is a constant, the quantity of oil burned is:

$$Q = c_2 f v,$$

and the space used for the combustion is:

$$V = fs = c_1 v d^2 f;$$

also:

$$V = c_3 Q d^2,$$

i.e. the volume increases with the quantity of oil and becomes smaller according to the completeness of the atomization of the oil.

The great advantage of steam boilers heated by oil lies in the simplicity of stoking and the saving in the number of men required for this purpose. In addition, particularly in the case of steamship boilers, there is a considerable saving of weight and space owing to the higher calorific value of oil over coal. As an example of this, the consumptions of certain steamship boilers for the journey from Hamburg to New York are given in the table below, when using coal and when using oil.

3 cylindrical boilers with 3 furnaces; heating surface, 7060 sq. ft.	Coal:	665 tons;	30,000 c. ft.
	Oil:	465 ,,	16,500 ,,
5 cylindrical boilers with 3 furnaces; heating surface, 14,600 sq. ft.	Coal:	1085 tons;	49,000 c. ft.
	Oil:	758 ,,	26,700 ,,
46 water-tube boilers; heating surface, 203,000 sq. ft.	Coal:	8600 tons;	388,000 c. ft.
	Oil:	6000 ,,	212,000 ,,

12. GAS FURNACES

The combustion of gaseous fuels arranged with the exact quantity of air required for perfect combustion can be carried out with a much higher efficiency than that of solid fuels in a grate. As heat, however, must have been expended in the gasification process (except in cases where natural gas is available), the overall efficiency is considerably less than the apparent efficiency of the gaseous combustion only. Boilers, muffles, and other furnaces which are

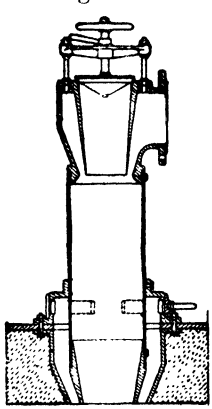


Fig. 55.—Terbeek Gas Burner

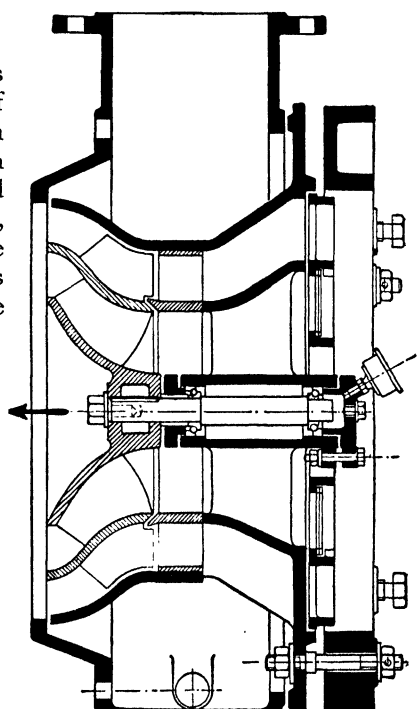


Fig. 56.—Eickworth Gas Burner

heated wholly by gases may be divided into three sections: (1) those in which cold gas is burnt, (2) those heated principally by the sensible heat from other furnaces, (3) those heated by gas from adjoining gas producers, both sensible heat and calorific value being of importance. In some cases the waste gases from other furnaces may retain a proportion of combustible gas from which some calorific value may be obtained. The burners used may consist of a gas pipe only or of gas and air pipes. The former kind obviously is all that is necessary where the sensible heat of waste gases only is being utilized, but they are also used in certain cases of combustible gas where low temperatures only are required, the air necessary for combustion being introduced separately into the

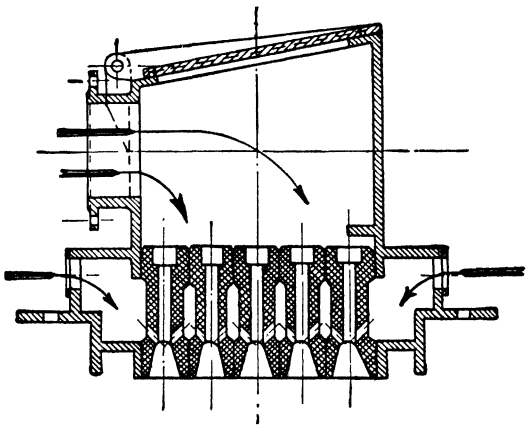


Fig. 57.—Pellix Gas Burner

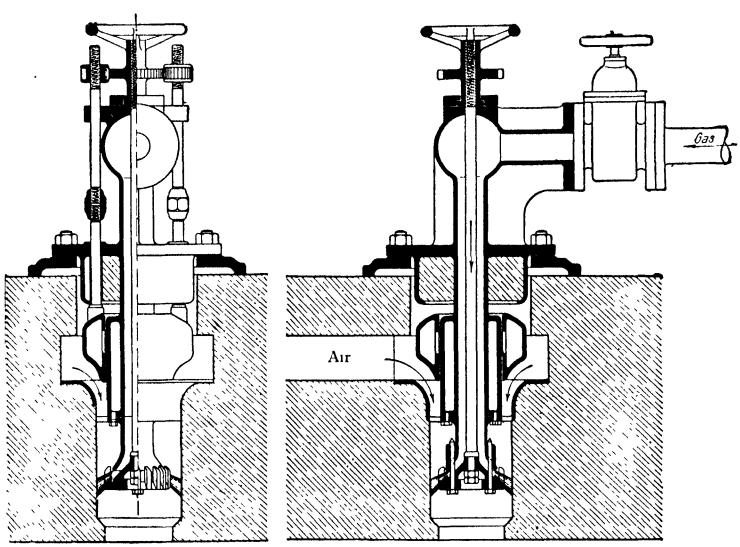


Fig. 58.—Gas Burner

heating chamber. They consist of a single pipe with a contraction at the exit to increase the gas velocity, directed to the heating surface.

The burners may be divided into two main types; those for gases of

high calorific value between 130 and 550 B.Th.U. per cubic foot, and those for gases of low calorific value between 80 and 130 B.Th.U. per cubic foot. The burners are designed on the Bunsen principle and are illustrated in figs. 55, 56, and 57. Low-value gases require a small air supply, the Terbeek

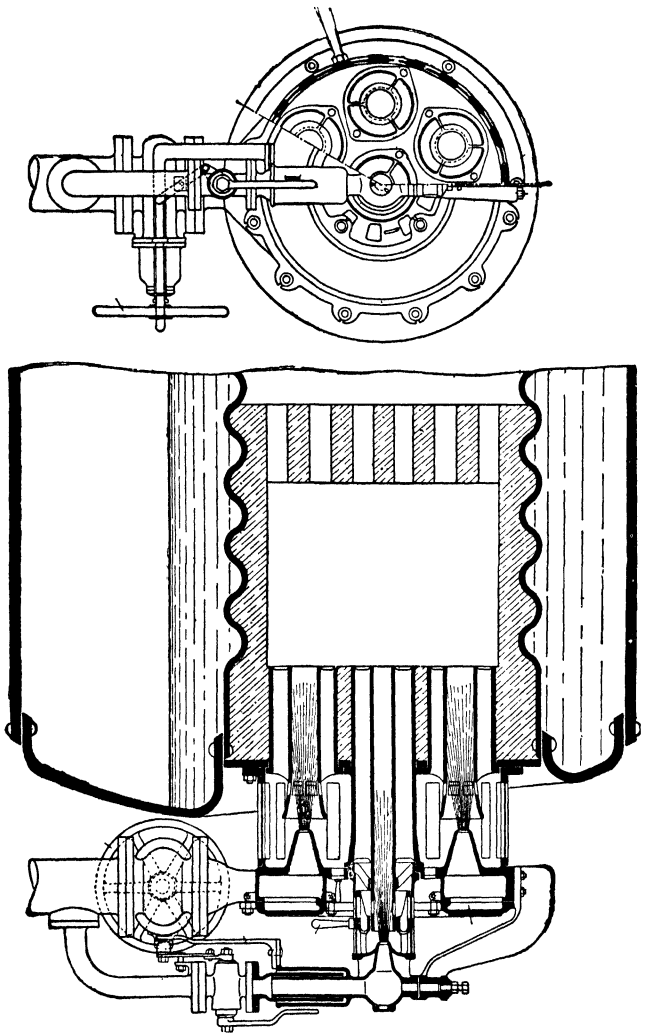


Fig. 59.—The Application of Gas Heating to a simple Boiler

burner being particularly suitable. With blast-furnace gas of about 110 B.Th.U. per cubic foot and containing 5·8 per cent CO_2, 30·8 per cent CO, 0·6 per cent CH_4, 6·8 per cent H_2, 0·2 per cent O_2, and 55·8 per cent N_2, it has a consumption of 72·5 c. ft. per square foot per hour and a steam yield of 5·2 lb. per square foot per hour. The Stirling boiler used in this con-

nexion developed steam at 200 lb. per square inch, and had a heating surface of 3800 sq. ft. and 1000 sq. ft. superheating surface. After the first nest of boiler tubes, the gas contained 21·5 per cent CO_2, 0·3 per cent CO, 0·9 per cent O_2, and had a temperature of 750° F. behind the boiler, and 400° F. behind the superheater. Of the calorific value of the gas 79·5 per cent was utilized, 8 per cent remaining in the gases as they passed into the chimney, while 12·5 per cent was lost by radiation and conduction. The Bone-Schnabel boiler for steam at 120 lb. per square inch, with a heating surface of 210 sq. ft. and using a gas of 515 B.Th.U. per cubic foot requires 69 c. ft. per square foot per hour for a steam yield of 310 lb. The useful heat amounts to 93·8 per cent.

An Eickworth burner is shown in fig. 56. The gas entering through the pipe presses against the outer blade ring and sets it in motion. The primary combustion air is drawn in through these blades and mixes with the gas. The secondary combustion air is drawn in by the inner blade of the rotating wheel, compressed, and mixed with the gas. The wheel is designed to give a constant ratio of gas to air. An external slide valve adjusts the air supply according to the type of gas. Tests have shown that it can be used with gas pressures ranging between 0·8 in. and 3 in. of water, the gas volume used increasing from 20,000 to 30,000 c. ft. per hour. With a double-flue boiler the following results were obtained:

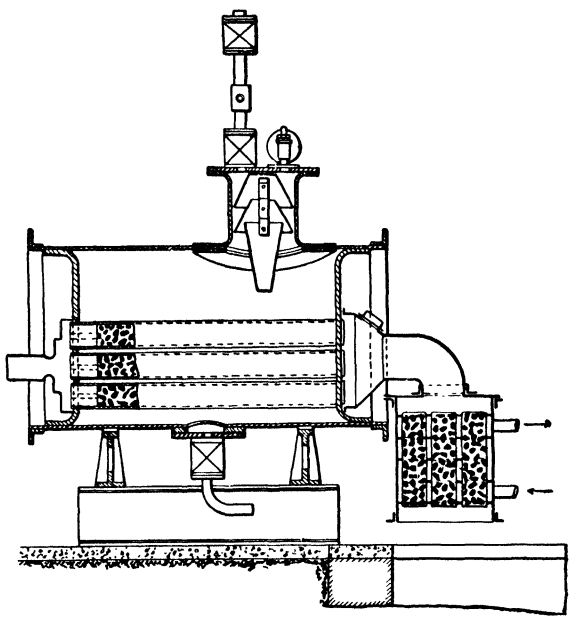


Fig. 60.—Boiler arranged for Surface Combustion

Consumption of blast-furnace gas for 2 burners per hour, 55,000 c. ft.
Calorific value of gas, 106 B.Th.U. per cubic foot.
Heating area, 985 sq. ft.
Steam per square foot, 3·7 lb.
Steam per cubic foot of gas, 0·068 lb.
CO_2 contents of flue gas, 24·3 per cent.
Temperature of flue gas (short boiler), 600° F.
Boiler efficiency, 74·5 per cent.

Fig. 57 shows the Pellix burner. It is made of cast iron, the nozzles being made of fireproof material. Its design enables a thorough mixing of

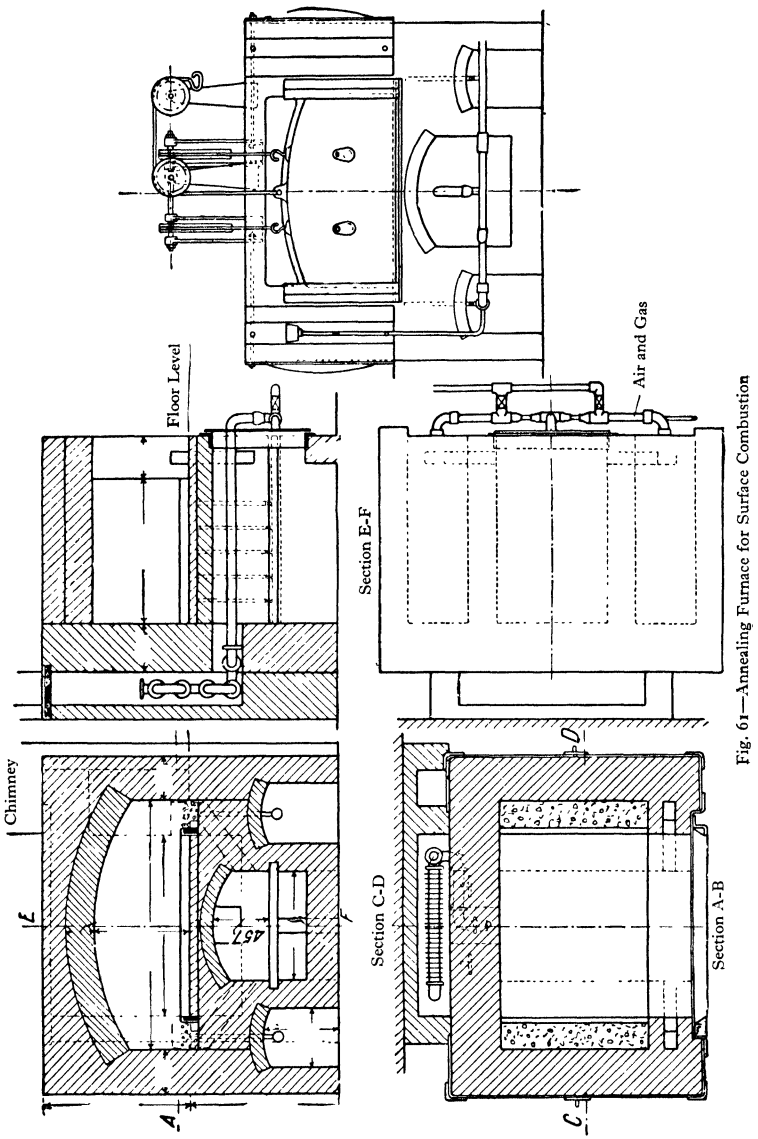


Fig. 61—Annealing Furnace for Surface Combustion

gas and air to take place, so that a complete combustion and a maximum temperature are obtained. The shape of the nozzle prevents the flame from breaking away.

Another type of burner is shown in fig. 58, the gas and air regulation being obvious from the diagram.

Burners used for heating industrial furnaces are made of firebrick. The gas and air are supplied in separate pipes. The two are mixed either by the gas and air particles flowing into one another, or by the mixture flowing through a fairly fine-meshed grating. The combustion point, i.e. the point of maximum temperature, varies its position in the furnace according to the direction and velocity of the air and gas flow. This point is of importance, e.g. in reverberatory and welding furnaces, as the object to be treated can derive the maximum heat at or above this point, while the parts lying near are pre-heated by the heat of the escaping gases.

The velocity of the gas and air in the feed pipes to the burners varies between 4 and 60 ft. per second; the velocity in the burner itself is usually

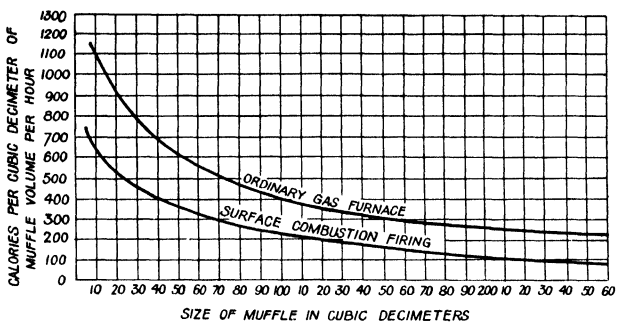


Fig. 62.—Gas Consumption Curves

greater for air than for gas, and in Siemens-Martin furnaces reaches 60 to 160 ft. per second, and in glass furnaces it is 12 to 40 ft. per second. The values for other types of furnace lie between these limits, except in re-heating and annealing furnaces, when they are somewhat lower.

The application of gas heating to a simple boiler is shown in fig. 59. Three gas jets are shown issuing into a preliminary chamber for efficient mixing and distribution of the gases over the whole cross-section. The burning gases then pass by a number of short passages into the flue, where the heat transfer occurs as with a coal fire.

A furnace of a different character is obtained with what is known as flameless combustion. An application of this to boiler heating is seen in fig. 60, while an annealing furnace is shown in fig. 61. A mixture of air and combustible gas is conducted through a porous layer of incandescent fire-proof substance, complete combustion being secured without the formation of flames. The combustion takes place with almost the exact quantity of air required in theory. Heat transmission then follows through conduction and radiation. Experiments have shown that for the same output, the gas consumed in combustion without flames is less than that with flames. A comparison of the consumption curves is given in fig. 62.

If, as stated, the combustion takes place almost without any excess of air, it follows that high combustion temperatures and efficiencies will be obtained.

Owing to the air and gas flowing through the porous fireclay walls, a

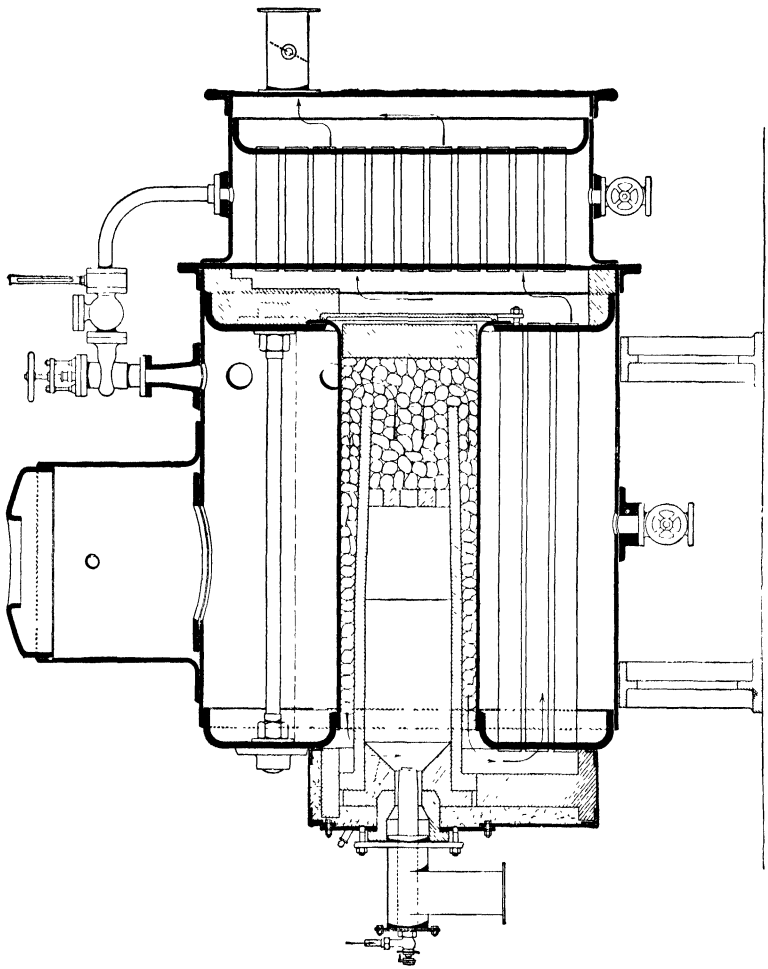


Fig. 63.—Surface Combustion Boiler (*Bone-Schnabel*)

thorough mixing takes place which enables every oxygen molecule to combine with a gas molecule. The walls of fireclay are cold when the furnace is ignited. The heat in the combustion chamber derived from the ignition heats the layer of fireclay adjacent to the chamber. Ignition then also takes place in this layer. If the flow velocity of the gas is smaller than the ignition velocity, the combustion zone moves nearer and nearer to the side where the gas and air is introduced until it finally reaches it. The gas and air velocity

must then be increased either by an increase in the pressure or by means of larger jets for the gas and air to flow through, or by a greater porosity or even small borings in the walls. This principle has a wide application to furnaces for industrial uses. The place of uniformly porous walls is often

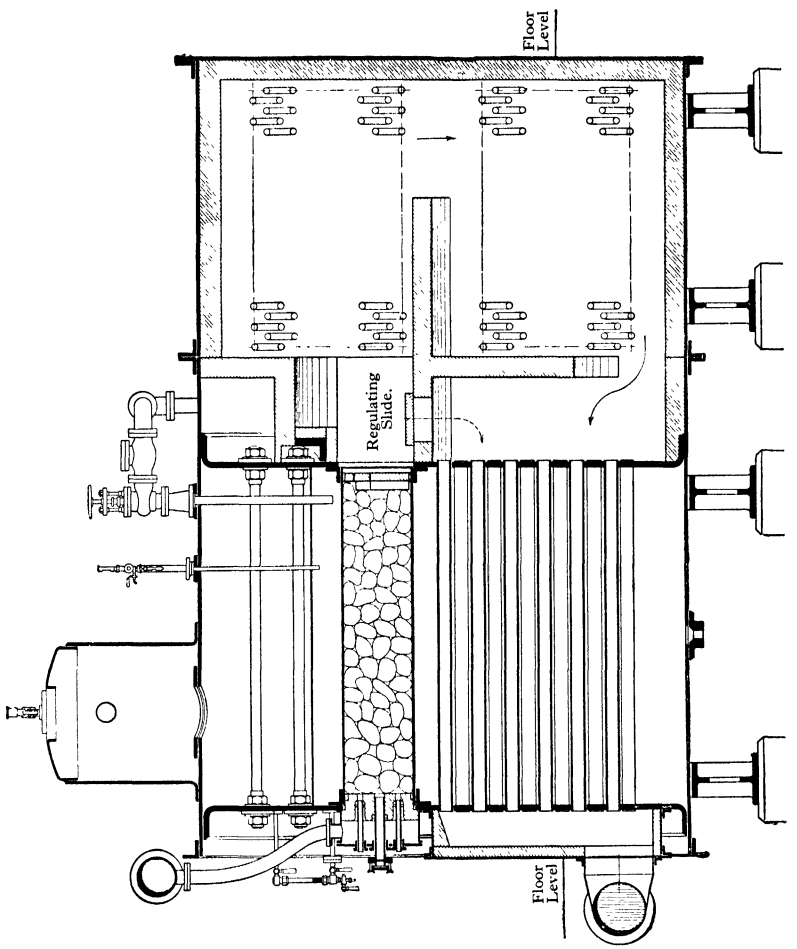


Fig. 64.—Surface Combustion Boiler (*Bone-Schnabel*)

taken by a kind of grid burner, i.e. a more or less porous wall with round or rectangular openings.

In fig. 63 is shown an arrangement for a partial combustion without flames. A portion of the gas is first burnt in the central combustion chamber, and the combustion is completed without flames during the passage through the fireclay. A design without the initial open chamber is shown in fig. 64. The gas-and-air mixture is drawn out by means of an exhauster placed

beyond the pipes. The temperature distribution along one of the heating tubes packed with fireclay nuts for flameless combustion is given in fig. 65.

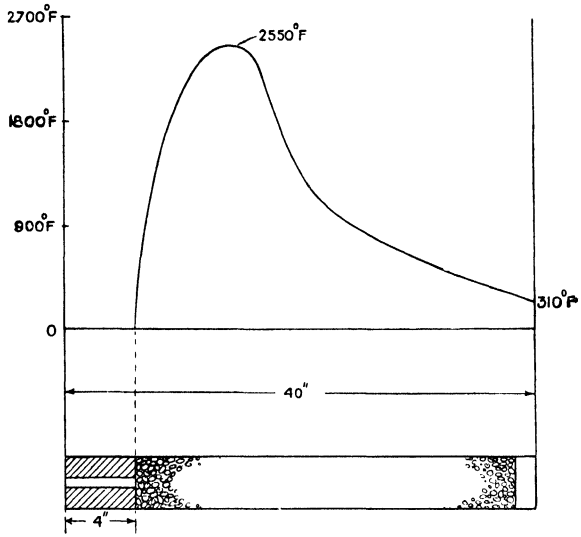


Fig. 65.—Temperature Distribution along a Surface Combustion Tube

A recent invention for the very efficient heating and vaporizing of water is the Brünler submerged furnace, which is shown in principle in fig. 66. Gaseous or liquid fuel, together with the air necessary for combustion, is forced from a tube whose exit is under water and is there ignited. Steam is formed and rises to the surface together with the gaseous products of combustion, and the mixture can be used in an engine or for other purposes. Practically the whole of the calorific value of the fuel can thus be transferred to the working mixture. If, however, the mixture is used to work an engine it is necessary to have high pressures to obtain high engine efficiency, and considerable work has to be expended to introduce the fuel and combustion air into the boiler against these pressures.

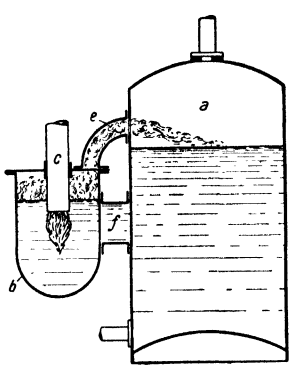


Fig. 66.—Under-water Firing (*Brünler*)

a is the main boiler; combustion takes place in the auxiliary boiler *b*; *c* is the inlet pipe of the gas mixture; *e* and *f* are connecting pipes between the two boilers.

13. HEAT FROM WASTE GASES

In many cases the exhaust gases from furnaces have calorific value as well as sensible heat, and the gases can be utilized to drive gas-engines. In other cases the heat energy is that of sensible heat only. Even where the exhaust gases have calorific value, it is very often advisable to utilize them in producing steam for a steam-engine rather than to use them to drive an internal-combustion engine. With a full load the thermal efficiency of a gas-engine may reach 30 to 35 per cent while that of a steam-engine may be only 12 to 16 per cent, and, furthermore, the steam-engine and boiler require more attention owing to the importance of efficient combustion in the boiler furnace. Under reduced load, however, the efficiency of the gas-

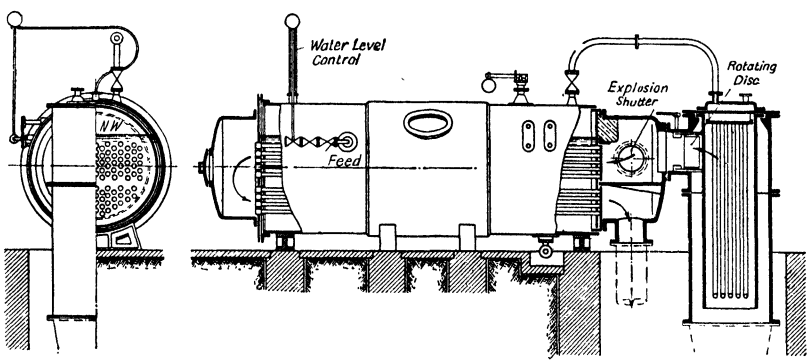
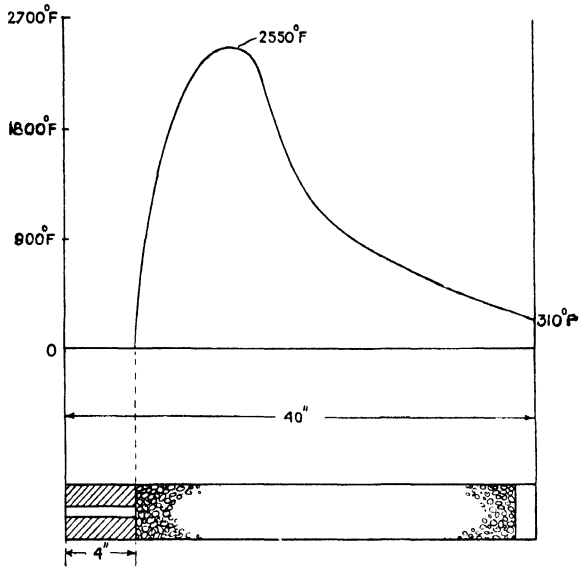


Fig. 67.—Exhaust-heat Boiler with Double Draught for Gas Engine with a Superheater inserted in Exhaust Pipe

engine is greatly reduced while that of the steam-engine is affected only to a small extent. In many industries steam is required for heating or other purposes, and a steam-engine plant suffices to provide both power and steam, either directly from the boiler or from the exhaust of the engine. Where the gases are to be used in an internal-combustion engine, it is necessary that they should be freed from the fine dust and ash which they contain. This may be effected by either wet or dry process and the solid deposit manufactured into briquettes. The process necessarily adds to the cost of using the waste gases in this manner.

In a blast furnace 160,000 c. ft. of gas are produced per ton of iron or coke with a calorific value of 85 B.Th.U. per cubic foot and a flue temperature of 400° F. With coking ovens 13,500 to 15,000 c. ft. per ton of coke or 9000 to 11,000 c. ft. per ton of coal are produced, one quarter being available for external purposes. Their calorific value is 400 to 500 B.Th.U. per cubic foot and temperature 900° F. The exhaust gases of annealing furnaces have a heat content of about 300,000 B.Th.U. from 100 lb. of coke. The waste gases of a Siemens-Martin furnace at temperatures of 1100° F. to 1300° F. contain 30 per cent of the calorific value of the fuel, and more

beyond the pipes. The temperature distribution along one of the heating tubes packed with fireclay nuts for flameless combustion is given in fig. 65.



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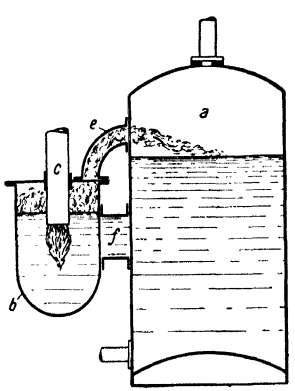

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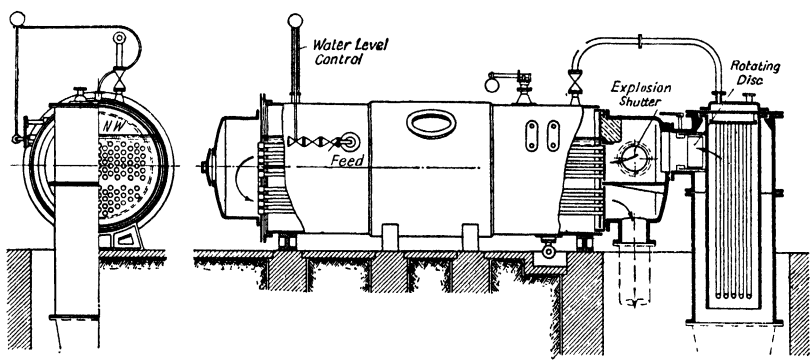

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than half of this can be utilized for the production of steam in a steam boiler.

The following figures of a test on a Siemens-Martin furnace show how important it is to utilize the heat value of the waste gases:

Yield of ingots, 75 tons.
Coal consumption, 23·7 tons.
Calorific value of coal, 10,700 B.Th.U. per pound.
Waste gases, 378 tons.
Temperature of waste gases, 1250° F.
Sensible heat of waste gases, 290 B.Th.U. per pound.
Proportion of heat in waste gases

$$= \frac{378 \times 290}{23{\cdot}7 \times 10700} = 0{\cdot}43 \text{ or } 43 \text{ per cent.}$$

Proportion of waste heat utilized in boiler, 52 per cent.
Proportion of total heat utilized in boiler

$$= 0{\cdot}52 \times 0{\cdot}43 = 0{\cdot}22 \text{ or } 22 \text{ per cent.}$$

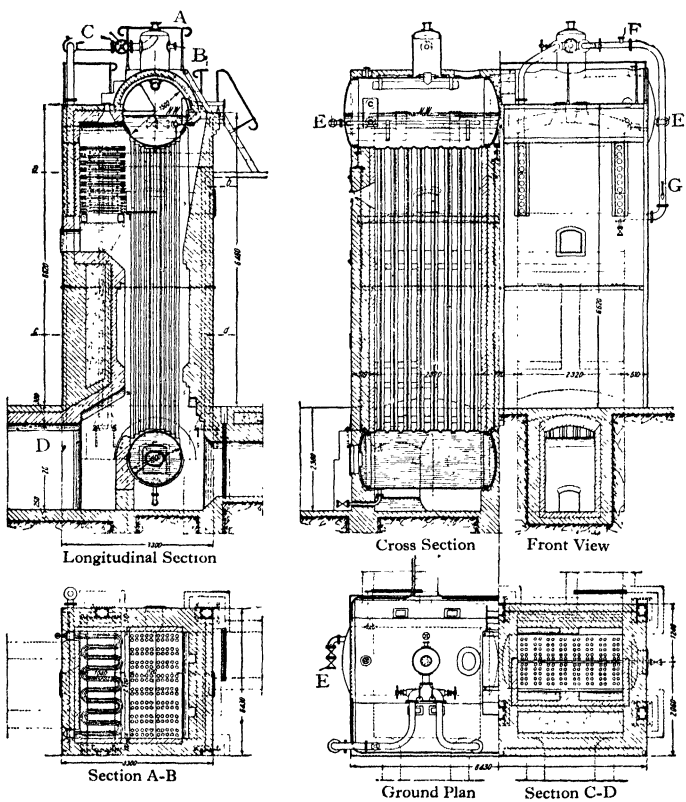


Fig. 68.—Exhaust Gas Boiler

A, Duplex safety valve. B, Pump steam connexion. C, Triple stop valve. D, Exhaust gas inlet. E, Feed connexion. F, Superheater safety valve. G, Thermometer.

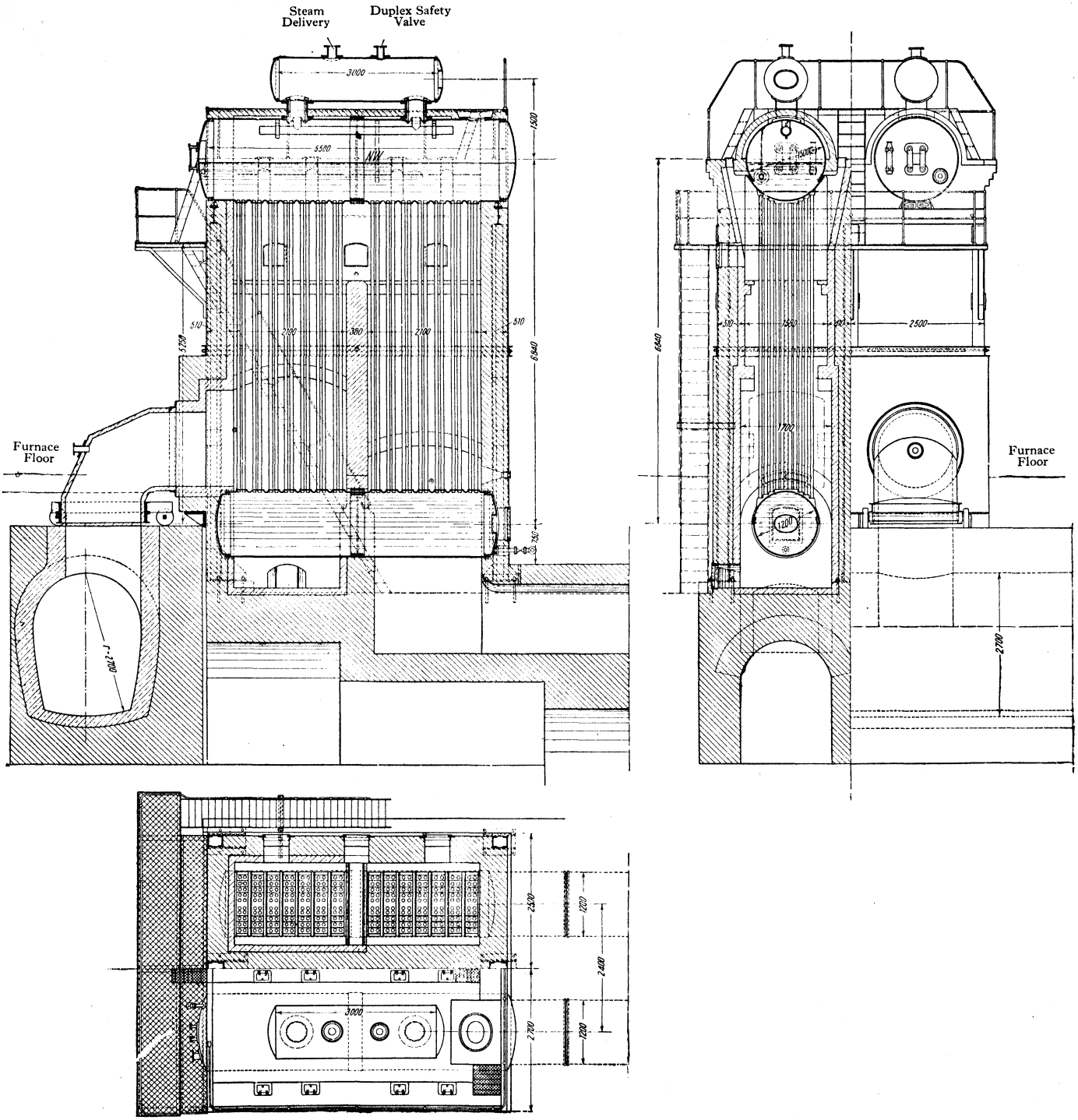


Fig. 69.—Garbe Exhaust Gas Boiler

[*Facing p.* 114

The increase of resistance introduced by the passage of the gases through the boiler must be allowed for by a higher chimney or an artificial draught. The waste gases from puddling or heating furnaces are likewise used in boilers placed close to or immediately over the furnace, the temperature being even higher than those of the gases of the Siemens-Martin furnace. Boilers over puddling furnaces can produce 2 to 3 lb. of steam per pound of coal in the furnace, welding furnaces 3·5 to 5·5 lb. The boilers with which these values were obtained had in the first case heating surfaces of 400 to 800 sq. ft., and in the second case 1200 sq. ft.

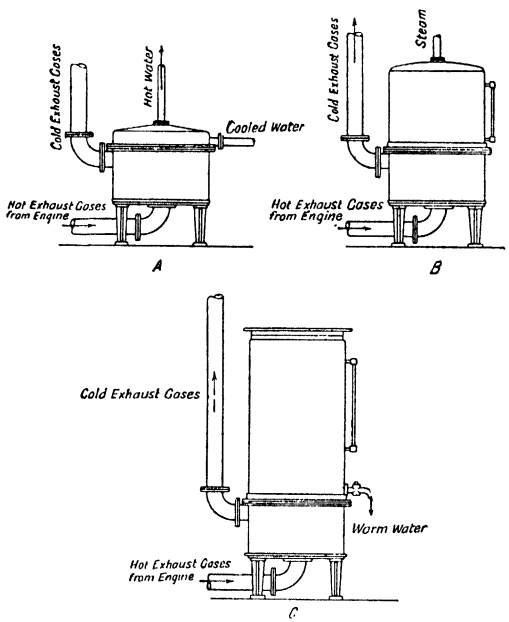


Fig. 70.—Utilization of Exhaust Gas for Heating Water (*Augsburg-Nürnberg*)

An example of a simple exhaust-heat steam boiler of the fire-tube type is shown in fig. 67. The hot gases first pass a superheater in which the steam is circulating, then through the tubes in the upper portion of the boiler, and return through the tubes of the lower portion. Preheaters for the water may also be included. Boilers of this character using the exhaust gas from gas-engines at temperatures of about 1000° F. will produce from $1\frac{3}{4}$ to 2 lb. of steam per hour at 12 to 15 atmospheres pressure for each horse-power of the engine.

Fig. 68 shows a boiler for utilizing the waste gases of a zinc smelting furnace. On entering the boiler, the gases have a temperature of 1600° F., and on leaving 850° F. The boiler has a heating surface of 2500 sq. ft. and generates steam at 120 lb. per square inch. It requires 270,000 c. ft. of waste gas per hour (measured at 32° F.), the composition being 18 per cent CO_2,

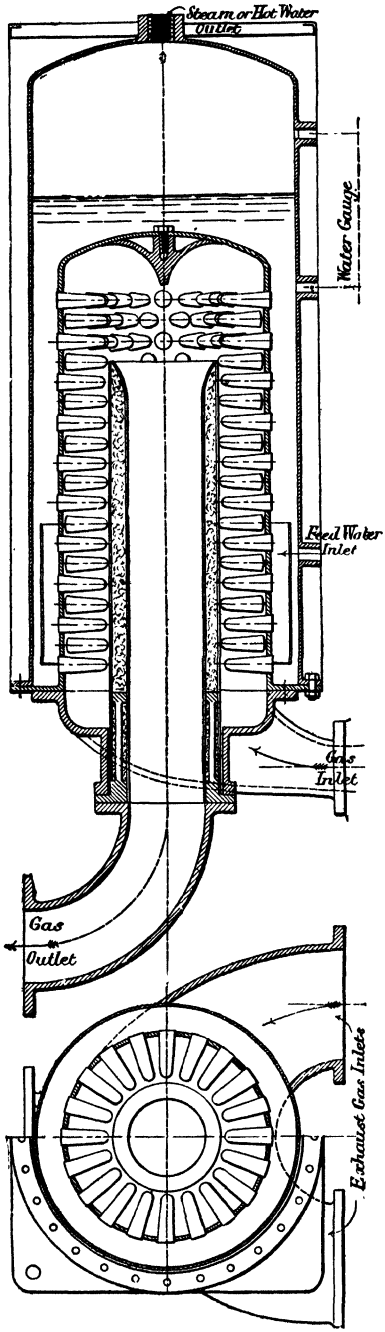


Fig. 71.—Waste Heat Boiler for Oil Engines

6 per cent O_2, and 76 per cent N_2. The output is 1·7 to 2·2 lb. of steam per square foot of heating surface. The boiler has also an economizer with a surface of 1200 sq. ft.

In fig. 69 is shown a boiler for the waste gases of a Martin furnace, with a heating surface of 3000 sq. ft., an inlet temperature of the waste gas of 1600° F., and a production of steam of 3·5 lb. per square foot.

Simple designs of boilers for the production of hot water are shown in fig. 70. These are suitable for cases in which the amount of heat to be dealt with is small, as with the exhaust gases of small internal-combustion engine plants. A boiler of higher efficiency is the Clarkson boiler (fig. 71). The gases pass upward through the inner cylinder, which is provided with a large surface by the insertion of short closed tubes. A circulation of water between the inner and outer cylinder was found to extract 60 per cent of the sensible heat of the exhaust gases. In tests carried out on a heavy-oil engine of 84 b.h.p. this heat obtained from the exhaust gases was equivalent to 66 h.p. For cast-iron heating surfaces having exhaust gas on one side and water on the other, the average heat transfer is between 400 and 600 B.Th.U. per square foot per hour, 1 h.p. hour being transferred for every 2 sq. ft. The coefficient of heat transfer depends upon the velocities and temperature difference. The variation due to the latter cause is illustrated by the following test results.

Mean Temperature difference between Gas and Water.		Heat Transfer per square foot per hour per 1° F. difference.
100° F.	....	1·2 B.Th.U.
150 ,,	....	1·7 ,,
200 ,,	....	2·05 ,,
300 ,,	....	2·6 ,,
400 ,,	....	2·9 ,,

When utilizing the exhaust heat of gas-, oil-, or petrol-engines either in boilers or for general heating purposes, the resistance experienced by the gases in boilers or pipes must be reduced to a minimum, as the imposition of back pressure on the engine results in serious loss of power. The back pressure should not exceed 3 lb. per square inch.

14. FURNACES WITH GAS PRODUCERS

In fig. 72 is shown the general arrangement of a reverberatory furnace using producer gas from an attached generator. The gas enters at the be-

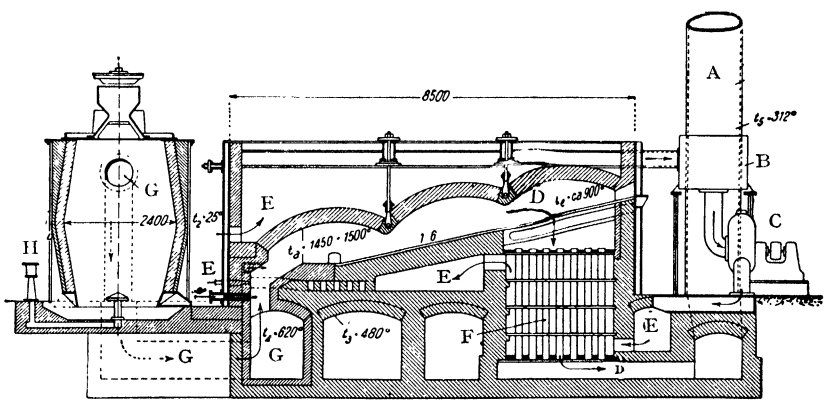


Fig. 72.—Gas Furnace

A, Chimney. B, Pre-heater. C, Ventilator. D, Exhaust gases. E, Air. F, Recuperator. G, Gas. H, Steam-jet blower.

ginning of the hearth possessing both sensible heat and calorific value, and there meets the air required for combustion, which has been pre-heated by passing through the recuperator. After combustion the waste gases heat the recuperator in their passage to the chimney. The following figures refer to a test on such a plant:

Calorific value of coal in generator, 12,800 B.Th.U. per pound.
Ash content of coal in generator, 8 per cent.
Depth of charge in generator, 2 to 3 ft.
Coal consumption in 10 hr., 5·1 tons.
Furnace charge, 92·5 tons.
Furnace charge temperature, 2010° F.

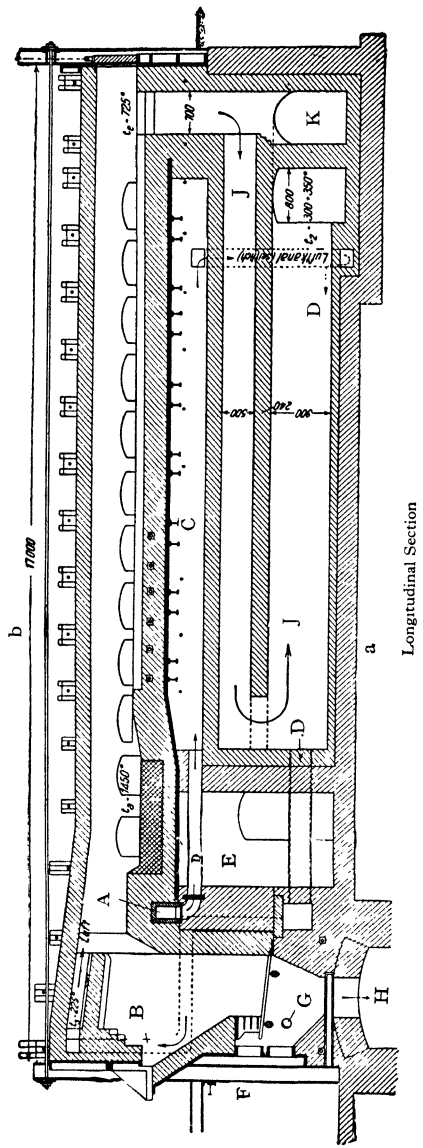


Fig. 73.—Stoss Oven with Recuperator, semi Gas-fired

Gas entering furnace, 1185° F.
Gas after combustion (beginning of hearth), 2650° F.
Gas, end of hearth, 1650° F.
Gas, end of recuperator, 600° F.
Air, external temperature, 65° F.
Air leaving recuperator, 900° F.

Assuming that in an ideal case heat could be extracted from the gases until they were reduced to 32° F.:

$$\text{Efficiency of the furnace} = \frac{2650 - 1650}{2650 - 32} = 0{\cdot}38.$$

$$\text{Efficiency of recuperator} = \frac{1650 - 600}{1650 - 32} = 0{\cdot}65.$$

$$\text{Efficiency of hearth and recuperator} = \frac{2650 - 600}{2650 - 32} = 0{\cdot}78.$$

$$\text{Efficiency of whole plant} = \frac{92{\cdot}5 \times 0{\cdot}16 \times (2010 - 32)}{5{\cdot}1 \times 12{,}800} = 0{\cdot}45.$$

In fig. 73, the arrangement of a Stoss plant with similar action is given. The gas flow in the upper chamber is 2100 c. ft. per minute at a pressure of 0·5 in. of mercury. The efficiencies are very similar to those of the plant in fig. 72. The two examples have a furnace standing apart from the generator. In the forge furnace shown in fig. 74, the generator is constructed as an integral part of the furnace, increasing the efficiency by reducing the heat losses. The reverberatory furnaces, which are specially constructed for forge workshops, have a coal consumption of 10 to 15 per cent of the heated iron as compared to the open furnaces or coke furnaces used in forges, which have a fuel consumption of 75 to 100 per cent of the heated iron.

Fig. 75 shows a re-heating furnace plant heated with blast-furnace gas. This furnace produces 800 tons in 10 hr. with a warm charge, and about half the amount with a cold charge.

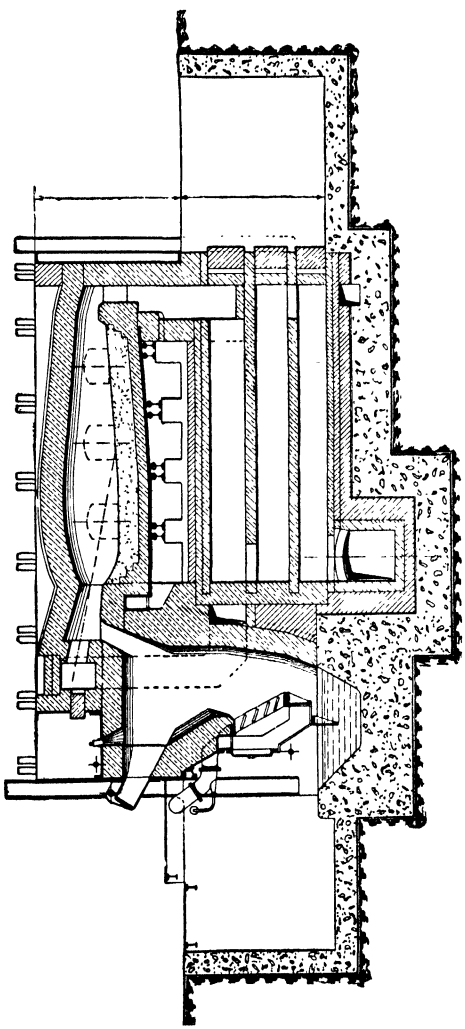


Fig. 74.—Forge and Welding Furnace with Recuperator (*W. Ruppmann*)

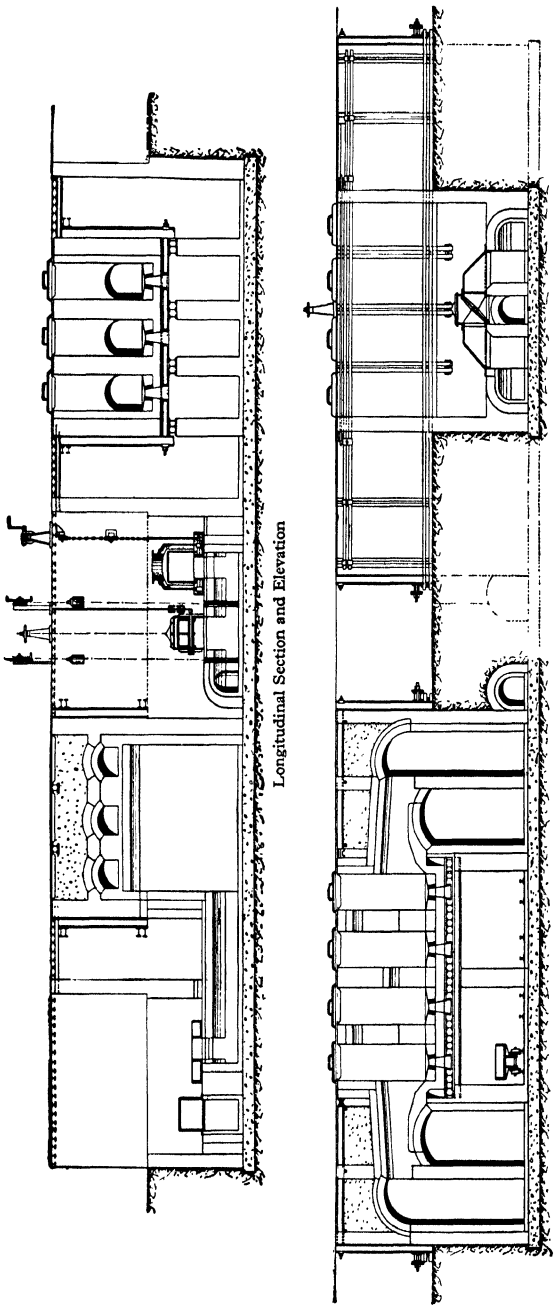


Fig. 75.—Soaking Furnaces Heated with Blast Furnace Gas (*Siemens*)

One ton of heated ingots requires about 3000 c. ft., while cold ingots require 6000 c. ft. of gas and air pre-heated to 2200° F.

The use of the calorific value and sensible heat of a special producer-gas plant in the porcelain industry is illustrated in fig. 76. In order to utilize the hot gases more fully, two or more burning chambers are arranged one over the other. The hot gases, on leaving the gas producer, enter a dust chamber in which the fine dust is deposited, while the clean gas escapes into the main gas pipe. With bituminous fuels there is, in addition, an arrangement for separating the tar and water between the dust chamber and the porcelain furnace. Water-gas and fuel-gas, free from tar, flow through

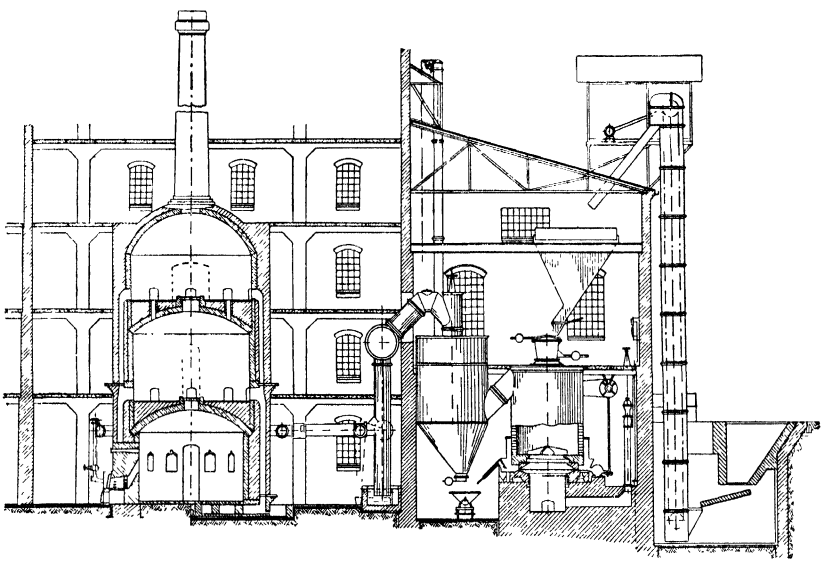


Fig. 76.—Gas Heated, Three-storied Porcelain Circular Furnaces with Gas Producer

the gas pipe encircling the furnace to the gas valves, which are placed in front of the twelve burners of the lower combustion chamber. The passage of the gas into the furnace takes place half-way up the chamber, the gas-and-air mixture burning upwards with a rising, inclined flame. A uniform distribution of the flame over the whole section of the furnace is secured by means of the symmetrical arrangement of the twelve gas-delivery pipes and of the discharge shafts inserted in the floor of the furnace for the combustion gases. The hot exhaust gases, which escape through the floor of the lowest chamber, flow through pipes in the walls to the second story, where their sensible heat is utilized for a preliminary heating of the porcelain. The gases can also be led straight to the third story by a valve adjustment. Failing this, they pass into the third story through openings in the roof of the second story, leaving the furnace through the flue.

15. PRE-HEATING

The pre-heating of gas or air, which has been referred to several times during the preceding descriptions, is of great importance. It can be carried out either in recuperators or regenerators. The object of the pre-heating is to supply the gas with sensible heat before the combustion, thus raising the combustion temperature. This sensible heat is taken from the waste gases, thus raising at the same time the thermal efficiency of the plant.

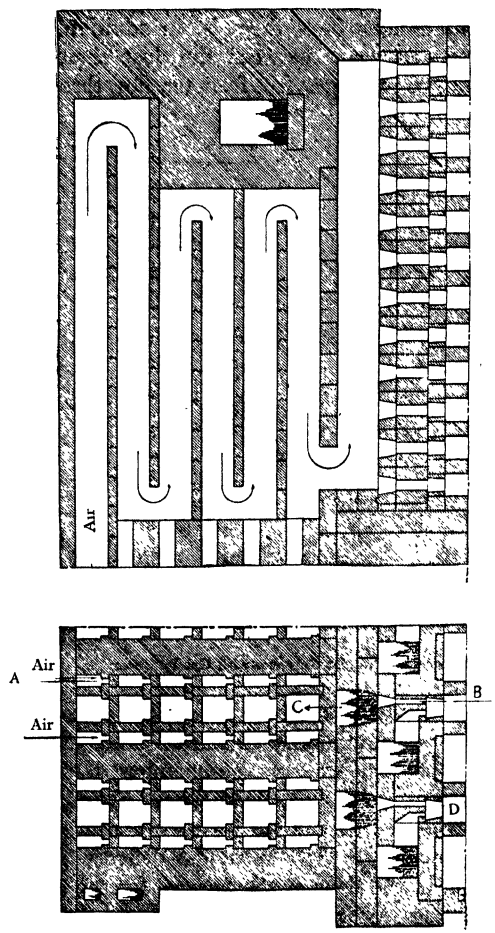


Fig. 77.—Recuperator for Illuminating Gas Furnace
C, Exhaust heat. D, Hot blast furnace chamber.

The principle of the recuperator is that the waste gases are conducted through a system of pipes, and that the heat penetrating through the walls of the pipe system is then transmitted to the air or gas, which is usually flowing in the opposite direction. The principle of the regenerator is that the waste gases are conducted through a grating made of fireclay or brick and are then made to give up their heat. After a certain time has elapsed the flow of exhaust gas is shut off and conducted through a second chamber, while the air or gas moving in the opposite direction abstracts the heat contained in the bricks or fireclay in the first chamber.

Thus the process in a recuperator is continuous, while that in a regenerator is intermittent, and requires at least two chambers for continuous production of hot gas, with the necessary valves for directing the alternate exhaust and fresh gas flows.

The recuperator is placed either underneath or at the side of the furnace. The technical arrangement can be seen from fig. 77.

As already stated, the heat transfer is

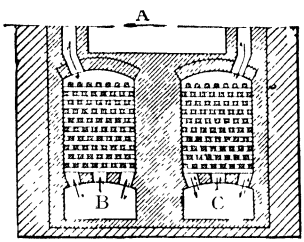
$$Q = Kf(T_2 - T_1).$$

T_2 is the temperature of the waste gas and T_1 that of the air at a point opposite to T_2.

The coefficient of heat transfer can be obtained from the coefficients for the heat flow from exhaust gas to wall, through the material of the wall, and from wall to air. Thus:

$$k = \frac{1}{\frac{1}{\beta_1} + \frac{1}{\beta_2} + \frac{d}{\lambda}},$$

where d is the thickness of the wall and λ its heat conductivity. The



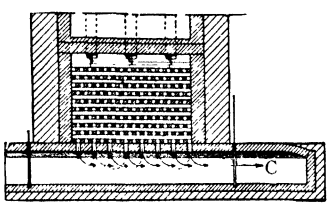
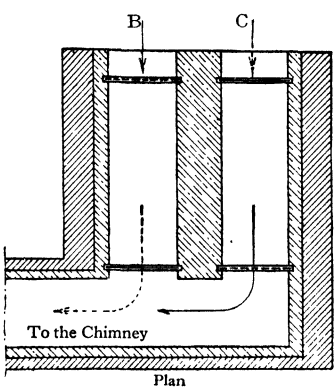


Fig. 78.—Regenerator

A, Hearth. B, Air. C, Exhaust heat



coefficients β_1 and β_2 include both conduction and radiation factors. When the waste gases enter the recuperator, i.e. at temperatures of about 1500° F. to 2000° F., the radiation from them is greater than the conduction, and when the gases leave the recuperator at 650° F. to 900° F., the reverse is true. With a waste gas velocity of 13 ft. per second and a gas velocity $6\frac{1}{2}$ ft. per second, waste gas at 1800° F. falls to 800° F. and air at 70° F. rises to 1300° F. Litinsky found temperature differences through the fireclay walls, which were $2\frac{1}{2}$ in. thick, of 260° F. With equal velocities of waste gas and air, the corresponding temperature gradient was 190° F. He also found that the coefficients from gas to wall or from wall to air followed the law $\beta = a + b\sqrt{v}$.

In fig. 78 is shown the arrangement of a generator which may be used either for air or for gas. In the recuperator, the flow of heat between gas and air is governed by the heat transfer across the two surfaces, and the penetration through the fireclay wall, while in the regenerator the flow of

heat is determined by the surface transfer from waste gas to the firebrick and from firebrick to the air, and the air or gas can be heated to a higher temperature. In the regenerator, the fireclay serves as a heat accumulator. The heat has to penetrate into the brick, and conversely the heat must flow from the centre of the brick outwards when the air is heated. For this reason, a large number of thin bricks is more efficient than a small number of thick ones. The gas and air velocities in the brick lattice work of regenerators are open to great variations. On the average, they vary between 20 and 50 ft. per second, while the free surface per cubic foot varies between 86 and 194 sq. ft. The ratio

$$\frac{\text{capacity of lattice work}}{\text{capacity of furnace}}$$

is often given as a basis for design, but as for different cases and types of furnace it varies from 0·2 to 1·0, the ratio thus appears to provide no criterion of efficiency.

Recuperators have the advantage of being easy to manipulate. Between the flue gas at a low pressure and the air at an excess pressure, there is a pressure difference which causes air to enter the flue gases through cracks and parts that are not air-tight. For this reason it is not advisable to heat gas in a fireclay recuperator. If the gas needs to be only slightly warmed, a system of wrought-iron pipes is used in place of the fireclay pipes.

With furnaces which are continuous in action and whose efficiency depends upon a gradation of temperature from one part to another, it is necessary that the gas and air flow be continuous. As the flow of exhaust gases and of new gas or air in a regenerator are in opposite directions, the connexions of a regenerator to such a furnace are somewhat complex, and not so simple as that to a recuperator, and hence a recuperator is more commonly used in these cases. It may also be pointed out that the inlet and outlet pipes of a regenerator must be suitable for flow in either direction, and for either fresh gas or combustion products, whereas in a recuperator each section serves only one purpose.

16. HEAT TRANSFER IN FURNACES AND BOILERS

The heat transfer takes place by conduction or radiation or both, according to the temperature and flow conditions. Heat conduction takes place across the boundary surface between two media of different temperatures. The greater the temperature gradient, the greater the amount of heat transferred. The exact relationship between the heat conduction from or to a solid body in contact with a flowing liquid or gas at a different temperature has not yet been exactly elucidated, but it increases with the velocity of the liquid or gas. Turbulence of flow, apart from increase of mean velocity, is effective in increasing the heat transfer. A particle of hot fluid having

come into contact with the wall and given up its heat is removed by the turbulence and another takes its place. With a steady uniform motion such a particle would remain in contact with the wall and prevent further heat transfer except by conduction through itself. For the flow of fluid, either liquid or gaseous, in a pipe the flow becomes turbulent when the velocity exceeds the critical velocity given by Reynolds's equation in any units:

$$v_c = 2 \times 10^3 \mu/d\rho,$$

where μ = viscosity, ρ = density, d = diameter of pipe. Thus for water at 100° F., $v_c = 0{\cdot}014/d$ ft. per second where d is in feet, and at 200° F. about half this velocity results in turbulence. For air at 100° F., $v_c = 0{\cdot}3/d$, and it increases with temperature to $1/d$ at 600° F.

The quantity of heat crossing a boundary surface owing to heat conduction from a flowing gas has been the subject of many tests by Nusselt, but owing to the difficulty of carrying out experiments at different temperatures and with apparatus of varying size and shape, and yet under the same conditions of turbulence, it can only be deduced that the heat transfer increases with velocity and also with density, in the latter case proportionally to a power of density about $\rho^{\frac{3}{4}}$ or $\rho^{\frac{4}{5}}$. Thus

$$Q = \beta d T,$$

where dT is the temperature difference and β is a function of the velocity, density, and also the viscosity of the gas. If heat is being given up by one fluid to another, both flowing parallel to the boundary surface, the temperature difference will vary from point to point as one gains and the other loses heat.

Let d = width of boundary surface.
l = length of boundary surface.
a = $w_1 s_1$, where w_1 is flow per second of fluid giving up heat and s_1 its specific heat.
b = $w_2 s_2$, where w_2 is flow per second of fluid receiving heat and s_2 its specific heat.
T_2 = increase in temperature of w_2.
T_1 = initial temperature difference.
β = heat conduction coefficient assumed constant over temperature changes concerned.

If the two fluids are flowing in the same direction,

$$l = \frac{ab}{\beta d(a+b)} \log_e \frac{aT_1}{aT_1 - (a+b)T_2}$$
$$= \frac{b}{\beta d}\left\{\frac{T_2}{T_1} + \tfrac{1}{2}\frac{a+b}{a}\frac{T_2^2}{T_1^2} + \tfrac{1}{3}\frac{(a+b)^2}{a^2}\frac{T_2^3}{T_1^3} + \ldots\right\}.$$

If l is very great, $T_2 = aT_1/(a+b)$.

If the two fluids are flowing in opposite directions,

$$l = \frac{ab}{\beta d(b-a)} \log_e \frac{a\mathrm{T}_1 - a\mathrm{T}_2}{a\mathrm{T}_1 - b\mathrm{T}_2}$$
$$= \frac{b}{\beta d}\left\{\frac{\mathrm{T}_2}{\mathrm{T}_1} + \tfrac{1}{2}\frac{a+b}{a}\frac{\mathrm{T}_2^2}{\mathrm{T}_1^2} + \tfrac{1}{3}\frac{a^2+ab+b^2}{a^2}\frac{\mathrm{T}_2^3}{\mathrm{T}_1^3} + \ldots\right\}.$$

If l is very great, $\mathrm{T}_2 = \mathrm{T}_1$.

Furthermore, for any finite length the increase of temperature T_2 and the heat transfer is greater with this contra-flow than with parallel flow.

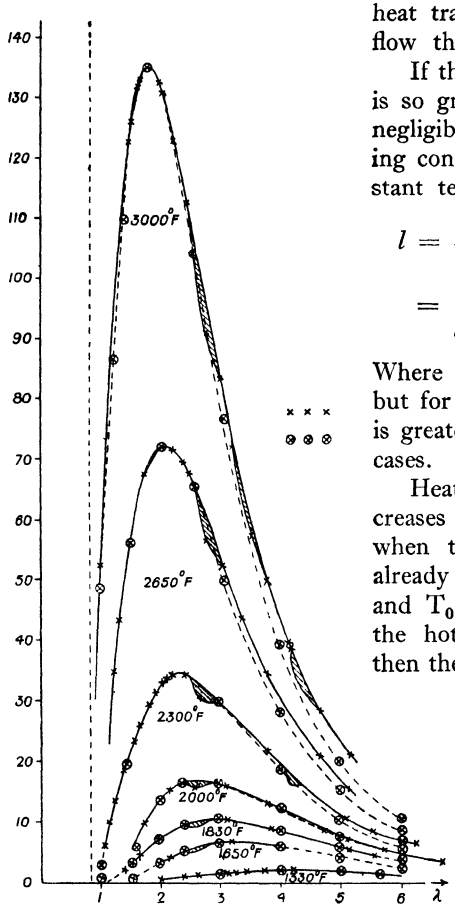


Fig. 79.—Energy Distribution in the Spectrum

If the flow of the fluid giving up heat is so great that its fall in temperature is negligible, or if it is a vapour that is being condensed and hence remains at constant temperature,

$$l = \frac{b}{\beta d}\log_e \frac{\mathrm{T}_1}{\mathrm{T}_1 - \mathrm{T}_2}$$
$$= \frac{b}{\beta d}\left\{\frac{\mathrm{T}_2}{\mathrm{T}_1} + \tfrac{1}{2}\frac{\mathrm{T}_2^2}{\mathrm{T}_1^2} + \tfrac{1}{3}\frac{\mathrm{T}_2^3}{\mathrm{T}_1^3} + \ldots\right\}.$$

Where l is very great, again $\mathrm{T}_2 = \mathrm{T}_1$, but for any finite length the heat transfer is greater than in either of the other two cases.

Heat transmitted by radiation increases very rapidly with temperature when the temperature is high. It has already been shown in Chapter I that if T and T_0 are the absolute temperatures of the hot body and of its surroundings, then the heat transfer is equal to

$$c(\mathrm{T}^4 - \mathrm{T}_0^4).$$

This formula refers to the total radiation, which is made up of radiation of all wave-lengths. The connexion between radiation, temperature, and wave-length is given by Planck's formula,

$$\mathrm{E} = c_1\lambda^{-5}(e^{c_2/\lambda\mathrm{T}} - 1)^{-1},$$

where c_1 and c_2 are constants and λ the wave-length. The experimental results for black body radiation agreeing with this formula are given in

fig. 79. It will be seen that when the temperature is increased, the radiation due to shorter wave-length increases more rapidly than that of longer wave-length, and there is a displacement of the maximum towards the shorter wave-length. Thus at high temperatures it is advantageous to have short wave-length predominating, i.e. to have luminous flames.

The radiant heat in passing through a thickness of gas x is absorbed according to Kirchhoff's law:

$$A = E(1 - e^{-c_3 x}).$$

Under steady conditions, however, this heat must be again radiated out. The absorption constant c_3 depends upon the nature of the gas. The experiments of Schack gave the following results for the distribution of radiation in certain furnaces:

Martin furnace: gas temperature 3400° F., bath temperature 2600° F., 15 per cent CO_2, 6 per cent steam.

Radiation of arch	..	11,000 B.Th.U. per square foot per hour.
Radiation of CO_2	..	6600 ,, ,,
Radiation of steam	..	5500 ,, ,,
Convection of gas	..	1300 ,, ,,

Stoss furnace at entrance: gas temperature 3100° F., ingot temperature 2300° F.

Radiation of arch	..	5500 B.Th.U. per square foot per hour.
Radiation of CO_2	..	5500 ,, ,,
Radiation of steam	..	4400 ,, ,,
Convection	.. ..	— ,, ,,

Stoss furnace at centre: gas temperature 2000° F., ingot temperature 900° F.

Radiation of CO_2	..	3700 B.Th.U. per square foot per hour.
Radiation of steam	..	2600 ,, ,,

Stoss furnace, average:

Radiation of CO_2 and steam,		7300 B.Th.U. per square foot per hour.
Convection	.. ..	3000 ,, ,,

In a fire-tube steam boiler with a tube wall temperature between 400° F. and 480° F., a uniform gas formation and an initial gas temperature of 2200° F., the heat transmission from within the tube due to the various factors are:

Radiation of CO_2	..	5700 B.Th.U. per square foot per hour.
Radiation of steam	..	5200 ,, ,,
Convection	.. ..	8800 ,, ,,

Convection requires high gas velocities, while radiation requires low gas velocities and dense layers of radiating gas. For each design, therefore, there is a velocity which will produce maximum heat transmission.

CHAPTER IV

Distillation and Gasification of Fuels

1. DISTILLATION

Coal is a very complex substance, and the efficient combustion of all its constituents in a single plant is a difficult matter. Except in special plants, certain constituents from which valuable chemicals can be obtained are entirely wasted. Hence, if coal can be split up into substances each of which is less complex, the efficient combustion of each in a plant suited to its nature is simplified. The processes in which combustion is required are so many that for each constituent there are several processes for which it is more suitable than the crude coal. Attempts at extracting constituents of coal by chemical solvents have so far attained little success, and all successful processes depend on the application of heat with or without air and steam. When coal is distilled by heating in a restricted atmosphere, gas, water containing ammoniacal compounds, and tar are given off, while some form of coke forms the residue. The character of the products depends greatly upon the temperature of distillation. At high temperatures a considerable quantity of gas of medium calorific value and heavy tars is given off, while the coke residue contains little volatile matter. At low temperatures much smaller quantities of gas of high calorific value and lighter tar is given off, while the coke retains considerable volatile matter and is hence more easily ignited. There is no definite dividing line between low- and high-temperature treatment, but the term low-temperature carbonization is generally used in reference to distillation about or below 1000° F., while high-temperature distillation is usually that of a gas manufacturing plant, viz. about 2000° F.

2. GAS PRODUCTS OF DISTILLATION

The quantity of gas obtained depends upon the character of the coal used, the following being average figures in cubic feet from 1 lb. of fuel.

	Low Temperature.	High Temperature.
From 1 lb. of peat	0·6 c. ft.	2·5 c. ft.
„ 1 lb. of brown coal ..	0·35 „	1·5 „
„ 1 lb. of mineral coal ..	0·8 to 2·5 c. ft.	5 to 6 c. ft.

The chemical composition of the gas shows that the hydrogen contents in low-temperature gas are very small. It is only when the temperature rises above 1300° F. or 1400° F. that a sudden development of hydrogen arises, while the development of paraffin hydrocarbons ceases. The composition also depends upon the character of the coal and the mode of treatment. In the following tables this variation is apparent, but it is not sufficient to obscure the above general difference between the high- and low-temperature products. The last column refers to gas obtained during the production of hard metallurgical coke by heating coal packed tightly in retorts to a high temperature.

Low-temperature Gas.	From Mineral Coal, 800° F.		From Brown Coal, 850° F.
	Per Cent.	Per Cent.	Per Cent.
CO_2	4·7 (*a*)	3·8 (*b*)	42·6
CO	3·6	3·8	8·1
C_nH_{2n}	10·1	5·7	4·0
CH_4	45·5	54·2	8·0
C_2H_6	28·4	15·2	18·0
H_2	7·7	17·3	19·3

High-temperature Gas.	Coal Gas (Mineral Coal).	Coking Gas.
	Per Cent.	Per Cent.
CO_2	2 to 2·5	—
CO	12 ,, 9	8·1
C_nH_{2n}	3 ,, 1	—
CH_4	23 ,, 34	34·0
C_2H_6	1 ,, 4	—
C_nH_m	—	3·6
H_2	55 ,, 47	53·4
N_2	4 ,, 2·5	0·9

The calorific value of low-temperature gas is considerably higher than that of coal gas or coking gas. It varies from 800 to 1100 B.Th.U. per cubic foot, while coal yields on an average 550 B.Th.U. per cubic foot.

3. TAR PRODUCTS

Some of the differences between low-temperature tar and coking tar are very obvious. At room temperature the former is liquid and contains little or no paraffin. The specific gravity is 0·95 to 1·06 at 65° F. It smells of hydrogen sulphide or ammonium sulphide, but never of naphthalene. A thin layer of the tar has a reddish gold colour. Ordinary coking tar at normal temperatures is a viscous black mass, the specific gravity lying between 1·12 and 1·18, while that of gas-plant tar lies between 1·18 and 1·25. The heavy tar smells of phenol and naphthalene.

According to Fischer, low-temperature tar from various coals has the following composition:

	From Bituminous Coal. Per Cent.	From Gas Coal. Per Cent.	From Brown Coal (dried). Per Cent.
Viscous oils ..	15·2	10·0	16·5
Paraffin ..	0·4	1·0	14·5
Non-viscous oils	33·5	15·0	25·9
Phenol ..	14·0	50·0	29·0
Resin ..	4·2	1·0	2·6
Pitch ..	19·2	6·0	9·5
Water, &c. ..	13·5	17·0	2·0
From 1 lb. of coal ..	0·03 lb.	0·10 lb.	0·08 lb.

The composition of high-temperature (gas-plant) tar varies according to the type of furnace; it is black and viscous with a horizontal furnace, while with inclined or vertical furnaces it becomes brown and aqueous. Analyses in the *Journal für Gasbeleuchtung*, 1906, give the composition as follows:

	Tar from Horizontal Furnace.	Tar from Vertical Furnace.
Specific gravity ..	1·24	1·16
Water ..	2·80 per cent	3·00 per cent
Light oil ..	6·09 ,,	7·65 ,,
Medium oil ..	5·26 ,,	9·78 ,,
Heavy oil ..	9·43 ,,	16·55 ,,
Anthracene oil ..	8·90 ,,	17·42 ,,
Pitch ..	67·52 ,,	45·60 ,,

The yield is 5 to 5½ per cent of the coal under treatment.

Crude tar from coke plant consists of

Oils and solid deposits ..	about 41 per cent.
Pitch	,, 55 ,,
Water and distillation losses ..	,, 4 ,,

A more detailed analysis of a tar from a coking plant gave the following result:

0·1 to 0·5 per cent	light benzol.	0·25 per cent	pyridine bases.
0·5 ,, 1·0 ,,	heavy benzol.	20 to 30 per cent	heavy tar oil.
0·5 ,, 1·0 ,,	carbolic acid.	50 ,, 60 ,,	pitch.
4·0 ,, 6·0 ,,	naphthalene.	5 ,, 10 ,,	carbon.
0·3 ,, 0·5 ,,	anthracene.		

The water in low-temperature tar contains only a few ammoniacal constituents; that is to say, the greater part of the nitrogen contained in the coal remains there. The nitrogen is only driven out of the coal when subjected to high-temperature treatment, with which ammoniacal compounds are deposited.

Medium and lubricating oils, asphalt, and paraffin can be obtained from low-temperature tar by suitable treatment, while ammonia, oils of various grades, and pitch can be produced from gas-plant and coking tars.

Benzol is obtained from light oil. The greater part of the benzol

used to-day, however, is obtained from washing the coke-plant gas.

When coal is distilled under high pressures, the composition of the tar distillate is different. At 20 atmospheres and 1100° F., aromatic compounds and oils with high boiling-points are obtained. The quantities of gas and coke produced are greater. Tar can also be extracted from fuels by means of benzol. At atmospheric pressure, the extract amounts to 0·5 per cent of the fuel used, while at 55 atmospheres and 500° F., an extract of about 6·5 per cent is obtained. The extract is similar to petroleum. Extraction can also be carried on by means of SO_2, 10 per cent of soluble matter being obtained, and a disintegrating action being produced on the solid residue.

4. COKE

The difference in the constitution of the coke residue after low-temperature distillation (known as semi-coke) and that produced by high temperatures is shown in the following analyses. The most important difference is the presence of considerable volatile matter in semi-coke, although not all semi-coke exhibits so great an amount as that given.

(*a*).	Volatile.	C.	H.	O.	N.	S.
	Per Cent.	Per Cent.	Per Cent.	Per Cent.	Per Cent.	Per Cent.
Coal ..	39·7	82·2	5·2	8·7	2·1	1·8
Semi-coke	17·2 to 18·3	80·9	3·9	7·5	1·9	1·8
Coke ..	2·1	96·6	0·4	1·6		1·4

(*b*).	Volatile.	Ash.
	Per Cent.	Per Cent.
Coal	37·5 (moisture 2·5)	9·3
Semi-coke	15 to 16	18·2

The physical and chemical properties of coke are as follows.

Semi-coke is a soft, spongy substance which easily crumbles. It is, therefore, difficult to use as a fuel for direct firing. When ground into dust, however, it can be used in a dust furnace. As it is more easily ignited than high-temperature coke, owing to its gas contents, a dust furnace does not have to overcome the same difficulties as are encountered with a coke-dust furnace. It also burns with flames, unlike coke, which is practically without flame. Certain coal produces semi-coke which has not the property of crumbling, and can thus be burnt in the furnace without further treatment. The combustion of semi-coke, which has lost a part of the hydrocarbons of the coal, takes place without the formation of smoke. The calorific value of semi-coke is approximately equal to that of coal, so that the substance is not a low-percentage fuel.

Coke from a coking furnace is denser and more solid than gas-plant coke. The calorific value of both kinds is about 13,000 B.Th.U. per pound, with 8 to 11 per cent ash content and 0·5 to 1·2 per cent water. The substance produced with brown coal is lignite coke. This contains on an average 20 per cent of water, and 15 to 20 per cent ash, and has a lower calorific value of 7000 to 8000 B.Th.U. per pound. Lignite coke is granular, and glows faintly when ignited under atmospheric conditions.

5. PLANTS FOR DISTILLATION

The character of the plant depends not only on the character of the fuel to be used, but also upon whether the solid residue, the gaseous distillate, or

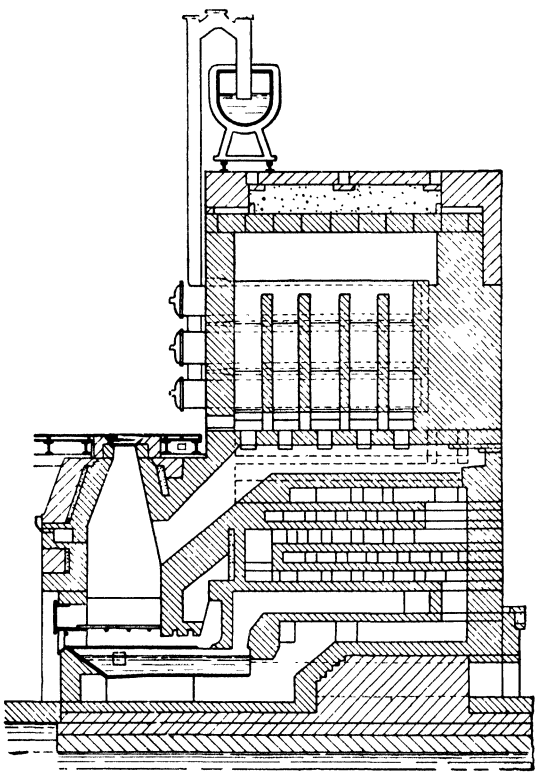


Fig. 1.—Horizontal Retort Gas Plant

the liquid distillate is the main object of the process. Furthermore, not only the quantity but also the properties of the main product may be varied within wide limits with suitable variation of the plant. The most important distillation plants are coal-gas plants and coke plants. In both cases the coal is

subjected to temperatures of 1500° F. to 2500° F. in closed retorts or chambers. In the former a maximum gas output is the main object, and in the latter a hard dense coke is the main requirement. The difference in the *modus operandi* is that in coke plants the coal is packed tightly, so that when it tends to swell on the application of heat and first liberation of gas there is a mutual application of pressure between the particles, preventing both the liberation of further gas and any further swelling.

Plants for generating coal gas are shown in figs. 1 and 2. In the older type, shown in fig. 1, the retorts are horizontal, while those in the latter type are inclined. A plant of a different character is shown in fig. 3, in which the retorts are vertical. The average length of horizontal and inclined retorts varies between 10 and 18 ft., while vertical retorts are 18 ft. and over. Horizontal and inclined retort furnaces are charged periodically at intervals of 8 to 9 hr. Some types of vertical retort furnaces are also charged periodically at 10-hr. intervals, while other types are charged continuously. In furnaces with a combustion chamber, charging takes place every 24 hr. Recently, as the result of experience with coke plant, walled-in chambers have been used in place of retorts. A favoured type is the inclined chamber furnace of fig. 4. A shaft with a rectangular cross-section takes the place of the retort. The furnace is stoked mechanically.

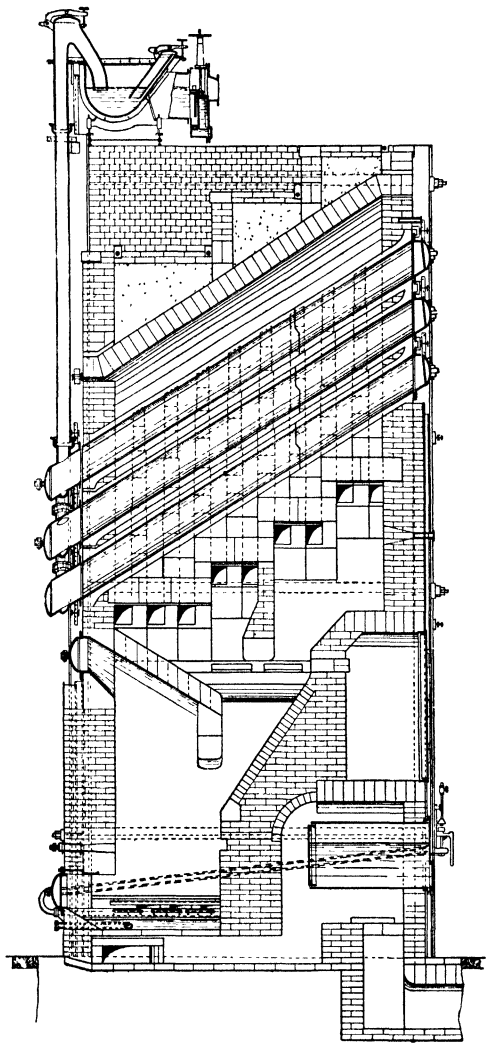


Fig. 2.—Section through an Inclined Retort Gas Plant

The following data are given as examples of the results obtained with retorts of different types.

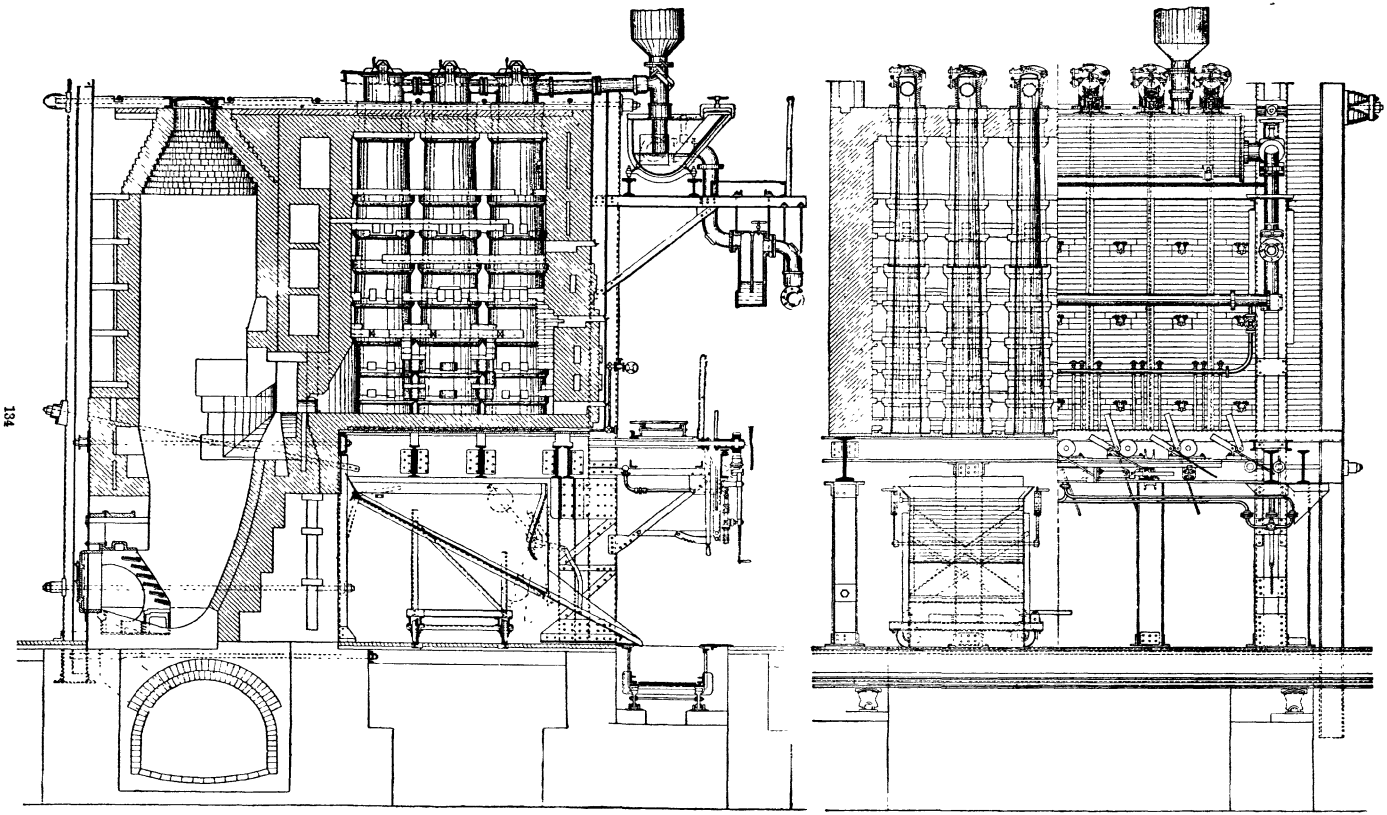


Fig. 3 —Dessau Vertical Gas Plant

Retort.	Horizontal.	Inclined 32°.	Vertical.
Coal.	Saar.	Ruhr.	Silesian.
Average retort charge.	1450 lb.	1100 lb.	1100 lb.
Gas yield from 100 lb.	470 c. ft.	470 c. ft.	615 c. ft.
Lower calorific value.	605 B.Th.U. per c. ft.	615 B.Th.U. per c. ft.	505 B.Th.U. per c. ft.
Coal consumption:			
per 100 lb. in retort.	12·6 lb.	14·0 lb.	14·1 lb.
per 100 c. ft. of gas.	2·8 lb.	3·0 lb.	2·3 lb.

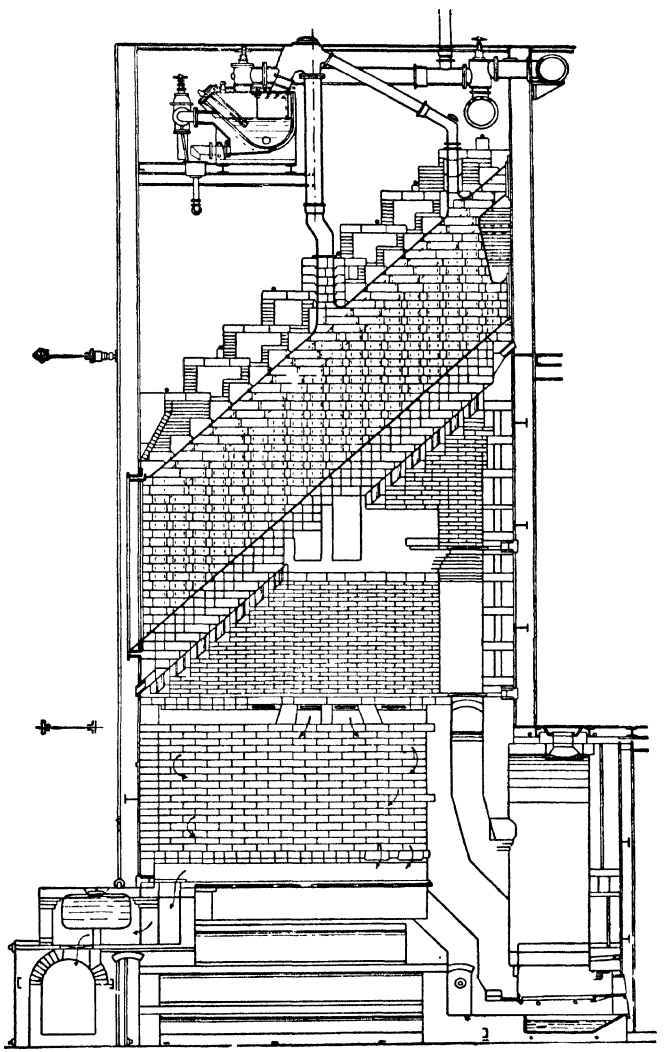


Fig. 4.—Inclined Chamber Gas Plant

The figures show a high gas yield with the vertical furnace, but the calorific value is less. In furnaces with a combustion chamber, the combustion figures

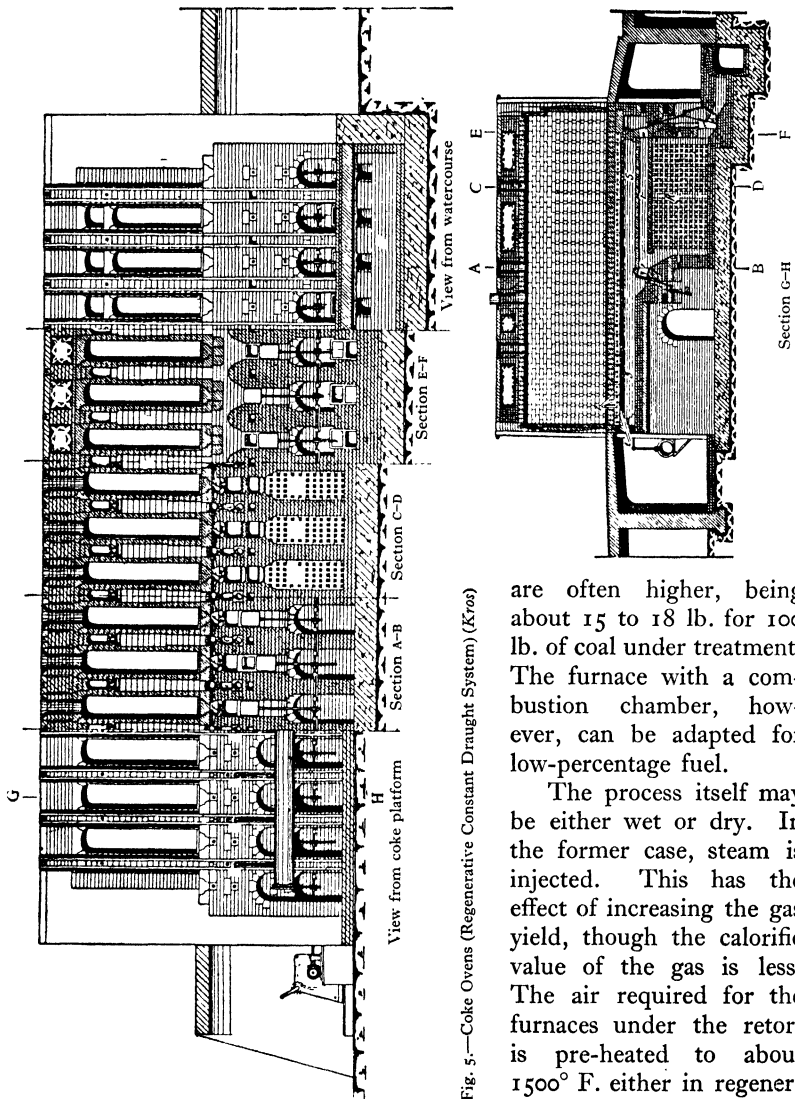


Fig. 5.—Coke Ovens (Regenerative Constant Draught System) (*Kros*)

are often higher, being about 15 to 18 lb. for 100 lb. of coal under treatment. The furnace with a combustion chamber, however, can be adapted for low-percentage fuel.

The process itself may be either wet or dry. In the former case, steam is injected. This has the effect of increasing the gas yield, though the calorific value of the gas is less. The air required for the furnaces under the retort is pre-heated to about 1500° F. either in regenerators or recuperators. High temperatures up to 2500° F. are obtained in modern furnaces in order to obtain a high gas yield. The average composition of coal gas by volume is:

49 per cent H_2, 34 per cent CH_4, 8 per cent CO,
1 per cent CO_2, 4 per cent N_2, 4 per cent $C_2H_4 + C_6H_6$, &c.

The yield from 100 lb. of good gas coal is 65 to 68 lb. coke, 5 lb. tar, 8 lb. gas liquor, 16 to 19 lb. gas (450 to 600 c. ft.). The above gas is often varied by an addition of 20 per cent of water gas, giving a mixture whose composition by volume is:

50 per cent H_2, 27 per cent CH_4, 14 per cent CO,
1 per cent CO_2, 4 per cent N_2, 4 per cent $C_2H_4 + C_6H_6$, &c.

In coking furnaces, the most important product is hard metallurgical coke. They are very similar to the vertical gas generator, and one design is shown in fig. 5. Some years ago " bee hive " furnaces (similar to wood piles for making charcoal) were used in Europe and America, these being replaced first by the Appolt and Coppée furnaces, and to-day by the distillation furnaces represented in fig. 5. The Coppée furnaces, as improved by Otto, had

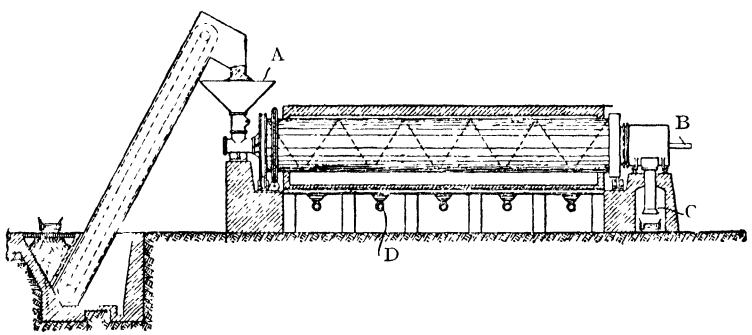


Fig. 6.—Revolving Type of Low Temperature Coal Distilling Furnace (*Thyssen*)

A, Coal supply. B, Discharge of distilled gas and tar vapours. C, Low temperature coke. D, Gas burner.

shafts 25 to 30 ft. long, 2 ft. wide, and 4·5 to 5 ft. high, which were surrounded by fireclay and placed one beside the other in batteries. The intermediate space was heated by the waste gases of the coking process. The yield from 100 lb. of dry coal was about 70 to 80 lb. of coke. The temperature is usually somewhat lower than in the manufacture of coal gas, being about 1800° F. The coking furnace is heated either by its own gases, or by gas from a special generator, or by blast-furnace gas of 85 to 135 B.Th.U. per cubic foot. The coking gas contains 450 to 460 B.Th.U. per cubic foot. The furnaces use about 1300 B.Th.U. for 1 lb. of coal under treatment. About 13,500 to 15,000 c. ft. of gas per ton of coke extracted is obtained when the furnaces are heated with gas from some external source. If the heating takes place by means of the gas extracted, an excess of 3000 to 4000 c. ft. remains for other purposes. This includes the heating of the auxiliary apparatus.

The waste gases of coke plants escape at a temperature of about 900° F., and the tar, ammonia, and benzol are separated out. These crude products then undergo further rectification. Modern coke furnaces produce, from 100 lb. of mineral coal, 70 to 80 lb. coke, 2 to 3 lb. tar, 1 lb. benzol, and 0·1 lb. ammonia, together with 900 to 1100 c. ft. of gas.

English cannel coal is particularly suitable for the production of a large yield of low-temperature tar, the extraction being carried out in upright rectangular or round cast-iron retorts which are heated by the waste gases. In Germany the plant most in use is Thyssen's revolving furnace, shown in fig. 6. Using gas coal in a Thyssen furnace, a yield of 10 per cent of tar free from water is obtained, while crude brown coal yields 6·5 per cent. The thermal efficiency of distillation is 90 per cent. The resulting semi-coke is burnt or gasified according to the type of coal.

6. GASIFICATION

The object of gasification is to convert as much as possible of the available fuel into gas. Some low-temperature tar may be extracted during the gasification, but in general it is gas only that is required. In order to increase the yield of gas, instead of distilling the fuel in a closed retort air is introduced and much of the carbon converted to carbon monoxide. The heat of combustion of carbon monoxide to carbon dioxide remains in the gas extract, but the gas is diluted by the atmospheric nitrogen introduced and the calorific value is much smaller than with distilled coal gas. The generators used are of two main types, fixed-grate gas producers and revolving-grate producers.

In an efficient gas producer, as much fuel as possible should be burnt per square foot of shaft section in order to reduce the percentage loss by radiation and conduction from the walls. In addition, the stoking should be made as simple as possible. The automatic removal of ash, whether solid or liquid, is important, either a wet or dry removal being possible. The formation of carbon monoxide should be aimed at, while that of carbon dioxide should be avoided as far as possible. The high temperatures that lead to the formation of carbon monoxide bring about the danger of the ash melting and the formation of lumps.

The removal of heat on the side walls by radiation and conduction to the outside should be reduced to a minimum. This can be effected by the insertion of firebricks, or in many systems by the utilization of the outer cover as a steam boiler. It is particularly important that the height of the burning zone in the generator should suffer no variation. This is ensured by a suitable supply of steam and air over the whole cross-section. The distribution of this supply should be so arranged that the outer part of the section, which undergoes a greater cooling by conduction, should receive a greater amount of air for the formation of carbon dioxide and the production of heat. Exactly the opposite is required with steam. In order to obtain a uniform charge care must be taken in the supply of the fuel. This is effected by means of rotating feed pipes, and by the fuel being supplied at a suitable angle of inclination. Special attention should be given to the necessity of distributing unequal material, such as large and small lumps and dust, uniformly over the cross-section. If care is not taken, the large lumps always tend to fall outwards on to the edge of the shaft, while the small pieces fall into the

centre and the dust round the axis of the shaft. This is remedied by the insertion of guide plates in the feed pipe. In addition to the importance of the supply of fuel, air, and steam, the removal of ash is also a leading factor in the retention of a uniform charge depth and the uniform generation of gas. The periodic removal of slag in generators with fixed grates results inevitably in a displacement of the layers of fuel, and a disturbance of the fuel zone required for an effective gas formation. This disadvantage does not exist in generators with revolving grates and those without grates, which have a continuous falling of ash. Even in these plants, however, the fuel charge has to be poked with stoking rods in order to break up the lumps of slag. Generators without grates have a conical shape, narrowing towards the bottom, as the fuel introduced above is reduced in volume after distillation and gasification.

The upper portion of the layer of fuel is at a much lower temperature than that below it. A smouldering combustion takes place from which the upper part of the generator and the feeding funnel receive heat. The depth of the whole fuel layer should be such that the temperature of any part of the furnace is sufficiently high to prevent condensation of the distilled gases. The condensation temperature varies between 900° F. and 1500° F. with dry fuels, and is correspondingly lower with wet fuels. The depth of the fuel charge also varies with the amount of fuel consumed per hour per square foot of shaft cross-section.

Unlike furnaces, the output of gas generators cannot be increased greatly by an increase in the supply of air or steam. The following are average amounts of fuel burnt per hour in pounds per square foot of shaft section.

	Generator with Fixed Grate.	Generator with Revolving Grate.	Gas-suction Plant.
Coke (1·2 in.)	21	32	12
Anthracite (0·6 in.)	21	28	22
Rough mineral coal	13	22	16
Westphalian flame coal ..	24	25	22
Non-bituminous coal ..	24	28	25
Lignite	26	39	22
Brown coal briquettes ..	26	35	22
Bohemian crude brown coal..	31	39	31
Central German crude brown coal	13	18	20
Broken machine-cut peat ..	26	35	22

The lower calorific values of the resulting gases are:

Coke	115 to 120	B.Th.U.	per cubic foot.
Rough mineral coal ..	145 „ 155	„	„
Westphalian flame coal ..	150 „ 155	„	„
Non-bituminous coal ..	155 „ 175	„	„
Brown coal briquettes ..	160 „ 185	„	„
Crude brown coal ..	130 „ 140	„	„

7. FIXED-GRATE GAS PRODUCERS

The fixed-grate generators may have a flat or inclined grate, rectangular or circular. The oldest design is the Siemens generator. Fig. 7 represents a generator for crude brown coal and brown coal briquettes. The air is delivered under the inclined grate, and the gas flows past *a* into the gas pipe. The older generators of this type all work with a natural air draught. In order to be independent of weather effects and to obtain a uniform gas, the more modern generators are designed to use compressed air with an admixture of steam. A design of this kind is given in fig. 8. Smaller plants have a horizontal grate in place of the inclined grate. They are often attached directly to retorts, muffles, or boilers, which are termed semi gas-fired. The gases have both sensible

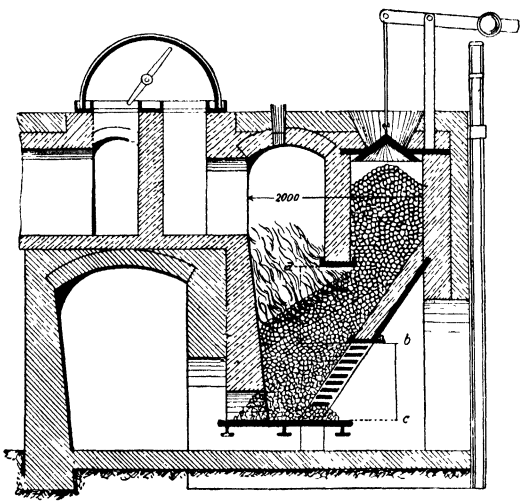


Fig. 7.—Crude Brown Coal Generator

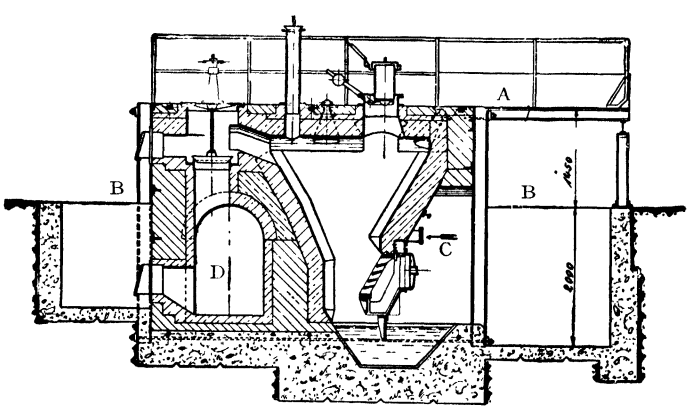


Fig. 8.—Inclined Grate Generator (*W. Ruppmann*)

A, Coal platform. B, Foundry floor. C, Air and steam connexion. D, Gas collecting vessel.

heat and calorific value on leaving the producer, both of which can be used if the plant is adjacent to it. With a fairly low coal charge a hot gas is produced and utilized, principally in industrial furnaces.

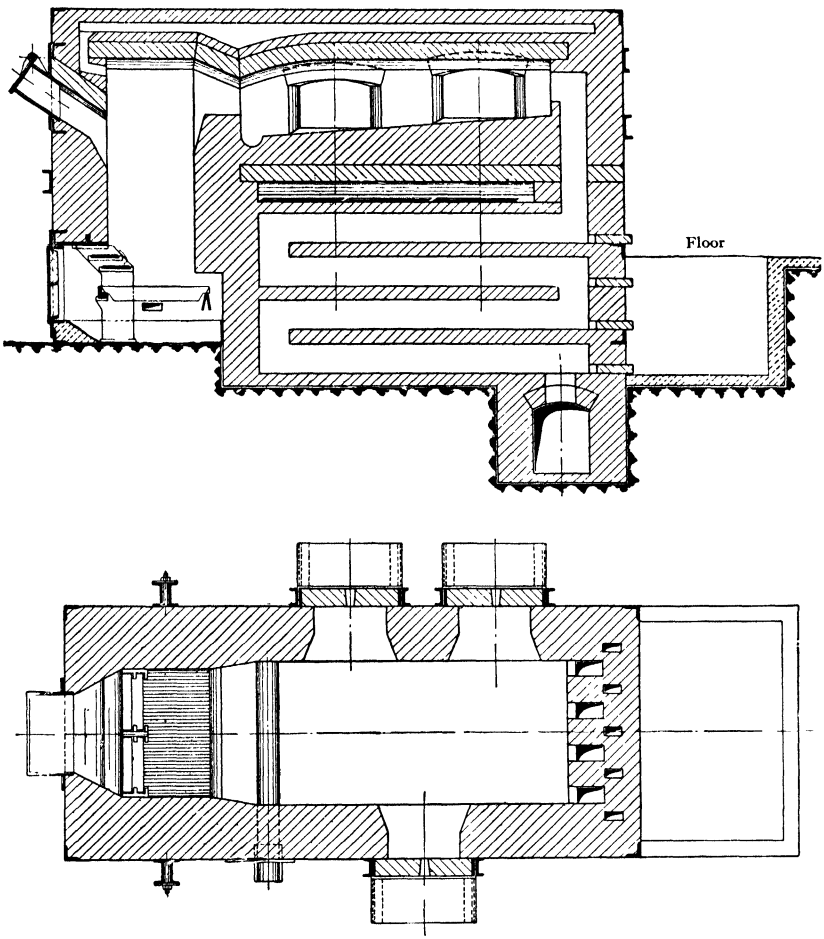


Fig. 9.—Forge Oven with Semi-gas Firing

A design of this character is shown in fig. 9. Some data concerning the loading and output of generators is given in the following table for mineral coal.

FLAT-GRATE GENERATORS

Size of Grate.	Load.	Coal Consumption.	Gas Production.
2·7 sq. ft.	37 lb./sq. ft./hr.	100 lb./hr.	320 c. ft./min.
3·8 ,,	36 ,,	136 ,,	425 ,,
6·9 ,,	35 ,,	242 ,,	740 ,,

INCLINED-GRATE GENERATORS

Width of Grate.	Width of Shaft at Upper Edge of Grate.	Load.	Coal Consumption.	Gas Production.
2·0 ft.	1·6 ft.	27 lb./sq. ft./hr.	85 lb./hr.	300 c. ft./min.
3·3 ,,	2·8 ,,	25 ,,	230 ,,	740 ,,
5·0 ,,	2·8 ,,	23 ,,	320 ,,	990 ,,
5·9 ,,	3·3 ,,	21 ,,	410 ,,	1340 ,,

These values can be increased by 30 per cent with crude brown coal and

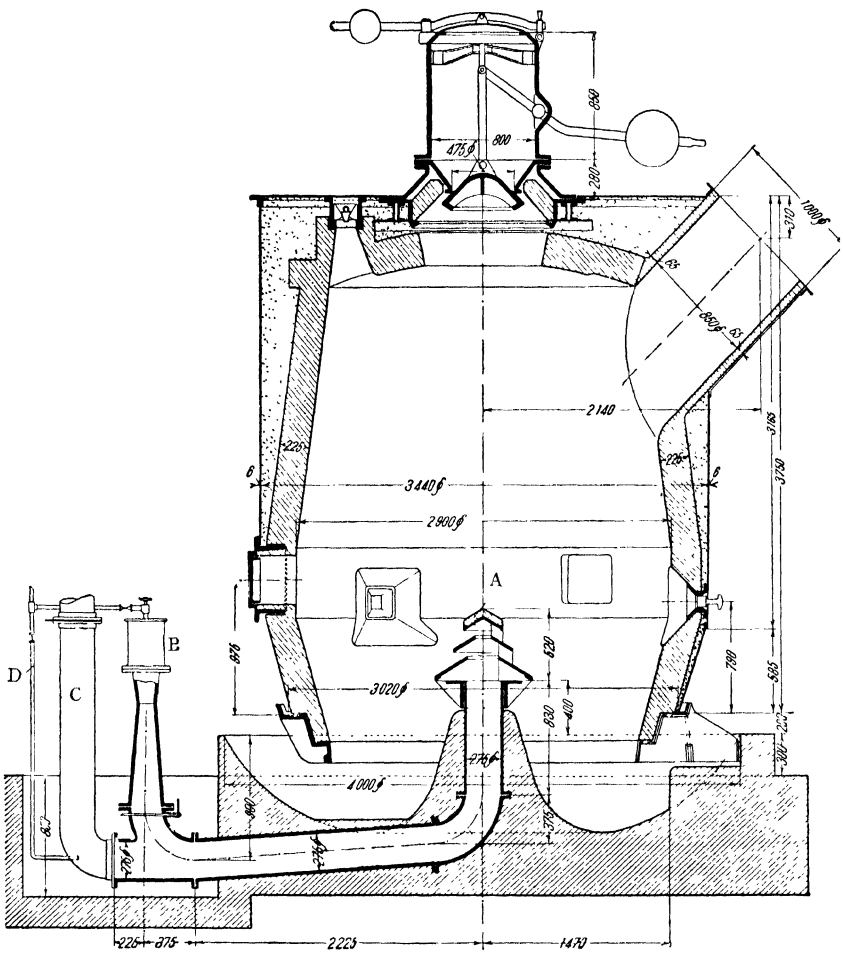


Fig. 10.—Morgan Gas Producer

A, Fireproof cover. B, Steam-jet blower. C, Air inlet pipe. D, Steam inlet.

brown coal briquettes. With dry coal an addition of steam of 0·15 to 0·30 lb. per pound of gasified coal is made, but not with wet coal or crude brown coal. These generators are not suitable for small or powdery coal, and only give good results with lump coal.

A circular fixed-grate generator is shown in fig. 10. In these types the ash and slag is necessarily removed intermittently. During this period the generation of gas is irregular or interrupted. The adjustment of the fire zone also presents difficulties. It is easy for the fire to sink downwards, thus causing the slag to melt and the air pipe to become dirty and stopped up. Powdery fuels and fuels with slag that easily cakes are unsuitable for these generators.

Fig. 11 represents a generator with a fixed grate on top of which is placed a double-armed cleaner. This grate is particularly suitable for the gasification of crude brown coal. The cleaner rotates over the grate and removes the ash through the openings in the fixed grate, leaving the piles of crude brown coal undisturbed. The gas yield should be higher, as with a fixed grate it is more easily possible to inject the requisite amount of air perpendicularly, and at the same time to remove the ash effectively.

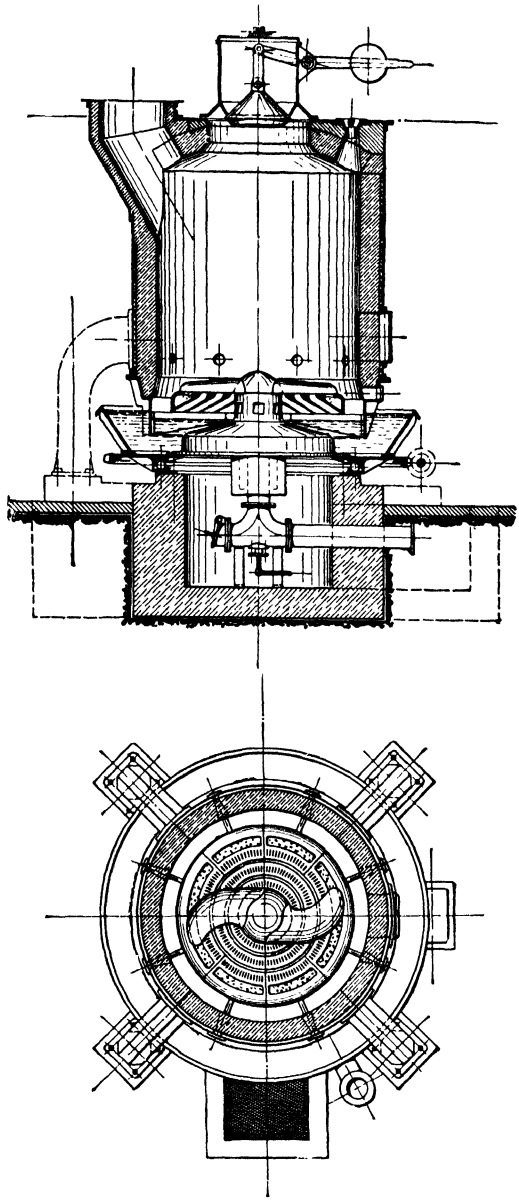


Fig. 11.—Gas Generator (*Eisenhüttenwerks*)

8. REVOLVING-GRATE GAS PRODUCERS

In order to produce a more continuous operation and to eliminate the variations in output due to intermittent stoking, the revolving-grate gas producers have been designed. The motion of the grate and fuel also reduces

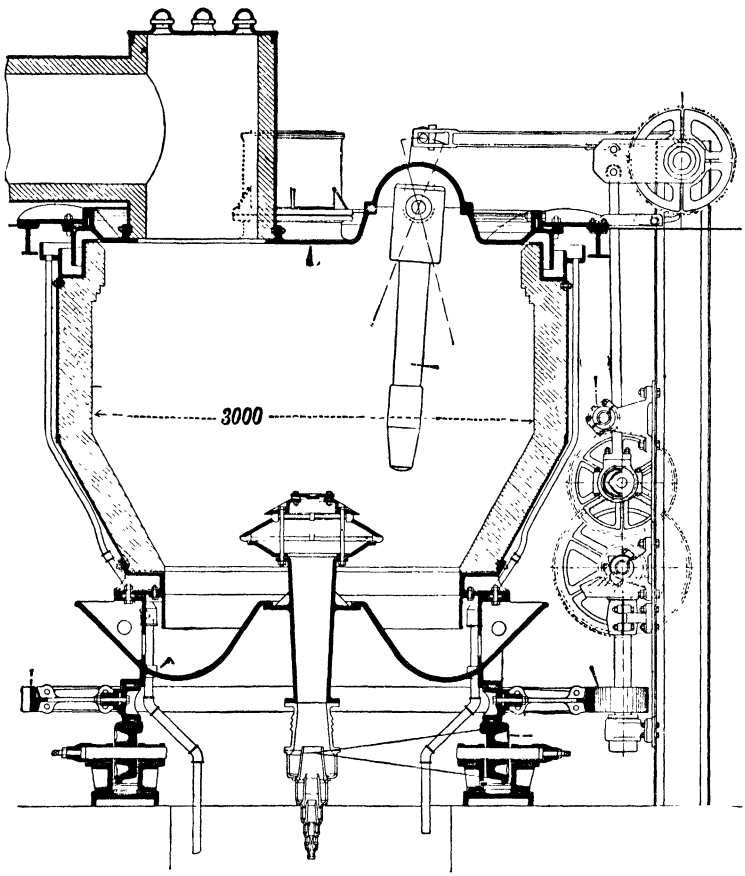


Fig. 12.—Hughes' Gas Producer

the tendency of the slag to become fixed, and keeps the grate cleaner. The grates are driven by cog-wheel or screw drive, and like the fixed grates, they may be flat, domed, or conical. Air and steam is injected between the grate bars. In order to distribute the fuel uniformly and to enable the ash to be removed easily, mechanical stoking apparatus is used for stirring up the fuel. The rotation of the grate causes the ash to accumulate at the lowest part.

Hughes' gas producer (fig. 12) has a lever for stoking the fuel. It is designed for a plant using washed coal nuts of 14,000 B.Th.U. per pound. The gas generated contains

10 to 11 per cent H_2, 2·5 per cent CH_4, 21 to 24 per cent CO, 5 to 6 per cent CO_2, 0·18 per cent C_2H_4, 0·1 per cent O_2,

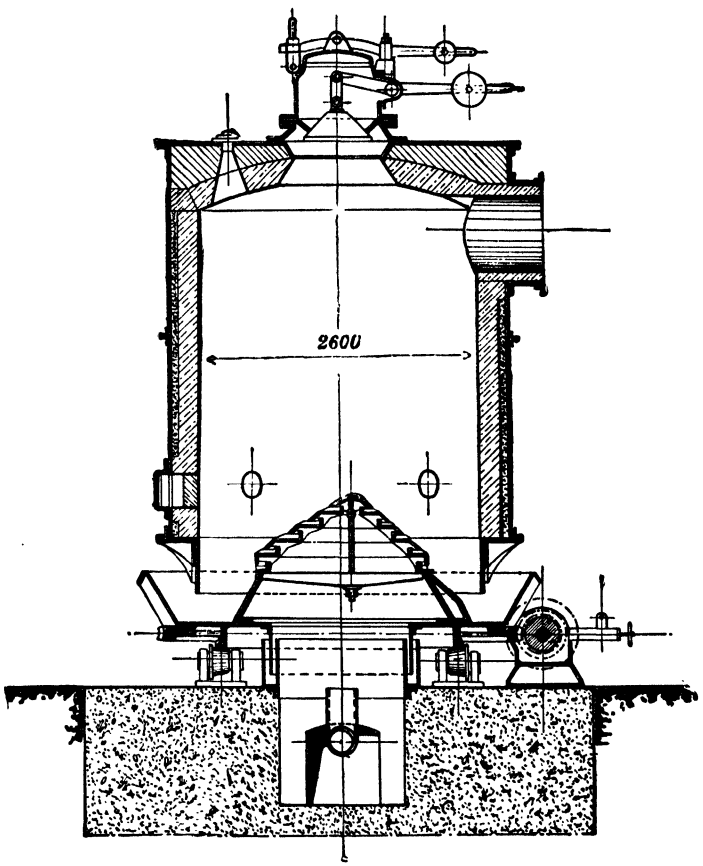


Fig. 13.—Revolving Grate Gas Producer (*Ruppmann*)

and its calorific value is 136 B.Th.U. per cubic foot. With an internal diameter of 10 ft., 1300 lb. of coal is gasified per hour, i.e. 17 lb. per square foot of shaft area. The injected steam required amounts to 30 lb. per 100 lb. of fuel at a pressure of about 1·3 atmospheres. The gas yield is 70 c. ft. per pound of fuel, with 6 oz. of water in 100 c. ft. The ash contains 6 to 20 per cent of combustible constituents. The power required for driving the generator shaft and stoking lever is about 3 h.p.

Figs. 13, 14, and 15 show other types of revolving-grate generators.

In fig. 16 is shown the design of a generator in which the heat of the shaft walls is used to produce steam, the boiler being formed as a jacket round the gas generator. This generator is used largely for the gasification of rough

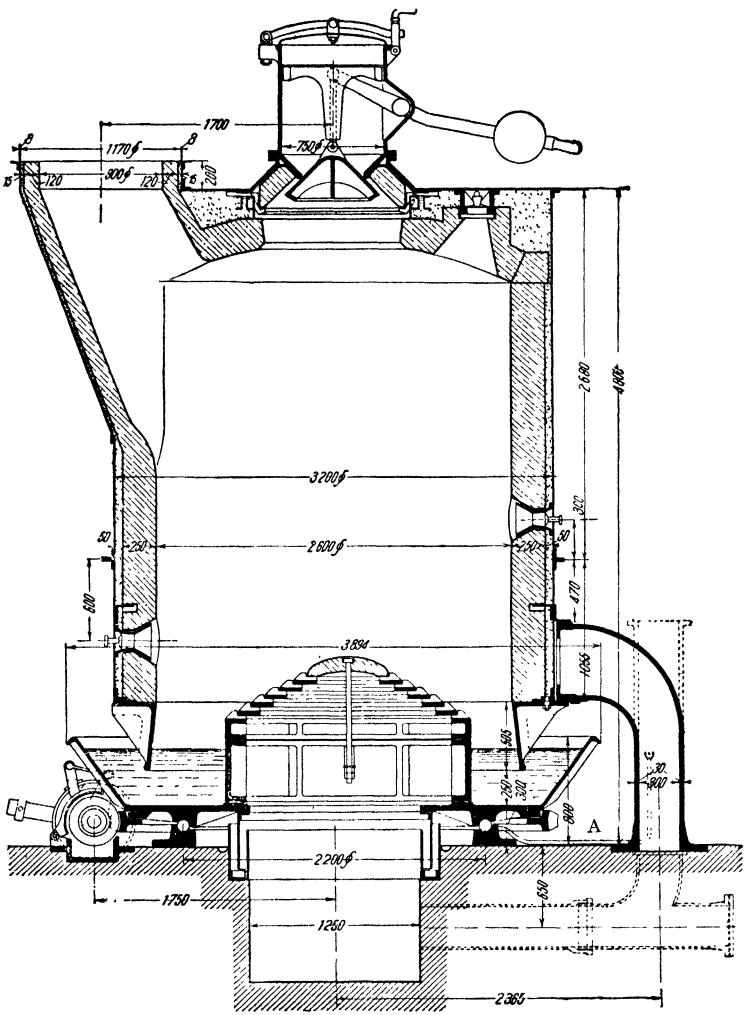


Fig. 14.—Revolving Grate Gas Producer (*Ehrhardt & Sehmer*)

A, Pipe for ball-bearing lubrication

coal. In a plant whose boiler area was 640 sq. ft., 2 lb. of steam was produced per square foot per hour, and 1·4 lb. of steam per pound of coke. The calorific value of the gas represented 75·4 per cent of that of the fuel, which was coke, while the steam contained 14·4 per cent, giving an efficiency

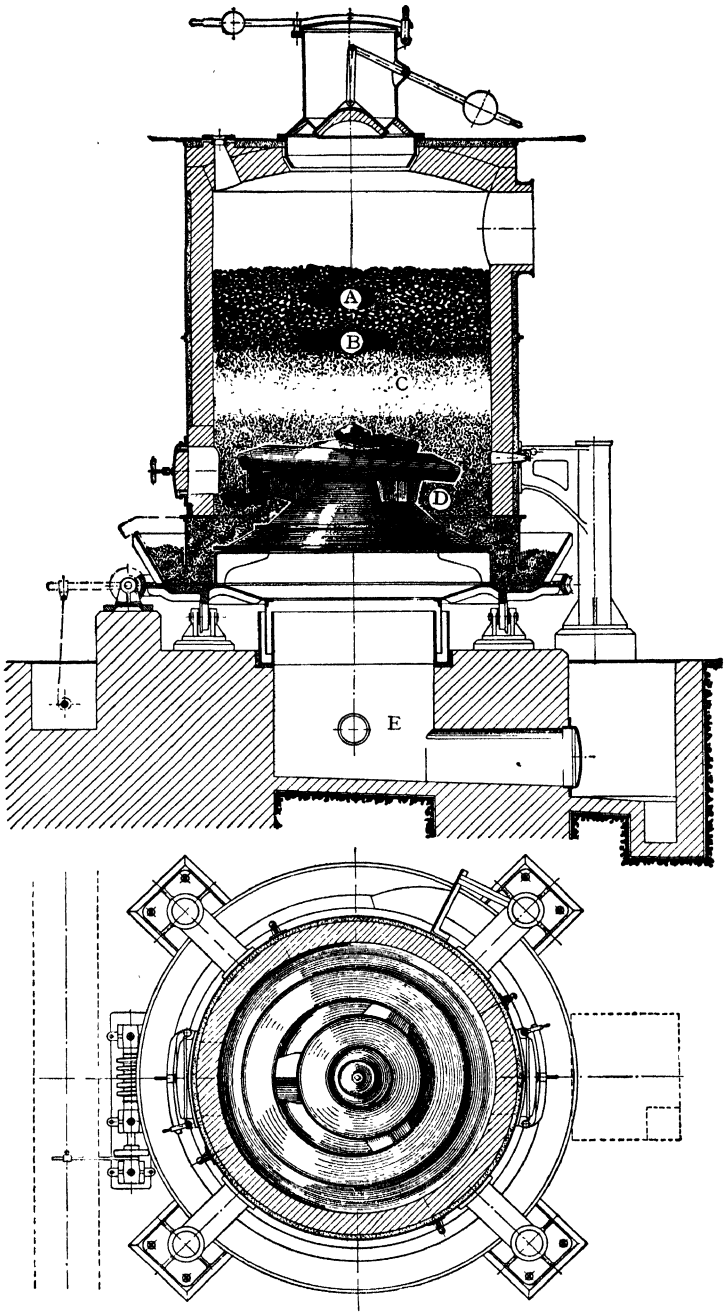


Fig. 15.—Gas Producer with Annular Grate for Crude Brown Coal (*Gasgenerator & Braunkohlenverwertung*)
A, Drying zone. B, Distillation zone. C, Gasification zone. D, Ash bed. E, Air.

of nearly 90 per cent. About 8 per cent of the steam was used in the generator itself.

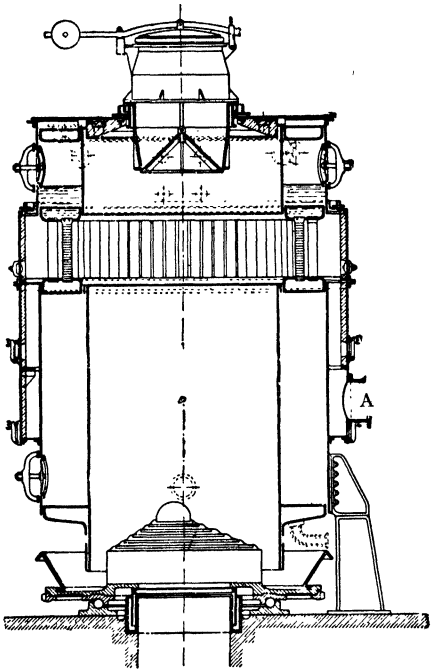


Fig. 16.—Combined Revolving Gas Producer and Steam Boiler (*Morischka*)

A, Gas discharge

The load of revolving-grate generators varies between 17 and 20 lb. per square foot per hour with mineral coal, and increases to 27 lb. with crude brown coal.

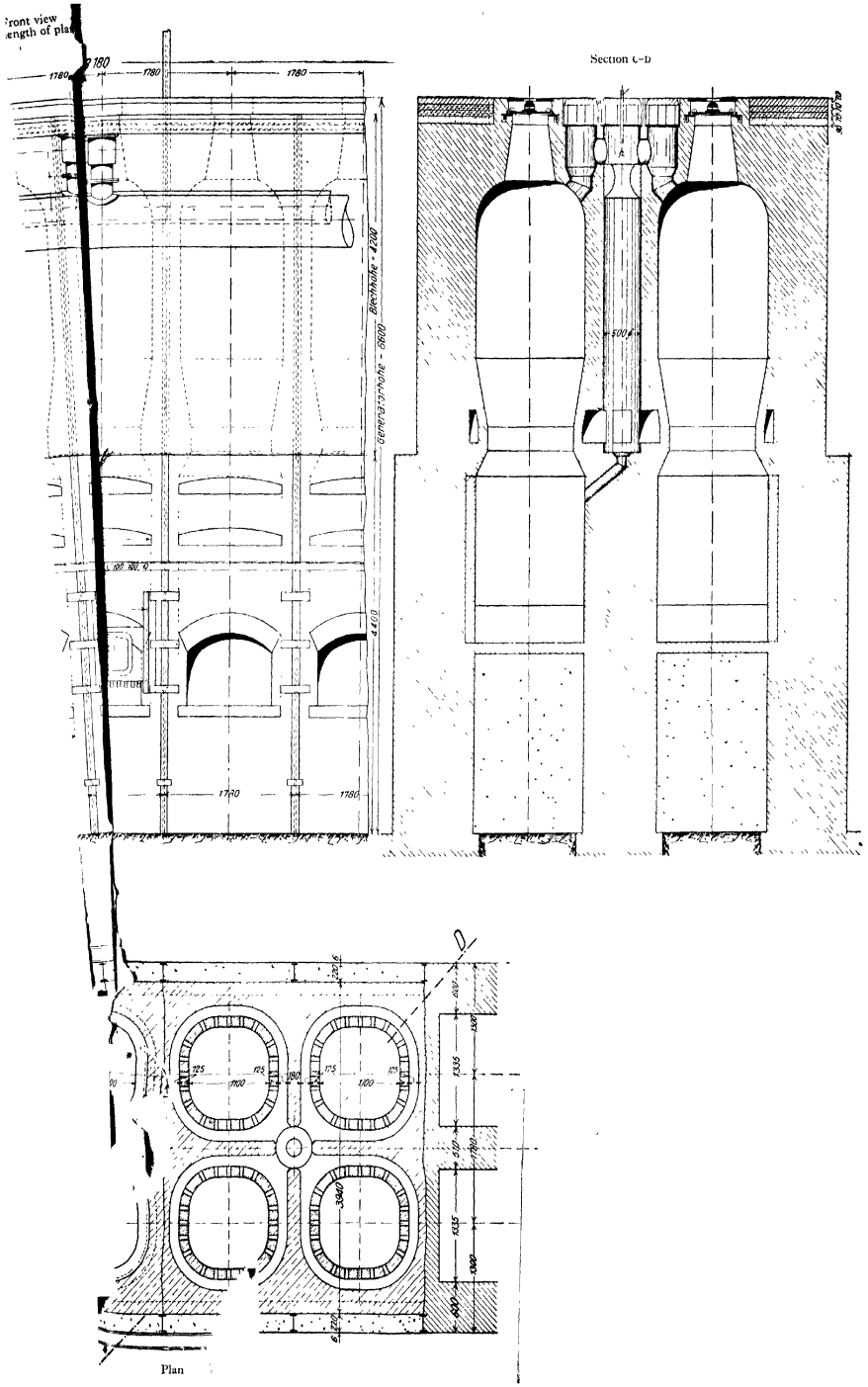
9. GASIFICATION OF INFERIOR FUELS, WOOD, AND PEAT

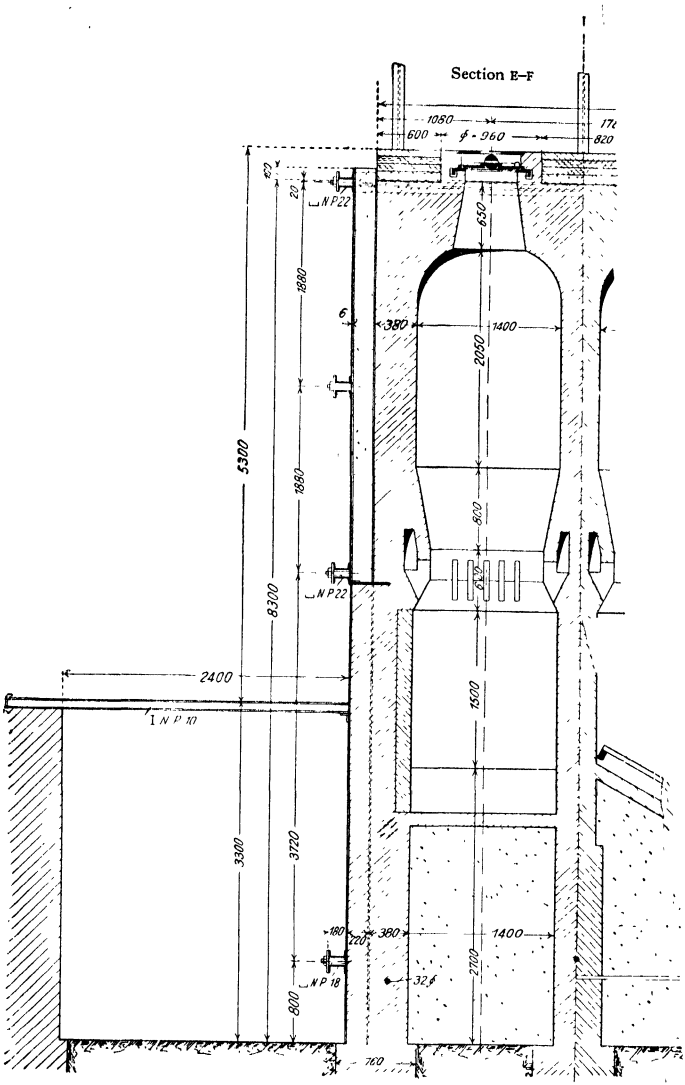
Fig. 17 shows a continuous circuit generator for treating inferior fuels, including what would otherwise be the waste product in the preparation and grading of mineral coal. Every four chambers form a circuit. While the action is taking place in three chambers, the fourth, with a fresh charge, is receiving a preliminary heating, (1) by means of the brickwork which is still red-hot from the old charge just removed, (2) by the mass of fallen ash from the previous process, (3) by heat from the adjacent chambers. During this preliminary heating, gas free from tar is distilled. The gasification is then completed at a higher temperature with an addition of steam and air. With shale coal containing 10 to 28 per cent of coal, 3·5 to 4·5 per cent of water, and 9 to 14 per cent volatile constituents, and with a calorific value of 4300 B.Th.U. per pound, the gas produced has an average composition of

18 to 22 per cent H_2, 2 to 3 per cent CH_4, 9 to 10 per cent CO,
11 to 12 per cent CO_2, 53 to 56 per cent N_2, 0·2 to 0·3 per cent O_2,

and has a calorific value of 120 B.Th.U. per cubic foot. The slag melts at 1650° F. to 1750° F., and the process in the generator is carried out at 1850° F. On leaving the generator the gases have a temperature of about 900° F. With a low-percentage Bohemian brown coal of 14 per cent slag content, the gas contains

21·6 per cent H_2, 4·5 per cent CH_4, 24·6 per cent CO,
6·2 per cent CO_2, 40·5 per cent N_2, 0·8 per cent O_2,


Front view
Length of pla
Section C-D
Plan


Section E–F
1080
600
4 - 960
820
650
20
NP22
1880
380
1400
2050
5300
1880
800
1500
8300
NP22
2400
I N P 10
3300
3720
180
380
1400
120
NP 18
32,6
2700
800
760

while the calorific value is about 190 B.Th.U. per cubic foot. Gas from crude brown coal from Westerwald has a composition

17·1 per cent H_2, 5·3 per cent CH_4, 14 per cent CO,
12·5 per cent CO_2, 50·7 per cent N_2, 0·4 per cent O,

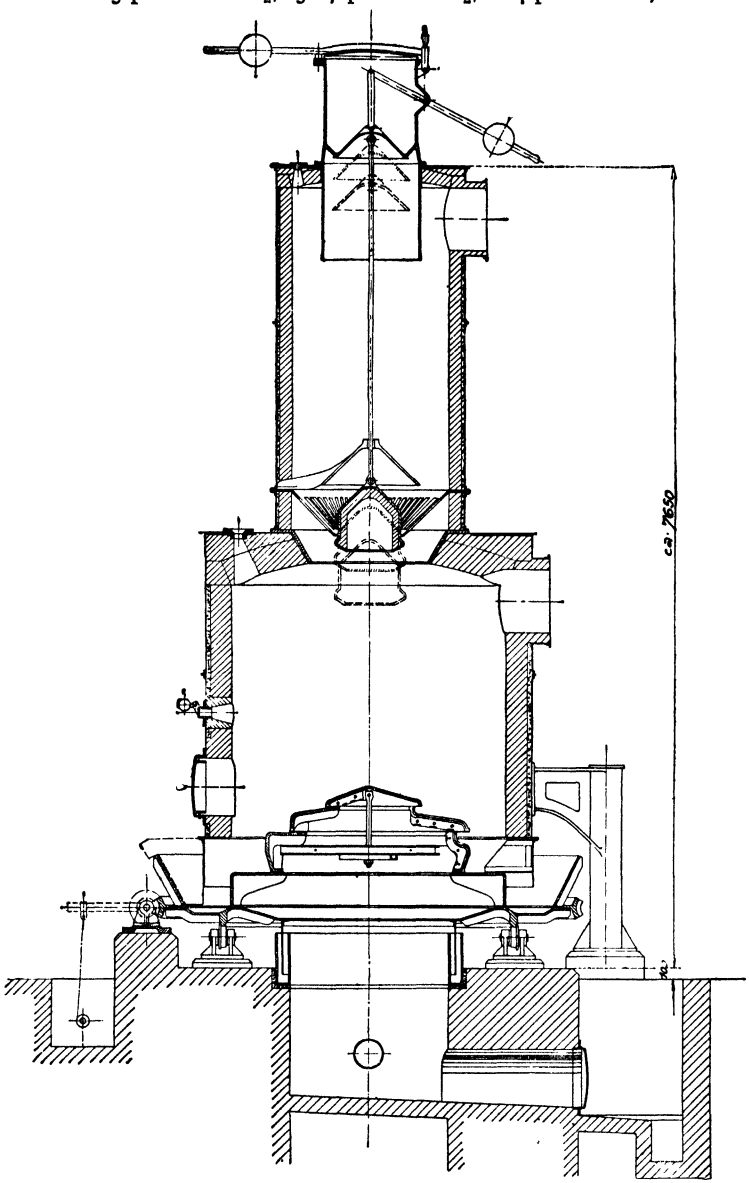


Fig. 18.—Gas Producer with Sulphur Apparatus for Crude Brown Coal and Brown Coal Briquettes for the Production of Low Temperature Tar and Pure Gas (*Gasgenerator & Braunkohlenverwertung*)

and a calorific value of 145 B.Th.U. per cubic foot. The cycle of operations requires a time of 31 hr., and each chamber has a capacity of 4 tons.

Simple shaft generators of this type are also suitable for use with the oil shale that is found extensively in Germany. The process can be carried out as a smouldering or gasification process. The extraction of low-temperature tar and oil comes under the former head, while the gasification is more favourably carried out by adding brown coal or peat, on account of the high ash content of the shale.

Although in these gasification processes the tar may be considered

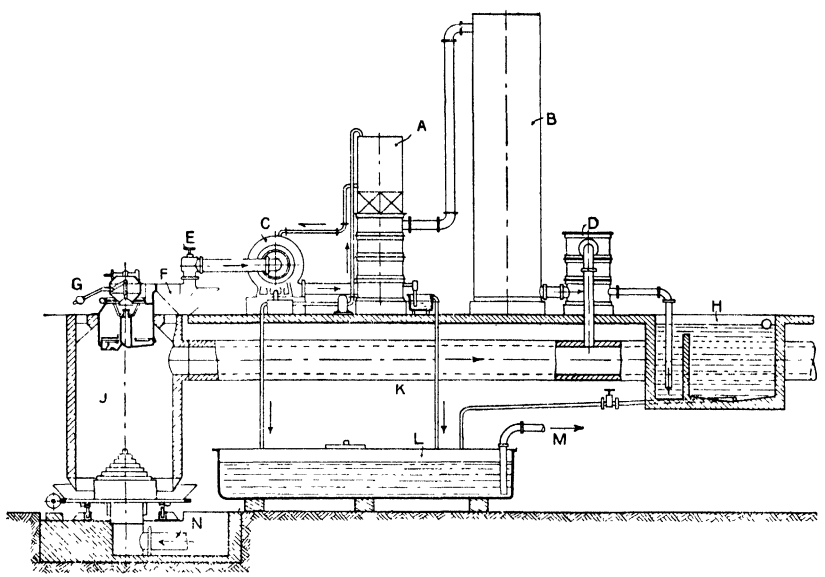


Fig. 19.—Generator with Plant for Extracting Low-temperature Tar (*Ehrhardt & Sehmer*)

A, Tar container. B, Condenser. C, Disintegrator. D, Drop collector. E, Stop valve. F, Dust separator G, Hand control. H, Tar separating tank. J, Generator. K, Main gas discharge. M, Tar pipe for the tar delivery. N, Air pipe.

a by-product, it is of considerable value. In extracting low-temperature tar from brown coal and briquettes, a temperature of 900° F. should not be exceeded. At higher temperatures, the valuable paraffin tars become decomposed to naphthalene tars. The hot combustion products at the bottom of the generator are cooled in the layer above and reduced to carbon monoxide. In the next layer the warm gases at about 900° F. are utilized for extracting the low-temperature tar which is absorbed in the gases. In the generators previously described, the tar can be separated by passing the gases through long cooled pipes in which the water and tar is condensed. As these cooling plants are of large dimensions, another method has been devised, a design being shown in fig. 18. The mixture of gas and tar may be passed out either directly into the main pipe or it may be led into the upper chamber for the

extraction of the tar, the purified gas passing from thence into the main pipe. Fig. 19 shows a generator of a different design with tar-extracting apparatus. With mineral coal about one-quarter of the total gas from the generator should be passed through the tar extractor, and one-third to one-half with brown coal. The generator can be used alone without extraction of the tar, the temperature being regulated to between 400° F. and 500° F. at the surface of the fuel instead of 1100° F. to 1500° F.

For certain purposes it has been found advantageous to have two fire

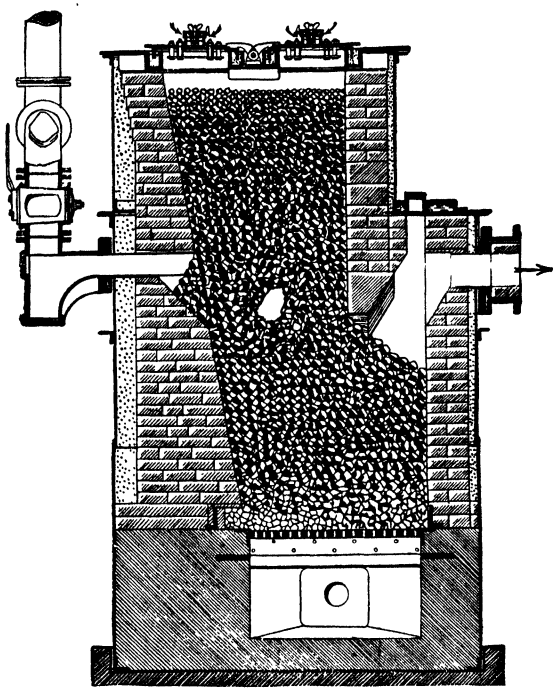


Fig. 20.—Generator with Two Fire Zones

zones in place of one in the generator. This is illustrated in fig. 20, in which there are two fire zones, one directly on the grate, another directly under the charging zone. The gas produced is drawn out in the middle of the generator, which thus works with a partial vacuum. The coal is distilled in the upper fire zone. The tar and water vapours set free are decomposed by passing through the incandescent zone, and flow towards the gas outlet pipe. The distilled coal is gasified in the normal way over the grate, and the resulting gas flows towards the outlet pipe. These generators are particularly suitable for the complete gasification of crude brown coal and brown coal briquettes, also peat and wood.

The following figures represent the performance of furnaces with double fire zone as shown in the last figure.

	Fuel, B.Th.U. per Pound.	Gas, B.Th.U. per Cubic Foot (lower).	Heat utilized, B.Th.U. per Pound.
Crude brown coal ..	3600 to 6300	110	3100 to 5400
Brown coal briquettes	7700 ,, 9000	120	5400 ,, 7200
Peat	5400 ,, 6300	100 to 110	2700 ,, 5400
Wood	5400 ,, 8100	100 ,, 110	2700 ,, 6300

From 1 lb. of crude brown coal the yield of gas may reach 56 c. ft., including the decomposed tar.

Some data with reference to single-furnace generators are given below for comparison with double-furnace generators.

	Generator with Two Furnaces.			Generator with One Furnace.
	Union Briquettes.	Lausitz Briquettes.	Bohemian Crude Brown Coal.	Crude Brown Coal from Central Germany.
C	54·70 %	49·90 %	32·00 %	20·97 %
H_2	4·60 ,,	4·40 ,,	41·00 ,,	32·45 ,,
$O_2 + N_2$ ..	21·10 ,,	27·80 ,,		
Ash	5·80 ,,	5·60 ,,	6·10 ,,	6·33 ,,
Water ..	12·80 ,,	11·30 ,,	20·90 ,,	40·25 ,,
S	1·00 ,,	1·00 ,,	—	—
Calorific value, B.Th.U./lb.	8900	7700	9100	5700
	Gas Free from Tar.	Gas Free from Tar.	Gas with Tar Removed.	Gas with Tar Removed.
CO	19·50 %	21·20 %	16 32 %	24·32 %
CO_2	8·10 ,,	9·20 ,,	9·70 ,,	5·85 ,,
CH_4	2·80 ,,	1·30 ,,	1·68 ,,	2·69 + 0·26 C_nH_{2n} %
H_2	16·20 ,,	14·90 ,,	14·73 ,,	10·56 %
O_2	—	0·20 ,,	0·20 ,,	1·49 ,,
N_2	53·40 ,,	53·20 ,,	57·37 ,,	54·83 ,,
Calorific value, B.Th.U./lb.	140	130	115 to 125, with 0·22 lb. steam per pound of coal.	145; 0·2 lb. tar per pound of coal; calorific value of tar, 5600.

Plants have been constructed in recent years for the gasification of peat. In ordinary times it can scarcely be expected that such processes could be economically successful in competition with coal gasification, but the process may be useful in special circumstances. The Körting generator is shown in fig. 21. Another type, the Görlitz, is shown in fig. 22. In the Körting generator the distilled gases from the upper part of the furnace are not completely decomposed owing to the presence of a large amount of moisture. They are, therefore, conducted through a pipe to the lower part of the furnace in order to be completely decomposed. Peat containing 20 to 50 per cent of moisture can be gasified, a gas of calorific value 115 B.Th.U. per cubic foot being obtained. Its composition is:

10·9 per cent H_2, 2·5 per cent CH_4, 17·6 per cent CO,
13·3 per cent CO_2, 55·7 per cent N_2.

Peat of 5500 B.Th.U. per pound, containing 29 per cent of moisture and 6·1 per cent of ash, 37·5 per cent C, 3·7 per cent H_2, and 23·7 per cent of $O_2 + N_2$, produced a gas of calorific value 133 B.Th.U. per cubic foot, its composition being:

5·9 per cent H_2, 6·2 per cent CH_4, 17·0 per cent CO,
11·2 per cent CO_2, 0·3 per cent O_2, 59·4 per cent N_2.

The Görlitz generator utilizes the sensible heat of the escaping gas in the counter-current for pre-heating the primary air, which is also heated by the radiation from the fireclay walls. The generator itself has no grate. With very wet peat part of the steam is drawn off through a pipe in the generator cover.

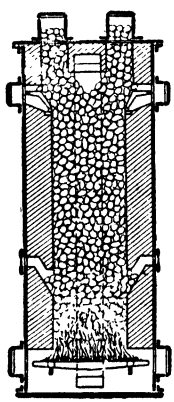


Fig. 21.—Körting Peat Generator

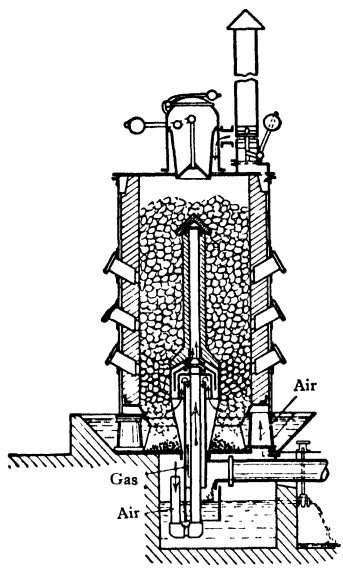


Fig. 22.—Görlitz Peat Generator

Experiments by Dr. Wolff with peat whose upper and lower calorific values were 5000 and 4250 B.Th.U. per pound respectively, and which contained 45·54 per cent of moisture, 2·51 per cent ash, 29·06 per cent C, 3·06 per cent H_2, and 19·56 per cent O_2, resulted in a gas of lower calorific value, 115 B.Th.U. per cubic foot, and composition

19·25 per cent H_2, 0·86 per cent CH_4, 14·96 per cent CO,
14·74 per cent CO_2, 0·28 per cent O_2, 49·73 per cent N_2,
0·18 per cent C_nH_m.

From 1 lb. of peat, 27·7 c. ft. of gas was obtained, so that the efficiency was $27·7 \times 115/4250 = 75$ per cent.

In the Wielandt process, the production of coke from peat is the primary object and not gas (fig. 23). From 10 tons of air-dried peat 3 tons of coke are obtained, or 4·1 tons of coke from 10 tons of peat free from water. The available energy from 1 lb. of peat free from water is:

From the coke, 0·41 × 13,500	=	5530 B.Th.U.	=	61.2 per cent
„ tar	=	1080 „	=	12·0 „
„ gas	=	2400 „	=	26·8 „
Total ..		9010 B.Th U.	=	100 per cent

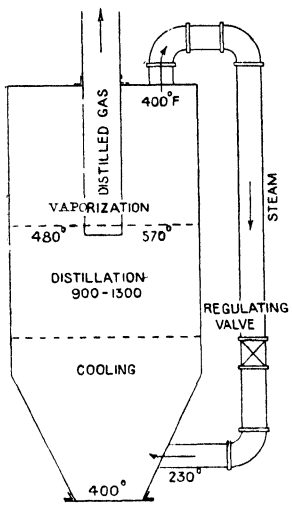


Fig. 23.—Diagram of Peat Cooking Chamber

The 2400 B.Th.U. in the gas are available for drying. Peat containing 25 per cent of water requires 445 B.Th.U. for expelling the water, i.e. 18·5 per cent of that available in gas. If the peat contained 50 per cent of water, 1330 B.Th.U. would be required for drying or 55·5 per cent of that available. It is assumed that the steam and gas escape at 570° F. One pound of dried peat will produce 5·6 c. ft. of gas of calorific value 425 B.Th.U. per cubic foot, with 4 to 5 per cent of tar, 0·5 per cent pitch, and 0·3 per cent of nitrogen in the form of ammonium sulphate, representing about one-third of the nitrogen in the peat.

A gas generator plant which can also be used for wood is shown in fig. 24, in which 100 lb. of wood produces 640 c. ft. of gas of calorific value 336 B.Th.U. per cubic foot, 10 lb. of tar, and 35 lb. of charcoal. The tar can be converted into acetic acid solution, light and heavy wood tar oil, wood tar oil containing paraffin, and black pitch. As the construction and maintenance of grates is expensive, plants without grates and similar to blast furnaces have been constructed. Figs. 25 and 26 show two generators of this type, the Heller generator and the Pintsch generator. The working cycle of these producers does not differ from that with a fixed or revolving grate. Distillation takes place in the upper layers, and fuel free from the light volatile parts sinks down to the neighbourhood of the air pipe. The maximum

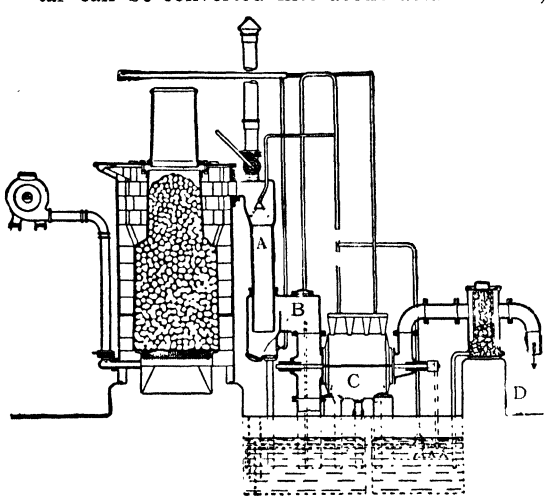


Fig. 24.—Generator Plant for Wood, Peat, and Crude Brown Coal (*Gasmotorenfabrik*)

A, Dust separator with sprinkler. B, Silt chamber. C, Tar separator. D, To the engine.

temperature is immediately above the pipe, and tar extraction occurs there. In general, steam is not added in addition to the air, so that a gas with a large percentage of carbon monoxide is obtained. With a generator similar in design to fig. 26, having a shaft diameter of 12·8 ft., the output per hour is 9200 lb. of coke and 700,000 c. ft. of gas, this also being the volume of the injected air. The gas contains

0·1 per cent H_2, 1·2 per cent CH_4,
33 per cent CO, 0·5 per cent CO_2,

with 1·2 oz. of water and 0·3 oz. of sulphur in 100 c. ft. The calorific value of the gas is 127 B.Th.U. per cubic foot. The output for 1 sq. ft. of shaft cross-section is therefore 72 lb. of coke, with 76 c. ft. of gas for 1 lb. of coke.

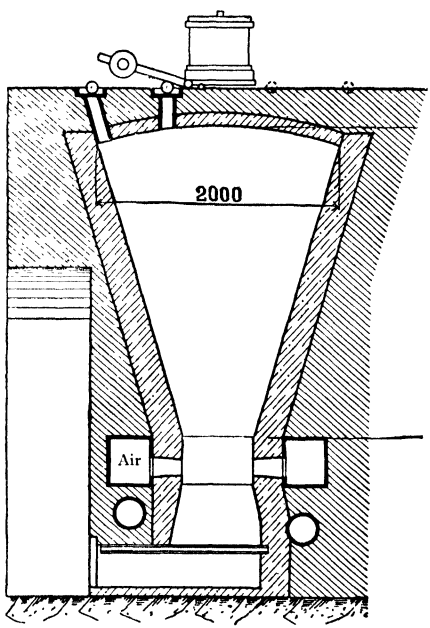


Fig. 25.—Heller-Generator

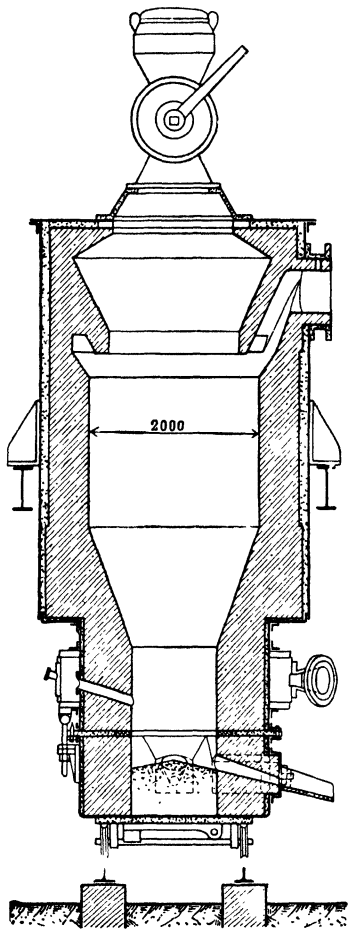


Fig. 26.—Pintsch Tapping Generator

In the gasification of low-percentage fuels with a high water content it should be borne in mind that in 100 c. ft. of gas a large quantity of water from the fuel is vaporized, e.g. with crude brown coal there are 25 oz. of water, while in the gas from mineral coal there are only 6 oz. of water. There is, of course, a limiting value at which it does not pay to gasify fuels containing water. In the same way, it is not economically expedient to gasify

fuels containing a large proportion of ash. As an example of the method of calculating the maximum ash content with which economical production is possible, the following may be considered. It is required that the calorific value of the gas produced shall be at least 125 B.Th.U. per cubic foot. The tar produced will have a thermal value equivalent to $3\frac{1}{2}$ per cent of this, the sensible heat of the gas will be 3 per cent, and the conduction and radiation losses $16\frac{1}{2}$ per cent, so that a total of 154 B.Th.U. for every cubic foot of gas is thus required. Now if 1 lb. of the pure fuel with ash removed has a calorific value of 13,500 B.Th.U. and produces 80 c. ft. of gas, 169 B.Th.U. are available for every cubic foot produced. The difference between 169 and 154 B.Th.U., viz. 15 B.Th.U., must be sufficient to raise both the fuel and its accompanying ash to the reaction temperature, and to vaporize the water. The sensible heat of the water vapour has already been allowed for. Let the fuel contain x per cent of ash, 10 per cent of water, and $(90 - x)$ per cent of combustible substance, the reaction temperature being 1600° F. Then for 1 lb. of fuel, if the specific heat of the ash and combustible substance be 0·2 and 0·3 respectively, while to raise 1 lb. of water to its boiling-point and to vaporize it requires 1100 B.Th.U., we have:

$$\frac{0{\cdot}2 \times x \times 1600}{100} + \frac{0{\cdot}3(90 - x)1600}{100} + \frac{1100 \times 10}{100} = \frac{90 - x}{100} \times 15 \times 80;$$

$$x = 50 \text{ per cent.}$$

If the ash content is greater than 50 per cent, a greater proportion of the gas would need to be burnt to raise the fuel to the required temperature, and the calorific value of the gaseous products would be less than the assumed 125 B.Th.U. per cubic foot.

Fuels containing a large amount of ash, such as oil shales, cannot therefore be used economically for the production of gaseous fuel. Gases distilled from them at a low temperature are just sufficient for the heating, and the net result of the process is the separation of the oil which is distilled with the gases.

10. WATER GAS

Carbon can be transformed into carbon monoxide not only by combustion with atmospheric oxygen, but also by decomposition of steam. It is necessary that the coke should be maintained at a red heat, but the action is endothermic, and hence heat must be supplied from an external source. The coke can be raised to the required temperature by allowing some of it to burn to carbon monoxide or carbon dioxide with atmospheric oxygen, after which steam can be passed over it for a period, and the temperature will gradually fall. The process can then be repeated.

$$\text{At } 1800^\circ \text{ F., } C + H_2O \text{ (steam)} = CO + H_2 - 50{,}400 \text{ B.Th.U.}$$
$$C + O = CO + 50{,}600 \text{ B.Th.U.}$$

By the passage of air and steam together in suitable proportion, as in some

of the plants described, the action may be carried on either with no production or loss of heat, or with just sufficient production to supply the heat losses. By the intermittent method, however, a water gas is obtained during the steam periods which is of higher calorific value as it is not diluted with

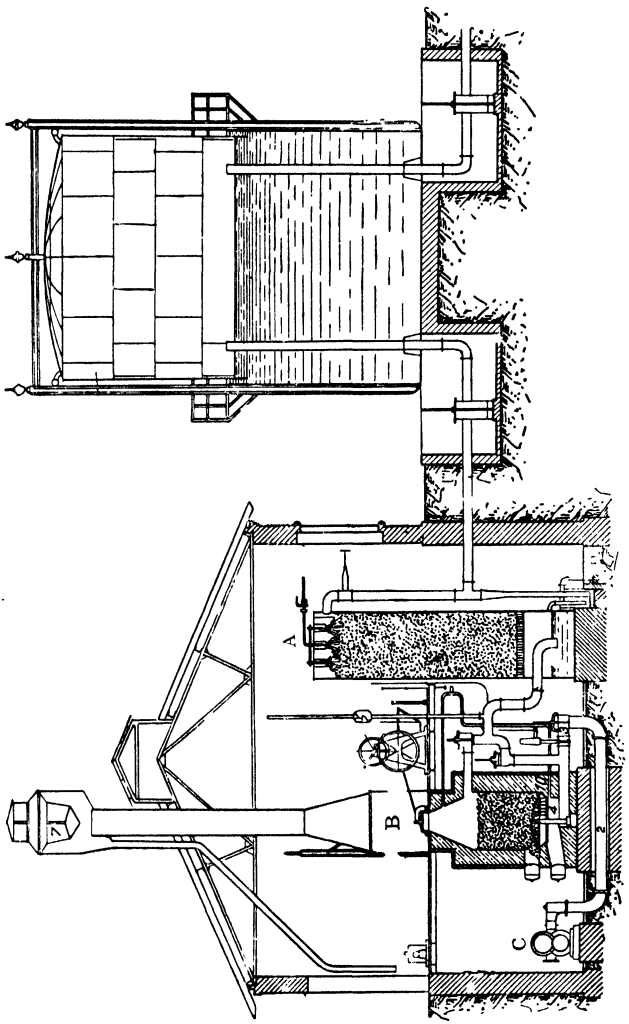


Fig. 27.—Water Gas Generator (*Dellwick-Fleischer*)
A, Washing apparatus. B Generator. C, Blower

atmospheric nitrogen. At lower temperatures, the action of steam on coke results in the formation of carbon dioxide according to the equation:

$$\text{Below } 1250^\circ \text{ F., } C + 2H_2O = CO_2 + 2H_2 - 34{,}000 \text{ B.Th.U.}$$

The composition of the water gas hence depends on the temperature of the process.

The following represent the composition of the gases obtained in one type of water-gas plant.

(1) Air gas during heating period:

8 per cent CO, 17 per cent CO_2, 75 per cent N_2;
30 B.Th.U. per cubic foot.

(2) Water gas during steam period:

45 to 51 per cent H_2, 45 to 41 per cent CO, 1 to 2 per cent CH_4,
2 to 6 per cent CO_2, 7 to 2 per cent N_2;
290 B.Th.U. per cubic foot.

(3) With mixed air and steam:

12 per cent H_2, 28 per cent CO, 5 per cent CO_2, 55 per cent N_2;
125 B.Th.U. per cubic foot.

A design of the Dellwick-Fleischer process is shown in fig. 27. During the heating process it is desirable that the heat should be supplied by the combustion of a small portion of coke to carbon dioxide rather than by the combustion of a larger quantity to carbon monoxide. In the formation of CO_2, 175,000 B.Th.U. are released, while the formation of CO only liberates 50,600 B.Th.U. In the first case the air gas has no calorific value remaining, and can be allowed to escape, while in the second there is considerable calorific value left, but often it is not sufficient to be used economically. In the type of plant shown, 64 c. ft. of air gas of calorific value 95 B.Th.U. per cubic foot and 16 c. ft. of water gas of calorific value 290 B.Th.U. per cubic foot are obtained from 1 lb. of coke. In a later type, the preliminary heating is more efficient. A layer of fuel of small depth is used, and air is supplied at high pressure. The air gas has only 30 B.Th.U. per cubic foot remaining in it, while owing to the smaller amount of carbon being burnt during the heating, 32 c. ft. of water gas can be obtained per pound of coke. The character of the air gas obtained in many plants is shown by the following analyses:

From coke:

0·8 per cent H_2, 25·5 per cent CO, 0·2 per cent CH_4,
4 per cent CO_2, 0·5 per cent O_2, 69·0 per cent N_2.

From mineral coke:

1·9 per cent H_2, 24 per cent CO, 1·9 per cent CH_4,
5·5 per cent CO_2, 62·7 per cent N_2;
84 B.Th.U. per cubic foot.

Water gas has only half the calorific value of coal gas, but much larger quantities are obtained.

Water gas can be " carburetted " with benzol or gas oil obtained from petroleum or brown coal tar. This process consists in saturating the gas with benzol or oil vapour. Carburation with benzol is known as cold combustion. The following table shows the absorption capacity for benzol.

Degrees F.	Vapour Pressure in Inches of Mercury.	Weight of 1 c. ft. of Benzol Vapour in Lb.	Benzol Content of 100 c. ft. of Gas.
20	0·66	0·224	0·49 lb. = 2·2 per cent.
40	1·29	0·215	0·93 ,, = 4·3 ,,
60	2·39	0·206	1·64 ,, = 8·0 ,,
80	4·03	0·198	2·68 ,, = 13·0 ,,

If 20 per cent of water gas of calorific value 290 B.Th.U. per cubic foot is added to coal gas of 560 B.Th.U. per cubic foot, the calorific value of the mixture is 505 B.Th.U. per cubic foot. An addition of 1·3 per cent by volume of benzol vapour, i.e. 0·0029 lb. per cubic foot carrying 50 B.Th.U. will increase the heat value to 555 B.Th.U., approximately the same as that of the undiluted coal gas. Since mineral coal gas contains 1 per cent by volume of benzol, there will be with the above addition a total of 2·2 per cent, an amount which can exist in the form of vapour at all temperatures above 21° F.

Carburation with gas oil is known as " hot carburation ". Water gas which is carburetted hot by gas oil has the following calorific value:

0·94	lb. of oil	per	100 c. ft. of water gas,	420	B.Th.U. per c. ft.
1·25	,,	,,	,,	455	,,
1·87	,,	,,	,,	515	,,
2·50	,,	,,	,,	570	,,
3·12	,,	,,	,,	630	,,

Oil tar is always deposited when the gas is cooled.

In the " double gas " process of Prof. Strache, a combination of distilled coal gas with carburetted water gas is obtained. The air gas resulting from the heating of the coke during the water gas manufacture is used to heat a fireclay grating. The water gas is passed over this grating while oil is being dropped on it and vaporized. The water gas thus saturated with oil vapour contains about 2·5 lb. of oil in 100 c. ft., and has a calorific value between 500 and 560 B.Th.U. per cubic foot. About 30 per cent is heavy hydrocarbons, rich in ethylene, these being more easily absorbed by coal gas than is benzol vapour. The oil gas is added to coal gas, and the resulting mixture has an approximate composition:

48 to 50 per cent H_2, 6 to 7 per cent CH_4,
27 to 30 per cent CO, 6 to 8 per cent CO_2,
8 to 10 per cent $O_2 + N_2$, 2·5 to 3·5 per cent C_nH_m

From 1 lb. of coal, with 34 per cent volatile constituents and of calorific value 13,800 B.Th.U. per pound, 23·5 c. ft. of gas are obtained of 380 B.Th.U. per cubic foot.

11. GASIFICATION OF OIL

Various processes for the gasification of crude oil have recently been evolved. In some of these the oil is made to drop on hot glowing coke. Another method is illustrated in fig. 28. Crude Mexican oil which has been

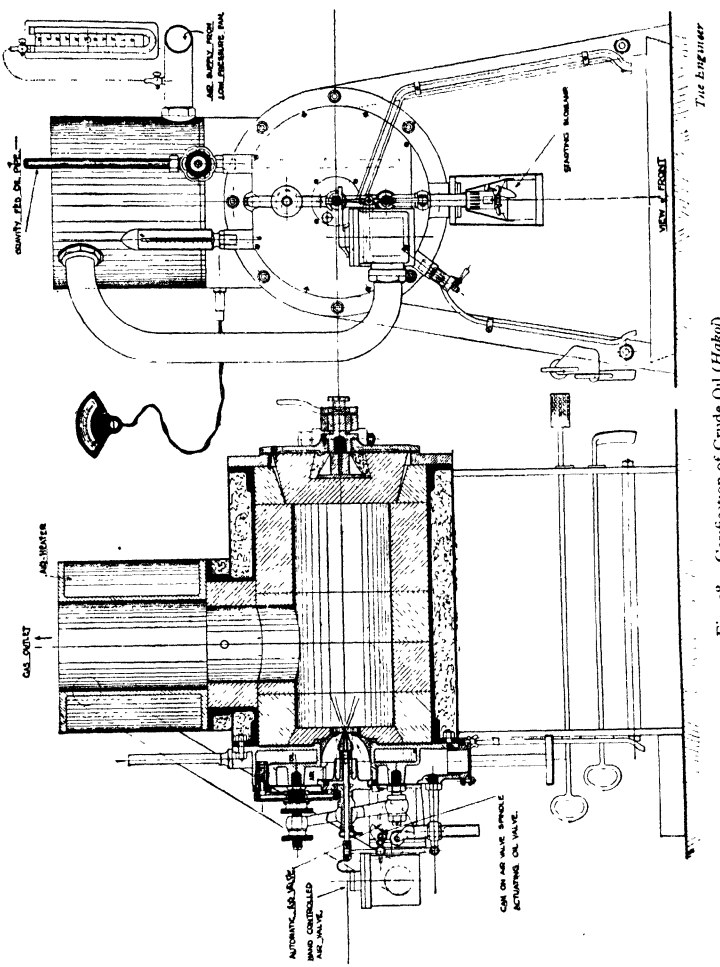


Fig. 28.—Gasification of Crude Oil (*Hakoi*)

pre-heated is injected into a small cylinder with fireclay walls, the jet being such as to reduce the oil to a state of fine atomization. Air which has also been pre-heated is also injected and a portion of the oil is burnt. The flow of oil and air is adjusted so that the temperature is between 1300° F. and 1800° F. Under the action of the heat the greater portion of the oil is transformed

into non-condensable gas of calorific value about 150 B.Th.U. per cubic foot.

12. OTHER COMBUSTIBLE GASES

It has already been mentioned that blast-furnace gas leaves the furnace with considerable calorific value. Its mode of formation is similar to that of producer gas: its calorific value is about 85 B.Th.U. per cubic foot, and its composition:

1 to 2 per cent $H + CH_4$, 24 to 30 per cent CO,
8 to 12 per cent CO_2, 55 to 60 per cent N_2.

The gases leave at about 400° F., while the chemical action at the throat takes place at about 3000° F. The decomposition of iron oxide occurs according to the equation:

$$Fe_2O_3 = 2Fe + 3O - 350{,}000 \text{ B.Th.U.}$$

One pound of iron requires 3150 B.Th.U. for reduction. If this heat is produced by the combustion of carbon and if the proportion of CO_2 to CO obtained is 2 to 1, then 0·41 lb. of C would be required for reducing 1 lb. of Fe. In practice, twice as much carbon is necessary, one half being used for the reduction of the Fe_2O_3 and the other half for creating the heat lost by conduction and radiation.

For the application of small amounts of heat at high temperatures, as required in welding, acetylene is commonly used. It is manufactured from calcium carbide according to the equations:

At 105° F., $CaC_2 + H_2O = C_2H_2 + CaO$.
At 280° F., $CaC_2 + 2H_2O = C_2H_2 + Ca(OH)_2$.

Acetylene should not be subjected to an excessively high pressure nor to too great a heat, as its decomposition is exothermic:

$$C_2H_2 = C_2 + H_2 + 100{,}000 \text{ B.Th.U.}$$

It is liquid at 32° F. and 21·5 atmospheres. The lower calorific value is 21,000 B.Th.U. per pound or 1520 B.Th.U. per cubic foot.

Hydrogen is used both for heating and welding purposes. It burns to water, releasing heat as follows:

$$H_2 + \tfrac{1}{2}O_2 = H_2O \text{ (liquid)} + 123{,}000 \text{ B.Th.U.}$$
$$H_2 + \tfrac{1}{2}O_2 = H_2O \text{ (vapour)} + 103{,}000 \text{ B.Th.U.}$$

The calorific value of hydrogen is thus 344 B.Th.U. per cubic foot if the water is condensed, and 290 B.Th.U. per cubic foot if the water remains as vapour. The temperature attained by the combustion of hydrogen at constant pressure would be 6150° F. if no dissociation occurred, but actually it is very much lower than this.

13. EFFICIENCY

If in the combustion of gas the ratio of air used to gas is x, the maximum temperature attainable is given approximately by the following table:

	With Mineral Coal Gas.	With Crude Brown Coal Gas.	With Brown Coal Briquette Gas.
$x = 1$	4000° F.	3850° F.	3050° F.
$x = 1{\cdot}5$	2750° F.	2850° F.	2350° F.
$x = 2$	2300° F.	2250° F.	1900° F.

In actual practice the following temperatures are reached in gaseous combustion:

Crude gas cooled to 210° F. .. .. .. 2400° F.
Gas with complete extraction of by-products .. 2850° F.

In the direct combustion of solid fuel the flame temperature reaches 2200° F., so that the gaseous combustion is more efficient than that of the more complex solid. As, however, considerable heat is necessarily used in producing the gas, the over-all efficiency is much less. The advantage of gaseous combustion lies in its convenience for the work to be performed. As an extreme example, the gas can be used to work an internal-combustion engine, whereas the solid fuel could not.

The same argument applies to the question of briquetting. A considerable amount of energy is lost in the process, but the briquettes may be more convenient for a particular purpose than the original coal. Approximately 1 ton of brown coal briquettes containing 8800 B.Th.U. per pound is obtained from 2·85 tons of crude brown coal of about 5400 B.Th.U. per pound. Hence only 573 B.Th.U. in the form of briquettes are obtained from 1000 B.Th.U. in the form of brown coal. When brown coal is gasified, 820 B.Th.U. are obtained in the form of gas and tar from 1000 B.Th.U. If the combustion of the gas and the briquettes can be carried out with the same efficiency, the yield from the gas will exceed that from the briquettes in the ratio 820/573 or 1·43. As the gaseous combustion is generally more efficient, the ratio will be greater than this. If the briquettes are gasified before combustion there is a further loss, and assuming the gasification has the same efficiency as with the coal (i.e. 0·82), the ratio of the available heats in the gas from the coal and that from the briquettes will be 1·43/0·82 = 1·74. Yet questions of convenience such as transport may render the intermediate briquetting economically necessary.

CHAPTER V

Utilization of Heat for Heating Purposes

Various types of furnaces for the combustion of fuels were described in Chapter III. In certain cases this involved some consideration of the recipient of the heat with which the fuel interacted. In other cases the furnace formed a separate entity and could be considered separately. In the present chapter the application of the heat will be considered rather than the production, but again the two in some cases are inter-involved.

Heat can be utilized for heating purposes either directly by transmission to the heat-absorbing substance, or indirectly by transmission first to some medium which in turn transfers it to the ultimate recipient. In the latter case there may be a flow of heat only, or there may first be a creation of power and subsequent retransformation into heat. Direct transmission occurs in industrial furnaces for chemical and physical processes, such as:

1. Smelting.
2. Roasting and Calcining.
3. Annealing.
4. " Burning ".
5. Sublimation.
6. Boiling and Evaporating.
7. Drying.

Indirect transmission with air, water, or steam takes place in:

8. Heating of rooms.

Other important cases of heat transfer occur in:

9. Heating of water for steam production.
10. Refrigeration.

1. SMELTING PROCESSES

Examples of these processes include blast and cupola furnaces of various kinds, and also furnaces for the smelting of glass, enamelling, and similar work.

(*a*) Blast Furnaces

The blast-furnace process consists of a smelting process complicated by chemical transformations and physical variations. The ore is smelted with

an addition of limestone which acts as a flux for the silica, a slag being formed by their combination. In some methods of treatment the ore undergoes a preliminary roasting or calcining. Calculations with reference to three crude ores and one roasted ore will be given as examples of the method of working. Representative analyses of the ores and the flux may be taken as follows:

	Roasted Spathic Ore.	Crude Spathic Ironstone.	Red Ironstone.	Brown Ironstone.	Limestone.
Fe_2O_3 ..	67·35	—	75·29	63·11	—
FeO ..	—	42·98	0·60	0·41	0·32
Mn_3O_4 ..	13·02	—	—	—	—
MnO ..	—	8·74	0·18	1·50	—
CaO ..	2·74	0·85	2·25	2·45	54·16
MgO ..	3·19	2·73	0·40	0·80	0·72
Al_2O_3 ..	2·58	0·69	2·73	4·72	1·45
SiO_2 ..	10·77	9·30	15·05	17·38	0·70
CO_2 ..	—	34·27	1·78	2·48	43·20
P_2O_5 ..	—	—	0·32	1·51	—
Cu ..	0·05 = 0·06 CuO	0·20 = 0·25 CuO	—	—	—
H_2O chem.	—	—	1·40	5·14	—
	= 100	= 100	= 100	= 100	= 100
S	0·35	0·36	—	0·01	—
H_2O mech.	9·00	0·06	9·65	13·90	0·47
Fe ..	47·10	33·41	52·67	44·45	0·25
Mn ..	9·44	6·77	0·14	1·16	—

The calculation of the coke required for the smelting and of the amount of limestone can be made in the following manner. It can be assumed that:

Amount of iron reduced	=	97 per cent
Amount of manganese reduced	=	55 ,,
Amount of copper reduced	=	100 ,,
Amount of phosphorus reduced	=	100 ,,

The smelting of 100 lb. of ore thus produces in pounds:

Type of Ore.	Iron.	Manganese.	Copper.	Phosphorus from Ore.	Phosphorus from Coke.	Total.
Roasted spathic ore ..	45·68	5·19	0·05	—	0·30	51·22
Crude spathic ironstone	32·40	3·72	0·20	—	0·30	36·62
Red ironstone	52·67	0·07	—	0·14	0·30	53·20
Brown ironstone ..	43·11	0·63	—	0·60	0·30	44·64

In addition to these quantities of pure metal, the yield contains approximately the following impurities:

	Sulphur.	Silicon.	Carbon.	Total.
	Per Cent.	Per Cent.	Per Cent.	Per Cent.
From roasted spathic ore ..	0·05	1	3·5	4·55
From crude spathic ironstone	0·05	1	3·5	5·55
From red ironstone	0·1	2	3·0	5·10
From brown ironstone ..	0·1	2	3·0	5·10

The total yield of metal is therefore, from spathic ironstone,

$$\frac{\text{sum of reduced elements} \times 100}{95{\cdot}45};$$

from red and brown ironstone,

$$\frac{\text{sum of reduced elements} \times 100}{94{\cdot}9}.$$

The amount of dried ore required to produce a yield of 100 lb. of mixed metal and the weight of the different constituents of the ore is thus obtained.

	Roasted Spathic Ore.	Crude Spathic Ironstone.	Red Ironstone.	Brown Ironstone.
	Lb.	Lb.	Lb.	Lb.
Yield	100	100	100	100
Ore required ..	186·50	260·60	178·40	212·60
Yield per cent ..	53·66	38·37	56·05	47·03
Fe_2O_3	125·6	—	134·3	134·2
FeO	—	112·0	1·1	0·9
Mn_3O_4	24·3	—	—	—
MnO	—	22·8	0·3	3·2
CaO	5·1	2·2	4·0	5·2
MgO	5·9	8·1	0·7	1·7
Al_2O_3	4·8	1·8	4·9	10·0
SiO_2	20·1	24·2	26·8	36·9
CO_2	—	89·2	3·2	5·3
P_2O_5	—	—	0·6	3·2
CuO	0·1	0·6	—	—
H_2O chem. ..	—	—	2·5	10·9

The amount of limestone to be added is calculated from the amount of silica present, using the ratio $SiO_2/CaO = 12/11$. The coke, however, contains 3·5 per cent SiO_2 and 0·5 per cent CaO, while the ore itself also contains a quantity of CaO, and these amounts must be allowed for in the calculation. Part of the SiO_2 remains in the iron, part forms dust, and the

remainder becomes slag. The amount of dust in the production of 100 lb. of metal is about 2·25 lb., of which 7 per cent is SiO_2 and 3 per cent CaO, i.e. 0·16 lb. and 0·06 lb. respectively. The limestone required is as follows:

Ore.	SiO_2 in Ore and Coke.	CaO required.	CaO in Ore and Coke.	CaO to be added.	Limestone to be added.	$\frac{\text{Limestone}}{\text{Ore}}$.
						Per cent.
Roasted ore	21·6	19·7	5·1	14·6	26·9	14·4
Crude spathic ironstone	25·7	23·5	2·2	21·3	39·3	15·0
Red ironstone	26·4	24·2	4·4	19·8	36·5	20·4
Brown ironstone ..	36·5	33·4	5·2	28·2	52·1	24·5

The gas yield per 100 lb. of metal produced averages 7200 c. ft., of which 58 per cent is nitrogen. The amount of air required is therefore $7200 \times 58/79 = 5300$ c. ft. The exhaust gases contain about 0·56 lb. of moisture per 1000 c. ft., giving a total of 3 lb. per 100 lb. of metal. The amount of moisture in the ore and lime may be very much larger than this, but it has been decomposed during the heat reactions. The tabulated figures of the ore required for a yield of 100 lb. refer to dried ore, and should be increased owing to the percentage of water content, as actually the ores are charged in a wet condition. With ores as given in the first table (p. 164) the ratio of this increase should be:

with roasted spathic ironstone, $\frac{100}{100 - 9{\cdot}0}$;

with crude spathic ironstone, $\frac{100}{100 - 0{\cdot}06}$;

with red ironstone, $\frac{100}{100 - 9{\cdot}65}$;

with brown ironstone, $\frac{100}{100 - 13{\cdot}9}$.

The moisture originally present is then given by:

	Moisture in the Ore.	Moisture in the Lime.	Total Moisture.	Hydrated Water to be expelled.
	Lb.	Lb.	Lb.	Lb.
Roasted spathic ore ..	18·40	0·12	18·52	—
Crude spathic ironstone	0·15	0·19	0·34	—
Red ironstone	19·74	0·21	19·98	2·50
Brown ironstone ..	29·55	0·24	29·79	10·93

The calculation of the heat involved in the smelting of the ore can be divided into several sections, the most important being the heat required for the reduction of the compounds from the oxides to the metal. In the pro-

duction of 100 lb. of metal the amount of the reduction heat is as follows. The quantity of each oxide in the ore has been given in preceding tables but only a small quantity of the silica is reduced.

Sub-stance.	Reduction Heat.	Roasted Spathic Ore.		Crude Spathic Ironstone.		Red Ironstone.		Brown Ironstone.	
	B.Th.U./Lb.	Lb.	B.Th.U.	Lb.	B.Th.U.	Lb.	B.Th.U.	Lb.	B.Th.U.
Fe_2O_3	2270	125·6	285,000	—	—	134·3	303,000	134·2	303,000
FeO	1900	—	—	112·0	213,000	1·1	2,000	0·9	2,000
Mn_3O_4	2560	24·3	62,000	—	—	—	—	—	—
MnO	2420	—	—	22·8	55,000	0·3	1,000	3·2	8,000
SiO_2	6630	2·0	13,000	2·0	13,000	4·0	26,000	4·0	26,000
CuO	460	0·1	—	0·6	—	—	—	—	—
P_2O_5	4920	—	—	—	—	0·6	3,000	3·2	16,000
H_2O	5570	3·0	17,000	3·0	17,000	2·5	14,000	3·0	17,000
			377,000		298,000		349,000		372,000

The sensible heat of the 100 lb. yield of metal will be approximately 50,000 B.Th.U. The amount of slag produced in the four cases is about twice that of the lime, i.e. 40 to 67 lb., and its sensible heat of 800 B.Th.U. per pound will range from 32,000 B.Th.U. to 53,000 B.Th.U. The exhaust gases of 7200 c. ft. at 300° F. contain 29,000 B.Th.U. as sensible heat in addition to their calorific value. The heat contained in the dust can be neglected.

To evaporate 1 lb. of water and raise the temperature of the steam to 300° F. requires 1160 B.Th.U. in sensible and latent heat. Hence the heat contained in the water from the roasted spathic ore is $9 \times \frac{100}{91} \times 1{\cdot}865 \times 1160 = 21{,}000$ B.Th.U., and similarly for the other cases. The heat to separate and vaporize the water in chemical combination is negligible. Heat is required to separate the carbon dioxide from the lime in the limestone and also to separate that in the ores. The combination, however, of the lime with the silica in forming the slag releases heat, resulting in a net heat consumption as below with respect to the limestone.

	Weight of Limestone.	Weight of CO_2.	Heat Consumption, B.Th.U. per Lb. of CO_2.	Heat, B.Th.U.
	Lb.	Lb.		
With roasted spathic ore ..	26·9	11·8	1830	22,000
With crude spathic ironstone	39·3	17·3	1830	31,000
With red ironstone	36·5	16·1	1830	29,000
With brown ironstone ..	52·1	23·0	1830	42,000

Except in the case of crude spathic iron ore, the amount of carbonates in the ore is small and the heat required to separate the carbon dioxide

can be neglected. With spathic iron ore there are carbonates of iron, magnesium, and manganese, and the net heat consumption required can be obtained as follows:

	Weight of CO_2.	Heat Consumption, B.Th.U. per Lb. of CO_2.	Heat, B.Th.U.
	Lb.		
$MgCO_3$ ..	9·1	1310	12,000
$FeCO_3$ ..	70·0	620	43,000
$MnCO_3$ ..	10·1	620	6,000
	89·2		61,000

Neglecting for the present the calorific value of the exhaust gases, which with ideal combustion would be zero (the gases containing carbon dioxide and water only) and which in other cases may be recoverable, the total heat consumption for a production of 100 lb. of metal from the different ores would be:

	Roasted Spathic Iron Ore.	Crude Spathic Ironstone.	Red Ironstone.	Brown Ironstone.
	B.Th.U.	B.Th.U.	B.Th.U.	B.Th.U.
Reduction	377,000	298,000	349,000	372,000
Iron	50,000	50,000	50,000	50,000
Slag	32,000	37,000	38,000	53,000
Gas	29,000	29,000	29,000	29,000
Water vapour	21,000	—	23,000	35,000
CO_2 separation ..	22,000	92,000	29,000	42,000
	531,000	506,000	518,000	581,000
Radiation and conduction losses, 10 per cent	53,000	51,000	52,000	58,000
Total heat requirements	584,000	557,000	570,000	639,000

The total heat requirements thus estimated must be supplied by the combustion of the fuel, the heat of the blast, or by other means. Now the reduction process itself requires the presence of carbon or carbon monoxide, which is converted to carbon dioxide, thus liberating a considerable amount of heat. This supplies a large proportion of the heat required for the decomposition of the metallic oxides. The remainder and the other heat requirements must be supplied by the combustion of carbon, which does not take part in the chemical reactions, and by other sources.

As an example, the reduction of ferric oxide may be represented by the equation:

$$Fe_2O_3 + C + CO = 2CO_2 + Fe_2.$$

lb. 160 12 28 88 112

Thus 1 lb. of Fe_2O_3 requires 0·075 lb. of C and 0·175 lb. of CO, and the formation of CO_2 from them produces 1870 B.Th.U. The decomposition of 1 lb. of Fe_2O_3, i.e. the reduction heat, requires 2270 B.Th.U., the figure used previously in the estimation of the heat requirements. The reaction represented in the equation could, of course, have been considered as a whole with a net heat requirement of 400 B.Th.U. per pound of Fe_2O_3, instead of calculating the gross heat production and gross heat requirements for the two sides of the heat balance. The gross heat production of the reactions for the various oxides in the ores is found to be:

Ore.	Oxide to be Reduced.		Substances Required.		Substances Formed.		Gross Heat Production, B.Th.U.
		Lb.	C.	CO.	CO.	CO_2.	
Roasted spathic iron ore ..	Fe_2O_3	125·6	9·48	22·14	—	69·57	235,000
	Mn_3O_4	24·3	2·55	—	—	9·38	37,000
	SiO_2	2·0	0·7	—	1·7	—	3,000
	CuO	0·1	—	0·04	—	0·05	—
	H_2O	3·0	1·9	—	4·5	—	8,000
			14·6	22·2	6·2	79·0	283,000
Crude spathic ironstone ..	FeO	112·0	9·29	—	—	34·1	136,000
	MnO	22·8	1·94	—	—	7·1	28,000
	SiO_2	2·0	0·7	—	1·7	—	3,000
	CuO	0·6	—	0·2	—	0·3	—
	H_2O	3·0	1·9	—	4·5	—	8,000
			13·8	0·2	6·2	41·5	175,000
Red ironstone ..	Fe_2O_3	134·3	10·1	23·5	—	73·85	252,000
	FeO	1·1	0·09	—	—	0·33	1,000
	MnO	0·3	0·03	—	—	0·09	—
	SiO_2	4·0	1·4	—	3·4	—	7,000
	P_2O_5	0·6	0·25	—	0·6	—	1,000
	H_2O	2·5	1·66	—	3·9	—	7,000
			13·5	23·5	7·9	74·3	268,000
Brown ironstone	Fe_2O_3	134·2	10·1	23·5	—	73·85	252,000
	FeO	0·9	0·07	—	—	0·25	1,000
	MnO	3·2	0·27	—	—	1·0	4,000
	SiO_2	4·0	1·4	—	3·4	—	7,000
	P_2O_5	3·2	1·35	—	3·2	—	6,000
	H_2O	3·0	1·9	—	4·5	—	8,000
			15·1	23·5	11·1	75·1	278,000

The total carbon required as carbon and in carbon monoxide for the reactions resulting in a yield of 100 lb. of metal is:

With roasted spathic iron ore, ... 24·1 lb.
With crude spathic ironstone, ... 14·0 lb.
With red ironstone, ... 23·5 lb.
With brown ironstone, ... 25·2 lb.

The difference between the total heat requirements and the heat production of the reducing reactions is thus in B.Th.U.:

	Requirements.	Production.	Difference.
With roasted spathic ore ..	584,000	283,000	301,000
With crude spathic ironstone	557,000	175,000	382,000
With red ironstone	570,000	268,000	302,000
With brown ironstone... ..	639,000	278,000	361,000

The heat would be supplied in the most efficient way if additional carbon were burnt to carbon dioxide. The temperature conditions preclude this and a mixture of carbon monoxide and carbon dioxide is formed. Taking the least efficient case with the additional carbon burnt to carbon monoxide only, 1 lb. of carbon is required to produce 3250 B.Th.U. With a small proportion of carbon dioxide and methane, 1 lb. of carbon may produce about 5000 B.Th.U. Using this value, the total quantity of carbon to be supplied for the production of 100 lb. of metal for reduction, for extra heat, and for the carbon retained as impurity in the metal is given in the following table. The amounts of coke are calculated by assuming an 85 per cent carbon content.

	Reduction.	Carbon Required.		Total.	Coke Required.
		Extra Heat.	In Metal.		
With roasted spathic ore ..	24·1	60·2	3·5	87·8	103
With crude spathic ironstone	14·0	76·4	3·5	93·9	110
With red ironstone	23·5	62·4	3·0	88·9	104
With brown ironstone ..	25·2	72·2	3·0	100·4	118

As stated in the preceding paragraph, the constitution of the gas produced and hence the liberation of heat depend upon the temperature conditions. According to the empirical formula of Brisker, if T_1 and T_2 are the absolute temperatures of the air and gas, the heat liberated per pound of carbon (taken above as 5000 B.Th.U.) is equal to $a - b$ where

$$a = 4450 + 5{\cdot}79\ C_p T_1,$$
$$b = 6{\cdot}79\ C_p T_2,$$
$$C_p = \text{specific heat at constant pressure} = 0{\cdot}24.$$

If the flue temperature = 300° F. and the injected air temperature is 1050° F.,

$a - b = 5300$ B.Th.U. and the amounts of coke required would be somewhat smaller than those tabulated.

The general comparison which has been made between the different ores is affected somewhat by their physical condition, e.g. their density. The effect of CO on dense ores, such as red iron ore, is not so great as on roasted spar, which has been made porous by the expulsion of the carbon dioxide from the crude ore. The same is true of the slag, which, if porous, is likewise more accessible for reduction.

In considering the total heat balance of the blast furnace, the sensible heat and calorific value of the blast-furnace gases should not be taken as a total loss, as much of it may be utilized in the regenerator or to heat boilers or work gas-engines. The production of 100 lb. of metal is accompanied by the production of 7200 c. ft. of gas, whose sensible heat has already been given as 29,000 B.Th.U. The calorific value of a blast-furnace gas with the composition assumed, viz. 58 per cent N_2, 14 per cent CO_2, 26 per cent CO, and 2 per cent $H_2 + C_nH_m$, is 57 B.Th.U. per cubic foot at 300° F., so that there is from this source an available heat of 410,000 B.Th.U.

The carbon requirements and hence the efficiency of the blast furnace have been shown to depend upon the temperature. It may further be shown that there is a definite temperature range above and below which the process will not work. If the heat of combustion of a fuel is h, the mass of the products of combustion m, and their specific heat is C_p, then the maximum attainable temperature without heat loss is

$$t = \frac{h}{C_p m}.$$

According to Le Chatelier the mean value of C_p between 0° and 8000° F. is 0·5 for CO_2 and between 0° and 4000° F. is 0·28 for air. If 1 lb. of coke is burnt with oxygen to CO_2, $h = 14{,}500$ B.Th.U. and $m = 3\frac{2}{3}$, so that

$$t = \frac{14{,}500}{3\frac{2}{3} \times 0{\cdot}5} = 7900° \text{ F.}$$

If air is used containing 21 per cent O_2 and 79 per cent N_2, the specific heat being smaller at lower temperatures,

$$t = \frac{14{,}500}{3\frac{2}{3} \times 0{\cdot}32 + 9{\cdot}3 \times 0{\cdot}28} = 3800° \text{ F.}$$

If there is an excess of air of 25 per cent, then

$$t = \frac{14{,}500}{3\frac{2}{3} \times 0{\cdot}32 + 9{\cdot}3 \times 0{\cdot}28 + 3 \times 0{\cdot}28} = 3100° \text{ F.}$$

Such an excess of air is necessary where the velocity is so high that combustion of all the oxygen cannot be attained in the time of contact. In the blast furnace, however, carbon is being burnt in the presence of an excess

of carbon, and under these conditions carbon monoxide is formed in proportions dependent upon the temperature. This is given in the following table.

t, °F.	CO, per cent.	CO_2, per cent.
840	2·0	98·0
1110	23·0	77·0
1290	68·0	32·0
1570	94·0	6·0
1830	99·3	0·7
1920	99·6	0·4
2010	99·8	0·2

It will be seen that at any temperature above 800° F. only part of the consumed carbon is burnt to CO_2 and hence the heat produced per pound of coke is less than 14,500 B.Th.U. To produce the large amounts of heat necessary for the reduction of the iron oxides, combustion at low temperatures with a large production of dioxide is desirable. On the other hand, the actual reduction is brought about by the action of carbon monoxide on the iron oxides, and the production of carbon monoxide is greater at the higher temperatures. The reactions

$$Fe_3O_4 + CO \rightleftharpoons 3FeO + CO_2$$
$$FeO + CO \rightleftharpoons Fe + CO_2$$

are also reversible, depending upon the temperature of the furnace, and hence the CO_2 should be quickly removed to avoid a return to FeO at places of lower temperature. The equilibrium of these reversible reactions is determined by the concentration of the different substances and the pressure to which the system is subject, as well as by the temperature. From the law of mass action the ratio of the product of the concentration of the disappearing masses to the product of the resulting masses is constant, i.e. for $CO_2 + C \rightleftharpoons 2CO$

$$\frac{\text{concent. CO} \cdot \text{concent. CO}}{\text{concent. } CO_2 \cdot \text{concent. C}} = \text{constant.}$$

If H is the heat production of the reaction, T the absolute temperature, R the gas constant in the same units as H, and k the equilibrium constant, then from Van't Hoff,

$$\frac{d \log_e k}{dT} = \frac{H}{RT^2}.$$

Taking $k = \frac{[CO]^2}{[CO_2]}$, as [C] is constant, then

$$\log_e \frac{k_1}{k_2} = \frac{H}{R}\left(\frac{1}{T_2} - \frac{1}{T_1}\right);$$

k thus increases with the temperature.

The data relative to the equilibrium of oxides of iron with CO and CO_2 are given in the following tables:

Fe_3O_4, CO, and CO_2

t, °F.	CO, per cent.	CO_2, per cent.	$k = \frac{[CO]^2}{[CO_2]}$
840	46	54	0·812
915	47	53	0·882
1020	44	56	0·786
1200	37	63	0·587
1560	26	74	0·351
1740	23	77	0·299
	FeO, CO, CO_2		
1025	54	46	1·160
1100	56	44	1·250
1200	58	42	1·380
1220	58·4	41·6	1·400
1250	59	41	1·440
1380	61	39	1·560
1560	68	32	2·120
1650	71·5	28·5	2·510

As the equations show that the reaction involves no change in volume of the gas phase, the constant is independent of pressure. These equilibrium figures are represented in fig. 1, an examination of which reveals the following:

All reactions take place on the right-hand side of the S-shaped curve for the equilibrium values $CO_2 + C \rightleftharpoons 2CO$.

To the left of this curve the CO is not stable.

Fe, FeO, and Fe_3O_4 are present on the right-hand side of the S-shaped curve.

Fe, FeO, C, CO, and CO_2 are co-existent at the point of intersection with the S-shaped curve at 1270° F.

FeO, Fe_3O_4, C, CO, and CO_2 are co-existent at 1200° F., the point of intersection with the S-shaped curve.

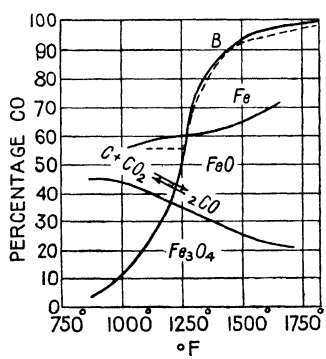


Fig. 1.—Equilibrium Diagram of the Reactions

In the hottest zone the reactions $FeO + CO \rightleftharpoons Fe + CO_2$ and $C + CO_2 \rightleftharpoons 2CO$ give rise to interesting conditions. If FeO were absent the gas curve would coincide with the S-shaped curve. CO, however, reacts with FeO; the equilibrium is thus disturbed, while the resulting CO_2 is reduced to CO at this temperature and affects the new FeO, so that a metallic Fe is gradually produced. Hence the actual gas curve resulting from the presence of all the substances together is lower than the S-shaped curve, for on account of the FeO more CO_2 is obtained at high temperatures than corresponds to

the S equilibrium. The dotted line gives the approximate gas curve. At about 1250° F. the reduction in the reaction velocity of FeO + CO ⇌ Fe + CO_2 brings the actual curve to agree with the S-shaped curve for C, CO, and CO_2 only. Below 1250° F. the reaction velocity is small when compared to the velocity of the rising gas. The gases rise to the flue without their formation being appreciably altered. There is not sufficient time for them to reach the equilibrium values with the oxides.

The transformation of the CO as it rises from the floor of the furnace is as follows. It first comes into contact with the finely diffused FeO, resulting in the production of CO_2. In the layer from 2500° F. to 2000° F. the CO_2 is reduced to CO but in the zone about 1500° F. some CO_2 is reformed. If the mixture remains for some time in contact with the coal at this temperature 6 per cent of CO_2 to 94 per cent of CO would be formed. The presence of metallic oxides has an accelerating effect, but the upward velocity of the gases tends to reduce the amount of CO_2 owing to lack of time required for equilibrium.

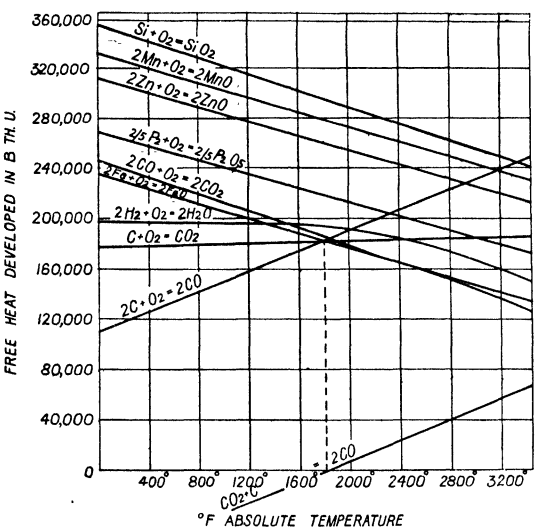


Fig. 2.—Maximum Working Capacity of Important Oxidation Reactions in Relation to the Absolute Temperature based on 1 Molecule of Oxygen

In addition to the blast furnace for the treatment of iron, similar furnaces are in use in foundries for metals such as mercury, lead, and copper. The calculations for the estimation of the required coke are made as for the iron blast furnace. The blast-furnace gases are rarely utilized, although they may have considerable calorific value. The heat liberated in certain of the reactions involved is shown in fig. 2. This useful heat is connected with the total reaction heat and equilibrium constant at the same temperature by the equations

$$H_1 = RT \log_e k,$$

$$H = RT^2 \frac{d \log_e k}{dT};$$

so that

$$\log_e k = \frac{H_1}{RT} = \int \frac{H}{RT^2}\, dT,$$

$$H_1 - H = T\frac{dH_1}{dT}.$$

Thus the difference between the free or useful heat and the reaction heat increases with rising temperature.

(*b*) Cupola Furnaces

In the cupola furnace the layer of coke is burnt to carbon dioxide by means of the injected air, the hot combustion gases then enabling the pig iron to be smelted. So much air is injected that the waste gases consist almost exclusively of CO_2. The entire combustion heat of the coke can thus be utilized. The requirements of the smelting process are calculated in a similar way to those of the blast furnace. The efficiency of such furnaces has already been examined (p. 54). It ranges from 10 per cent to 25 per cent.

Data relating to the dimensions, output, and consumption of cupola furnaces are given below.

Diameter in Feet.	2.	4.	5.
Smelting zone cross-section in square feet	2·95	12·2	19·0
Height, feet	13·1	21·3	22·9
Output, pounds per hour	4400	17,500	27,400
Coke consumption, percentage	6	9·5	11
Coke consumption, pounds per hour ..	260	1700	3000
Air, cubic feet per pound of coke ..	15·5	17·5	19
Air, cubic feet per minute	640	4200	9700
Air pressure, inches	14	29	40
Nozzles' cross-section, square inches ..	132	525	820
Number of nozzles	6	12	12

(*c*) Siemens-Martin Furnaces and Large Reverberatory Furnaces

The heat balance of the Siemens-Martin furnaces shows that from a heat-engineering point of view these furnaces still require improvement. Most attention has been given to the utilization of the waste gases, in spite of the fact that heat radiation gives rise to still greater losses. The present-day problem of the engineer is still how to diminish these losses. The abundant literature on this type of furnace shows that this is easily the one most used in industry, but that in spite of this it still presents numerous problems which are unsolved. Present designs are not efficient in producing an economical utilization of the fuel, and study of the composition of the waste gases should lead to improvement.

In expressing the result of a test upon a furnace, efficiencies can be stated in various forms. The whole energy involved in a process may be used as a basis or the net energy after allowing for certain recoverable portions. The efficiencies of the whole or parts of a process may be obtained according to the purpose of the test.

Let H_1 be the heat given to the object of the process, i.e. the useful work;
H_2 the heat given to other bodies, but recoverable;
H_3 the heat totally lost;
η_1 the thermal efficiency; and
η_2 the thermodynamic efficiency.

$$\text{Then } \eta_1 = \frac{\text{heat transformed into useful work}}{\text{total heat supplied}} = \frac{H_1}{H_1 + H_2 + H_3} = \frac{H_1}{H},$$

$$\eta_2 = \frac{\text{heat transformed into useful work}}{\text{heat used in the process}} = \frac{H_1}{H_1 + H_3} = \frac{H_1}{H - H_2}.$$

Applying this to a test of a Siemens-Martin furnace, the amounts of heat in the various products and reactions being calculated as for the blast furnace:

For the hearth.

Let H_a be the heat in the charge;
H_b the heat through gas combustion;
H_c the heat in waste gases;
H_d the heat of exothermal reactions;
H_e the heat of endothermal reactions;
H_f the heat through radiation and conduction;
H_g the heat in furnace residue; and
H_x the heat in material when taken out of furnace.

$$\eta_1 = \frac{H_x}{H_a + H_b},$$

$$\eta_2 = \frac{H_x}{H_a + H_b - H_c},$$

$$H_x = (H_a + H_b + H_d) - (H_c + H_e + H_f + H_g).$$

For the whole furnace.

Let H_p be the heat for warming the gas;
H_q the heat for warming the air; and
H_r the heat in waste gases at end of furnace.

$$\eta_1 = \frac{H_x}{H_a + H_b},$$

$$\eta_2 = \frac{H_x}{H_a + H_b + H_p + H_q - H_r}.$$

The foundry reverberatory furnace is a smelting furnace for casting pig iron, in which the hearth is similar to that in the Martin furnace and which is usually designed with the flame direction constant. Most of the usual fuels can be used, including those which are used in combination with others. From a heat-engineering point of view this furnace in particular has been badly neglected and is greatly in need of improvement. The calculation of

the efficiency of this type of furnace is similar to that for a Martin furnace. To this category also belong other metal-smelting furnaces of the reverberatory type.

(*d*) Glass-smelting Furnaces

The old glass-smelting furnaces heated directly by coal fires underneath have been superseded in modern plants by furnaces heated with producer gas. The greater ease of temperature regulation leads to greater efficiency, while glass of greater purity can be obtained. The fuel consumption per pound of crude glass even now is found to vary within wide limits for the same kind of glass, indicating that the process is still inefficient. The manufacture of the glass is carried out either intermittently in pots or continuously in tanks. The furnaces are fed from gas generators having inclined or circular grates and regenerators. The consideration of the plant and its efficiency may be divided into two parts:

1. The generator and regenerator.
2. The glass furnace.

1. Mineral coal, crude brown coal, or brown coal briquettes are gasified in the generators, which with few exceptions have inclined or circular grates (fixed or revolving). In some cases flat grates are used instead of inclined grates. The utilization of mineral coal produces a gas of higher calorific value than that from brown coal. Thus higher temperatures can also be obtained. This distinguishes the Rhenish method of blowing plate glass at a high temperature (long cylinders with a small diameter) and the German method (shorter cylinder with a larger diameter). The gas formation should remain constant, particularly during the blowing or casting process. If generators are loaded at long intervals of time it is found that luminous hydrocarbons are formed after a charge of fresh fuel has been inserted, while subsequently the formation of carbon monoxide and hydrogen predominates. With gas generators, therefore, the fuel should be charged at short intervals in order to obtain a constant gas.

The following illustrates the performance of some gas generators in use with glass-smelting plants.

(*a*) Inclined grate using mineral coal of 13,200 B.Th.U. per pound and composition:

77·65 per cent C, 4·53 per cent H_2, 10·0 per cent O_2, 1·05 per cent N_2, 3·53 per cent water, 4·16 per cent ash.

Yield, 4·95 lb. of gas per pound of coal, its calorific value being 1590 B.Th.U. per pound and volume percentage,

22·9 per cent CO, 7·4 per cent H_2, 1·9 per cent CH_4, 6·1 per cent CO_2, 61·7 per cent N_2.

Including the sensible heat at 1300° F., the available heat from 1 lb. of coal was 9500 B.Th.U. The slag amounted to 2·4 per cent of the fuel (5·2 per

cent of the slag being carbon), the ash 1·6 per cent (of which 29·5 per cent was C), the fine clinker 2·3 per cent (32·4 per cent C), and the clinker 10·5 per cent (53·8 per cent C).

(*b*) The same coal in a circular revolving grate yielded 4·78 lb. of gas per pound of coal, the calorific value being 2120 B.Th.U. per pound, and the composition

29·6 per cent CO, 8·5 per cent H_2, 2·3 per cent CH_4, 3·1 per cent CO_2, 56·5 per cent N_2.

Including the sensible heat, the available heat amounted to 11,500 B.Th.U. per pound. The total slag and ash was 6·04 per cent (of which 4·7 per cent was carbon). Little increase in efficiency of inclined grates can be expected from improved methods of charging, and the advantages of the revolving grate with its higher efficiency are obvious.

(*c*) Brown coal briquettes of 8800 B.Th.U. per pound in an inclined grate generator produced 45 to 51 c. ft. per pound of fuel, its calorific value being 152 to 164 B.Th.U. per cubic foot, and its composition

27 to 29 per cent CO, 10 to 13 per cent H_2, 2 to 3 per cent CH_4, 5 to 6 per cent CO_2, 51 to 54 per cent N_2.

The ash varies from 5 to 8 per cent. On a circular grate, briquettes gave a smaller yield by 2 to 3 per cent, while a revolving grate produced a small increase in yield.

(*d*) Bohemian brown coal containing about 20 per cent of water and 3 to 7 per cent sulphur gave no perceptible difference in yield from both types of grate. Fuel with higher water content can be used but the economic value of the process is doubtful. A considerable proportion of the calorific value is used in vaporizing the water, and the latent heat is usually lost when the water is separated out by condensation in long pipes. There is, of course, a possibility of utilizing it in reheating ovens and for other purposes.

The difference between generators for crude brown coal and for briquettes can be illustrated by reference to fig. 7, Chapter IV. For crude brown coal, $ab = 2{\cdot}3$ ft, $bc = 2{\cdot}3$ ft., while $ab = 3{\cdot}3$ to 4 ft., $bc = 1{\cdot}95$ ft. for briquettes. With crude coal, the fuel is more closely packed than with the four-cornered briquettes. In the latter case, therefore, there must be a greater length of resistance to the passage of the air in order to produce combustion in the lower layer, and in the adjacent reducing layer to reduce the CO_2 as completely as possible to CO. Thus with briquettes the height of the charge should be increased and that of the grate diminished as compared to the heights with crude coal. A middle course can be taken by the use of brown coal mixed with briquettes. In order to obtain a uniform burning process, however, great care must be taken to ensure a thorough mixing. It is most important to keep the fuel charge at the correct height. If the latter is too low the carbon dioxide produced in the lowest layer cannot be reduced to carbon monoxide, and the gas has little calorific value when it enters the furnace. If the charge

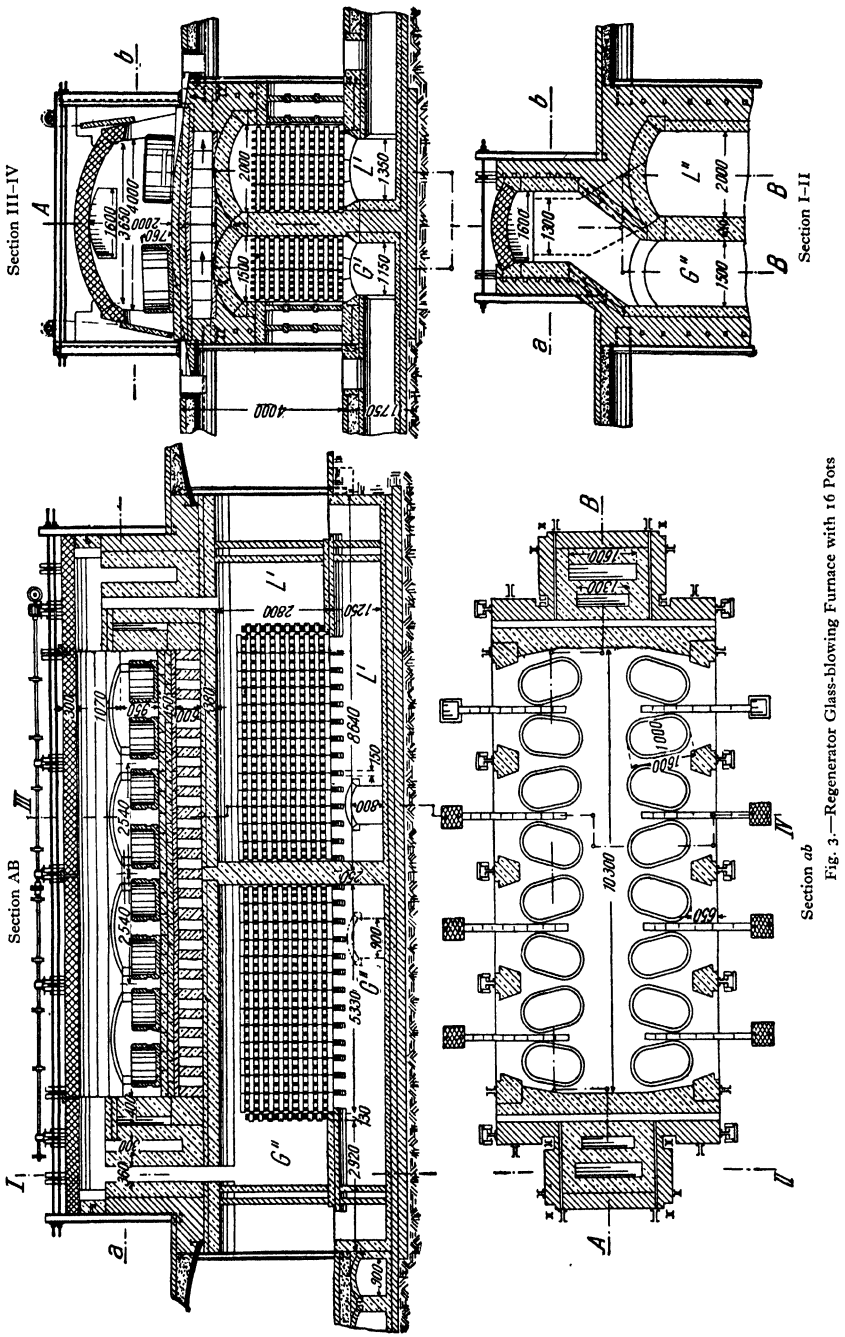
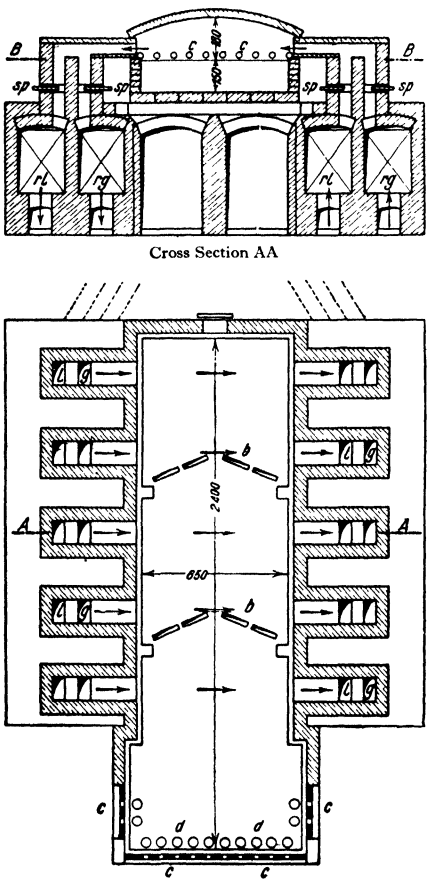


Fig. 3.—Regenerator Glass-blowing Furnace with 16 Pots

is too high the combustion is slow, the reducing layer remains too cold, and the carbon dioxide is again not reduced.

With regard to the utilization of steam or an underdraught of air, the former is used with mineral coal and brown coal briquettes and the latter with Bohemian brown coal. German brown coal cannot be used with the latter as it usually contains a large percentage of dust. Even the natural draught of the chimney must be carefully regulated.



Cross Section AA

Horizontal Section BB

Fig. 4.—Glass Tank Furnace

The generator gas is heated from 650° F. to 1800° F., the air from 60° F. to 1800° F. The efficiency of the regeneration is 88 per cent to 90 per cent, and the ratio of the regeneration loss to the total heat supplied to the generator is 6 per cent to 7 per cent.

2. In some types of glass furnace the smelting takes place in round or oval pots which are removed for charging and discharging. In other furnaces the smelting is carried out, either as an intermittent or continuous process, in stationary tanks which are usually rectangular. Examples are shown in figs. 3 and 4.

The work in the furnace shown in fig. 3 is carried out as follows. The mixture is inserted at 8 o'clock in the morning, the smelting being completed at 11 p.m.; it is then stirred cold until 3 a.m., and the temperature slightly raised at 5 a.m.; finally it is cast between 6 a.m. and 8 a.m. The principal constituents of the charge are sand, carbonates of lime and sodium, and sulphate of sodium. Coke is added to assist in the reduction of the sulphate, carbon dioxide is expelled from the carbonates, and silicates are formed. The charge contains on an average 5 per cent of water, and a yield of over 80 lb. of glass is obtainable from 100 lb. of charge.

Including the waste and residual glass, the smelting of 1 lb. of glass requires on an average about 1800 B.Th.U. This is made up as follows:

(1) Heating of the charge to the furnace temperature of 2400° F. The specific heat of the sand is 0·19, limestone 0·21, soda 0·28, sulphate 0·23, and glass 0·4.

(2) Vaporization of the water in the charge which amounts to about 5 per cent and requires 0·6 per cent of the calorific value of the fuel.

(3) Decomposition of the sulphate and carbonate:

$$CaCO_3 = CaO + CO_2 - 75{,}000 \text{ B.Th.U.}$$
$$Na_2CO_3 = Na_2O + CO_2 - 130{,}000 \text{ B.Th.U.}$$
$$Na_2SO_4 = Na_2O + SO_3 - 243{,}000 \text{ B.Th.U.}$$

The carbonates require 13·7 per cent of the heat supply and the sulphate 4·4 per cent. The actual reactions are more complex than indicated, the sulphate with carbon giving carbon dioxide and sulphite which in turn gives some sulphide.

(4) Smelting of the charge at 2400° F. The latent heat varies from 135 to 150 B.Th.U. per pound.

(5) Heat is supplied by the formation of the silicate:

$$Na_2O + 2SiO_2 = Na_2Si_2O_5 + 81{,}000 \text{ B.Th.U.}$$
$$CaO + SiO_2 = CaSiO_3 + 32{,}000 \text{ B.Th.U.}$$

(6) Heat supplied by the added charcoal whose calorific value is 13,500 B.Th.U. per pound.

The net heat required from external sources, viz. 1800 B.Th.U. per pound, is found to be from 13·5 to 14 per cent of the heat value of the coal supplied to generator. The losses represented by the remainder are accounted for by the following items.

(1) The heat capacity of the pots may require about 5 per cent of the calorific value of the fuel. They are heated to 2400° F. with the charge, and after it has been poured out they may fall to about 1200° F. before being recharged. Their specific heat is 0·24 to 0·25.

(2) The waste gases carry away 18 to 20 per cent of the heat value of the fuel. The temperature at the foot of the chimney is about 720° F. Combustion takes place with an excess of air of 25 per cent. Carbon dioxide from combustion and dissociation of the carbonates amounts to 17 or 17·5 per cent. The temperature of the gases issuing from the furnace may be 2200° F., the fall between that and 720° F. being utilized in the regenerators.

(3) Radiation and conduction losses are difficult to estimate and are usually taken as equal to all losses not otherwise accounted for.

The heat balance with the items enumerated above may be summarized as follows. The figures are, of course, subject to considerable variation according to the type of furnace and character of the glass.

HEAT SUPPLY

	Per cent.
1. Calorific value of the coal .. .. :	= 96·8
2. Heat value of the steam of generator	= 3·2
	= 100

HEAT CONSUMPTION

	Per cent.
1. Loss in the generator	= 12·2
2. Cooling of the generator gas	= 4·1
3. Glass formation	= 13·7
4. Waste gases	= 19·0
5. Regeneration loss	= 6·1
6. Heat capacity loss of pots	= 4·2
7. Conduction, radiation, and other losses	= 40·7
	= 100

Results of a test which are very similar to those given in the above table are shown in the heat distribution diagram of fig. 5, details being given in fig. 6.

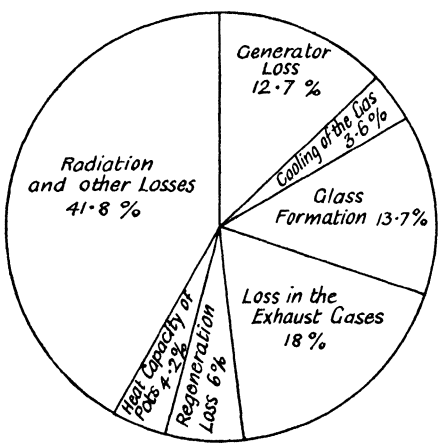


Fig. 5.—Heat Distribution of a Glass-blowing Furnace

The efficiency of such a glass-smelting furnace with generator gas could be increased to some small degree by reducing the losses due to cooling of the generator gas, and by utilizing more of the heat of the waste gases for the production of steam, but the principal cause of loss is radiation and conduction. Thick walls and small openings should be used. Great attention should also be paid to the direction of the flames of the glass furnace, so that the heat flows directly to the substance to be melted. In the continuous process, the maximum temperature developed in the refining zone should be 2450° F. to 2550° F. before the removal of the melting substance, which takes place at 2100° F. to 2200° F. in the manufacture of window glass. With bottle glass there is practically no refining zone, as the blowing is carried out at a higher temperature.

It has already been stated that 1 lb. of crude glass requires 1800 B.Th.U. for smelting only, and as this is 13·7 per cent of the total heat required, the latter is 13,150 B.Th.U. This is very nearly the calorific value of average mineral coal, so that the production of a given weight of crude glass requires the same weight of coal. The furnaces with T-shaped tanks require considerably over this amount. The production of water glass requires only one-third to one-quarter of this expenditure of coal. Recently experiments with purified generator gas have led to good results both as regards coal consumption and quality of the glass.

Glass-smelting furnaces with oil firing directly underneath are also used. The air is preheated in recuperators and combustion takes place with almost

the minimum amount for perfect combustion. They require somewhat delicate adjustment to maintain the correct furnace temperature.

(*e*) Enamelling Furnaces

These furnaces are designed almost exclusively as muffle furnaces, the muffles being made of cast iron or fireclay. In the past, directly heated enamel furnaces only were used, but these are now partly replaced by gas-heated

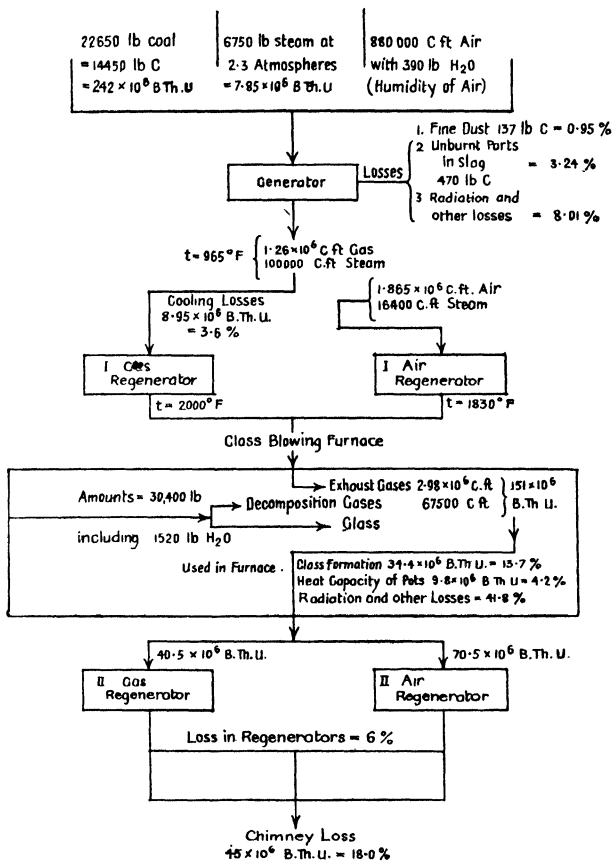


Fig. 6.—Diagram of a Glass-blowing Furnace

furnaces, which are much to be preferred on account of their uniform temperature and the possibility of a more intensive heat utilization. With a correct manipulation the coal consumption is also considerably reduced, while lower percentage coals can be used. A direct coal furnace can use mineral coal only.

(*f*) Crucible Furnaces

Crucible furnaces are chiefly used for smelting iron and other metals. The design varies according to the purpose for which it is intended and the fuel used. The fuels used are coke, oil, coal, and gas. It appears that surface combustion without flames has also possibilities in this direction. Graphite or clay is used for the material of the crucible. As in most cases a simple smelting process only, without any subsequent reactions, is carried out in these furnaces; the heat balance calculations are similar to those of the Martin furnace without the thermo-chemical reactions.

2. ROASTING, CALCINING, AND CHEMICAL PROCESSES

A simple process of calcining consists in the removal of certain constituents from chemical combination, most commonly water in combination or carbon dioxide. This action takes place in many cases during smelting, but with numerous kinds of ores it is advantageous to perform the operation separately before smelting. By suitable conditions, oxidation or reduction of other constituents may be performed in the same operation. Certain ores of copper, antimony, zinc, and iron are thus treated. The process consists of a slow roasting at a comparatively low temperature. Impurities of sulphur, which have a very deleterious effect on the properties of metals, are removed by oxidation. Water is also expelled and carbon dioxide from carbonates. With spathic iron ore oxidation to ferric oxide is performed, the tendency to form iron silicates in the blast furnace being eliminated. The expulsion of carbon dioxide from carbonates renders the ore more porous, which assists the blast-furnace reactions.

The original method of roasting consisted of mixing with a certain amount of fuel and burning in heaps in the open. Furnaces of various kinds are now used. Small vertical furnaces similar in shape to blast furnaces contain the ore and may be heated by either coal or gas. Small reverberatory furnaces are used particularly for copper pyrites, lead glance, and pyrites. Revolving cylindrical ovens which are used in some places have the disadvantage of producing considerable quantities of dust. The calculations for the heat reactions are similar to those which have already been detailed.

Many other chemical processes are similar to calcination and require the application of heat owing to the decomposition being endothermic. Various examples have been given in the section on blast furnaces. Saponification, the chemical resolution of fats into glycerine and fatty acids, takes place at high temperatures and under pressure in autoclaves at 8 to 10 atmospheres. In practice large amounts of heat, 25 to 50 per cent, are lost in external radiation and conduction in these processes. On the other hand, there are cases in which heat is required to promote combination.

$$N_2 + O_2 = 2NO - 78{,}000 \text{ B.Th.U.}$$

When nitrogen and oxygen are heated and the endothermic heat is supplied

there is practically no combination below 3000° F., but increasing amounts are obtained at higher temperatures, the combination also taking place with increasing rapidity.

3. ANNEALING PROCESSES

The process of annealing consists in the gradual application and withdrawal of a low heat, whereby a molecular rearrangement takes place without change of chemical constitution, the physical properties, however, being affected. The principal change is a softening of the material, but an equally

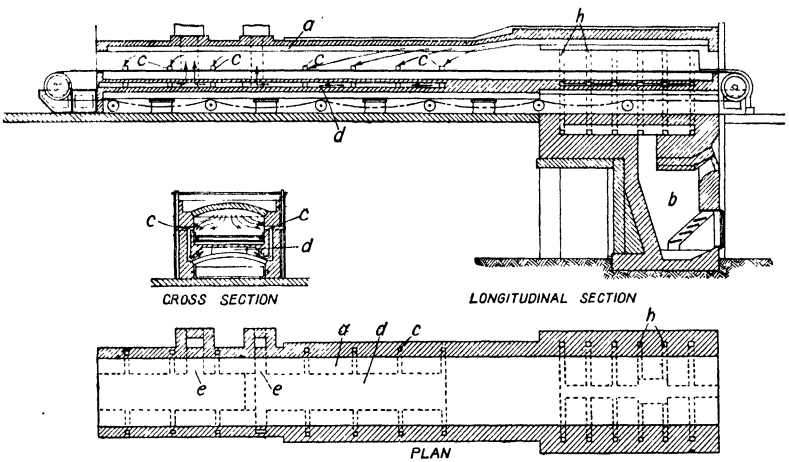


Fig. 7.—The " Teisen " Lehr

a, Annealing Funnel. *b*, Gas Generator. *h*, Gas Entrance Pipes. *c* Gas Exits. *d*, Gas Exit Passage. *e*, Flues

important effect is the removal of internal strains due to previous unequal heating or cooling. Metallic plates, sheets, and wire are annealed and also many forms of glass. Modern annealing furnaces or ovens are continuous in action. The design of a glass annealing " lehr " is shown in fig. 7. The glass is slowly passed through a tunnel-shaped chamber heated from above and below by the gases from the attached small gas producer. The sensible and calorific values of the gases are thus used. As the glass is moved on the carrier it is first slowly raised to a red heat and then slowly cooled. These furnaces may either be of the flame type, when the flames are in contact with the glass, or of the muffle type, when the walls of the chamber are brought to a red heat by the flames outside.

Another process for which the term " annealing " is not quite correctly used is the production of malleable cast iron. The hard castings are heated in covered boxes filled with hematite in granular form or other non-oxidizing agent. In European practice a temperature of 1500° F. to 1600° F. is maintained and in America 1300° F. to 1350° F. This temperature is produced

slowly, requiring 36 to 48 hr.; the steady temperature is continued for 60 to 75 hr., and the cooling occupies 40 to 52 hr., a total time of 6 to 7 days being thus required. With the most favourable type of furnace 1 lb. of castings requires 1·1 to 1·4 lb. of coal of 12,500 B.Th.U. per pound. The annealed castings are malleable and easy to machine. The conversion is produced by the decomposition of the impurity iron carbide, Fe_3C, to which the cast iron owed its hardness, and hence the process is not one of simple annealing.

4. " BURNING " PROCESSES

The term " burning " does not denote combustion but is the common term used for baking in the ceramic and cement industries. It is applied to the manufacture of pottery, bricks, and fireproof material and to the production of lime, cement, and plaster of Paris. Many types of furnaces are used, some of which have already been illustrated in Chapter III. In the ceramic industry, in particular, the cycle varies slightly at almost every application as regards temperature and the velocity or duration of the burning process, since the clay that makes up the initial substance can have many different properties. The amount of heat required per unit weight also varies.

The temperature of the process is a very important factor. Clay slate and kaolin have the highest melting-points, 3360° F. and 3330° F. respectively. By the addition of a flux such as K_2O, CaO, or MgO, the melting-point can be lowered in proportion to the amount added. Fireproof clays are those which melt above 2900° F. (Seger cone, 26). In addition to the melting-point, the sintering-point is also important. This is the point at which the clay and kaolin are in a stage of transition between a rigid and a melted state, parts of the inhomogeneous mixture being then in a liquid state intermingling with the other parts which are still in a solid state. Under these conditions the mass becomes smooth without change of shape. If the melting-point is only slightly above the sintering-point, it is necessary to maintain a very uniform temperature, as a slight increase might cause melting and deformation of the object being burned. Bricks made of clay rich in lime are burnt at 1450° F. to 1850° F. (Seger cone, 015*a*—05*a*), those made of clay poor in lime at 1900° F. to 2000° F. (Seger cone, 03*a* to 01*a*). With clay rich in lime the melting- and sintering-points are close together while with clay poor in lime the two points often differ by 200° F., thus requiring a less sensitive adjustment. Porcelain is fired in moulds at 2500° F. to 2700° F. (Seger cone, 14 to 18). The melting-point of the glaze must be lower than that of the fired material. The highest value used is that of porcelain glaze. Stoneware is burnt at 2150° F. to 2350° F. (Seger cone, 5*a* to 10), hard white ware at 2300° F. to 2500° F. (Seger cone, 8 to 12), and porous soft white ware at 2100° F. (Seger cone, 4*a*). Majolica ware and stove tiles have low burning temperatures of about 1800° F. to 1900° F., obtained by the addition of lead. With ordinary stoneware burning takes place first at 1450° F. to 1750° F. (Seger cone, 015*a* to 07*a*), then the glazing at 1800° F. to 1950° F., and finally a still higher temperature for the final firing.

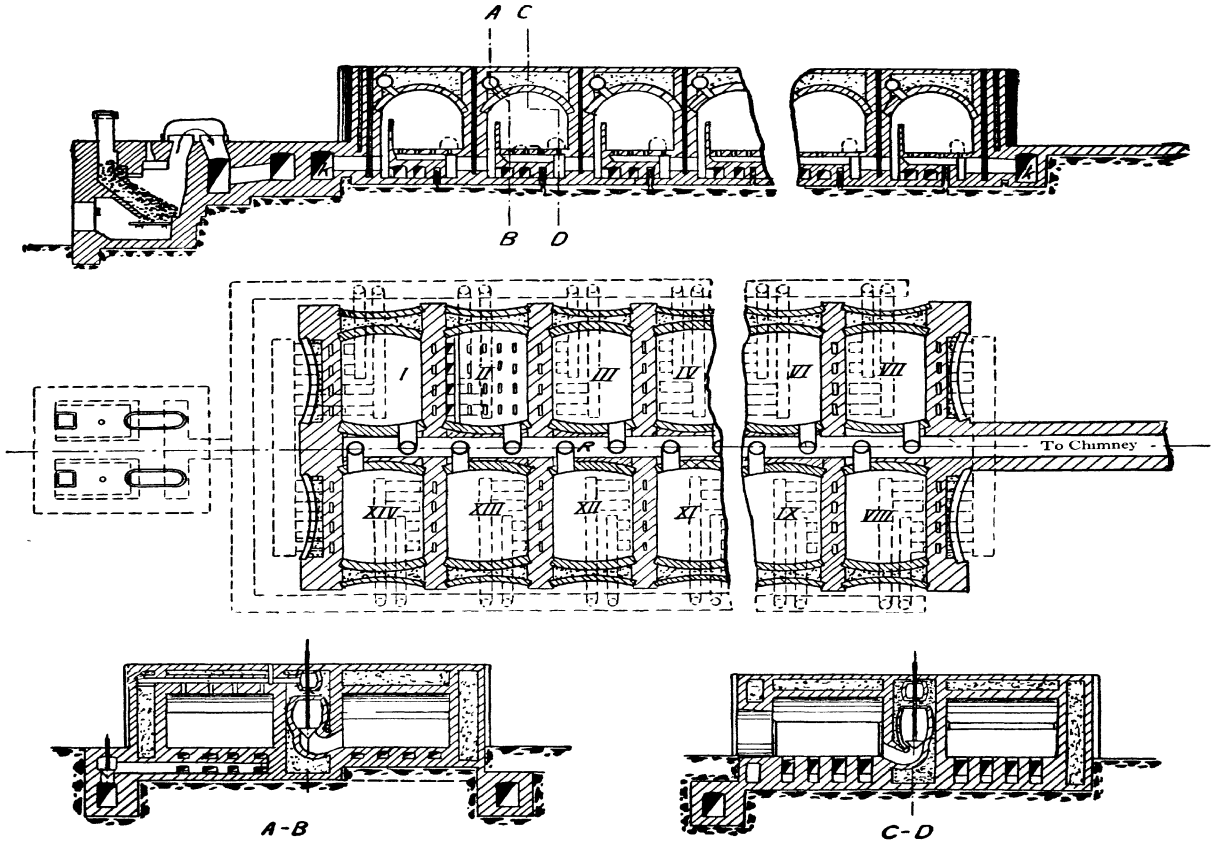


Fig. 8.—Gas Furnace for Firebricks

[*Facing p.* 186

The heat capacity of the material is not great and the heat required per pound of porcelain, bricks, or clay ware ranges from 1450 to 1800 B.Th.U. As an example of the performance of one type of ceramic furnace, the following data with reference to a gas-chamber furnace (fig. 8) are given:

Annual production of fireclay ware	20,000 tons.
	= 400,000 c. ft.
Period required for charging, burning, and emptying	10 days.
Charges per chamber per year	36.
Number of chambers	14.
Charge per chamber, 400,000/36 × 14	= 800 c. ft. = 40 tons.
Volume of each chamber, 800/0·6 (approx.) ..	= 1350 c. ft.

The heat balance for this production may be deduced as follows:

1. *Heat required to expel the water.* This is partly supplied by the waste gases. In some cases the greater part of the water is expelled by means of a dry current of air passed over the charge and in other cases special drying ovens are used. The last portion of water can only be expelled in the furnace. With 8 per cent water content expelled at 400° F., the single charge of 40 tons requires 40 × 2240 × 0·08 × 1200 = 8,600,000 B.Th.U.

2. *Heat to raise the temperature of the charge to the burning temperature.* If the average specific heat of the charge is 0·2 and the temperature 2600° F., the heat required is 40 × 2240 × 0·2 × 2600 = 46,500,000 B.Th.U.

3. *Heat in the waste gases.* Using crude brown coal with 47 per cent C and 12 per cent water and of calorific value 6400 B.Th.U. per pound, 1 ton of burnt fireclay requires the combustion of 0·3 tons of coal in the generator. The latter produces 3·6 lb. of gas per pound of fuel, resulting in 5·3 lb. of combustion products. If the products escape at 400° F. they carry away for each single charge 40 × 2240 × 0·3 × 5·3 × 0·24 × 400 = 13,600,000 B.Th.U.

4. *Radiation and conduction losses* amount to about 10 per cent of the heat, 40 × 2240 × 0·3 × 0·1 × 6400 = 17,200,000 B.Th.U.

5. *Heat lost by the endothermal reactions*, by burning of the walls, and evaporation of extraneous moisture is of the order 190 B.Th.U. per pound of charge, so that the heat lost per charge is 40 × 2240 × 190 = 17,000,000 B.Th.U.

6. *Heat lost through other causes*, such as leakage of gas, may amount to 12 per cent or 20,000,000 B.Th.U.

7. *Losses in the generator* not accounted for under the above items may be 20 per cent, or 33,000,000 B.Th.U.

The total heat requirements per 40 tons of charge are thus 155,000,000 B.Th.U. This would be supplied by 155,000,000/6400 × 2240 = 10·5 tons of coal. Therefore 1 ton of charge would require 0·26 tons of coal, which is somewhat less than that assumed in the preliminary calculations. The heat required for the actual burning, i.e. the raising of the temperature and endothermal reactions, is about 54,000,000 B.Th.U. The heat required,

reduced by that recoverable in the waste gases, is 141,000,000 B.Th.U. Hence the thermal efficiency

$$= \frac{54{,}000{,}000}{155{,}000{,}000} = 0{\cdot}35,$$

and the thermodynamic efficiency

$$= \frac{54{,}000{,}000}{141{,}000{,}000} = 0{\cdot}38.$$

5. SUBLIMATION

In this process the substance is converted from the solid to the vaporous state without passing through the liquid state, the resulting vapour being collected and condensed. The method is used mainly as a refining process, the temperature of sublimation being definite and different for different substances exactly as with boiling- and freezing-points. Muffle furnaces are used. A notable instance of its utilization is in the case of zinc. The fuel consumption for reduction of the ore and sublimation of the zinc varies from 3·5 to 7 times the weight of zinc obtained according to the ore used and type of furnace. The calculation of the heat balance is similar in character to those of preceding sections.

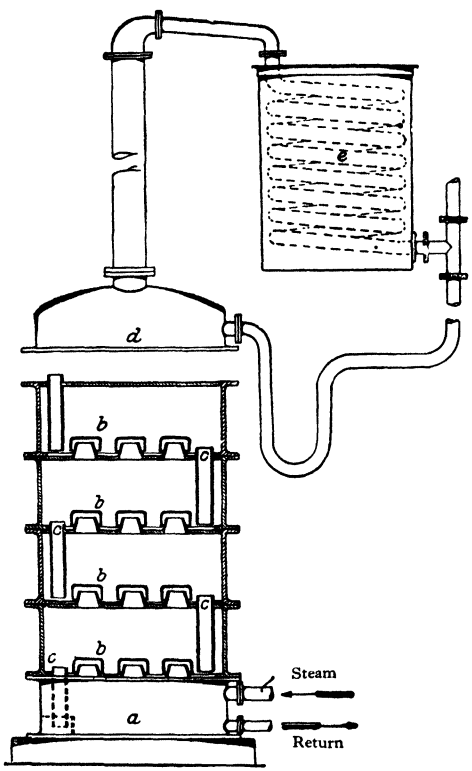


Fig. 9.—Distillation Apparatus

a, Base; *b*, Receivers; *c* Return Pipes; *d*, Hood; *e*, Condenser

6. BOILING AND EVAPORATION

The process of boiling is carried on in very many industries ranging from brewing and laundry-work to the manufacture of soap and chemicals. The heat may be applied directly from a furnace or by means of steam. Waste exhaust gases or exhaust steam are often used if a large supply is available. Plants of this

character are described in Chapter III. The heat calculations involved in the process of boiling are very simple and need not be dilated upon.

There are also many industrial processes in which a liquid has not only to be heated to the boiling-point but has to be vaporized. The separation of organic compounds by distillation at increasing temperatures is a simple case. In fig. 9 is shown a distillation apparatus for the rectification of alcohol, ether, or oils. The bottom chamber consists of a steam chest by means of which the fermentation products in the second chamber, containing perhaps 10 per cent of alcohol, are heated. The vapour will contain a larger proportion of the liquid of lower boiling-point and thus the condensate in the third chamber will be stronger in alcohol. The vapour from this will be still stronger in alcohol and so on to the top chamber, while the temperature will gradually decrease towards the top. The action is illustrated by the following table, which gives the comparative strength of the vapour and the liquid from which it has been formed.

Alcohol Contents of Boiling Liquid by Weight (per cent).	Boiling Point, °F.	Alcohol Contents of the Vapour by Weight (per cent).	$\frac{\text{Vapour Content}}{\text{Liquid Content}}$.
0	212	0	—
5	203	33·5	6·70
10	196	48·6	4·86
20	188	60·1	3·00
30	184	63·4	2·11
40	181	66·9	1·67
50	179	70·3	1·41
60	178	74·6	1·24
70	176	79·4	1·13
80	175	84·8	1·06
90	174	91·0	1·01
94	173	94·6	1·006

The latent heat of vaporization of alcohol and water at 173° F. is 376 and 990 B.Th.U. per pound respectively, while their specific heats are 0·615 and 1·0.

As the boiling-point of a liquid is lower at reduced pressures, a reduction in the amount of heat necessary for vaporization can be attained by conducting the evaporation at low pressures. In certain processes, as in the manufacture of sugar, it is necessary to evaporate the moisture at low pressures and temperatures in order to avoid chemical changes. If the distillate is conducted from the evaporation chamber and rapidly condensed there is a reduction of pressure within the chamber, a lowering of the boiling-point, and a stimulation of the evaporation until the flow of vapour is equal to that condensed. The condition can be produced and the latent heat of the vapour also utilized by conducting the evaporation in a series of chambers. The first vessel is heated with steam, and the vapour produced is used to heat the second chamber, which is of such a character that it can abstract the heat

rapidly and cause the required rapid condensation. Similarly the vapour from the second chamber is used to heat the third and so on. The last vessel

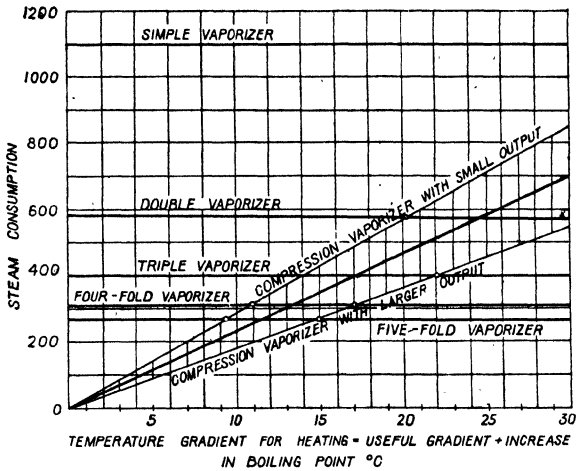


Fig. 10.—Steam Consumption for Water Vaporization (Fresh Steam)

of the series is connected to an ordinary type condenser with or without a vacuum pump. The diagrams in figs. 10 and 11 show the results obtained with vaporizers of this type having from one to five chambers, the diminished

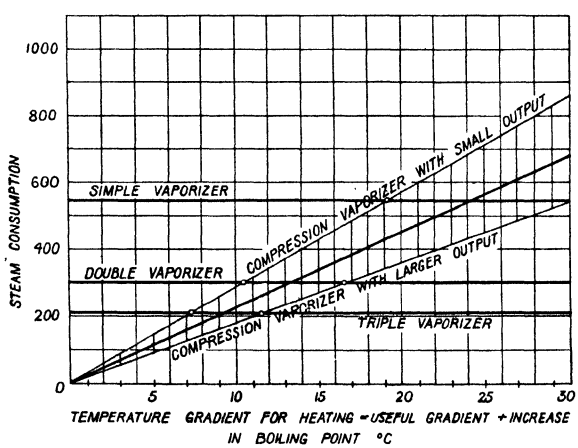


Fig. 11.—Steam Consumption for Water Vaporization (Exhaust Steam)

steam consumption for the multiple vaporizers being clearly shown. The curves represent experiments on the evaporation of water from a solution, the increase in boiling-point as the solution becomes stronger being plotted,

and the corresponding heat consumption for evaporation at that temperature.

7. DRYING

The operation of drying is physically the same as that of evaporation, and it can be conducted in the manner described in the preceding section. Where, however, the dried product is the object of the operation and the vapour is waste alternative methods are available. If a current of hot air is passed over the wet substance, it will cause the evaporation of some of the liquid and carry it away as vapour. The amount will depend upon the temperature and also on the time of contact of the air with the liquid, but there is a limit at any given temperature for the amount of vapour which unit volume of air will absorb. When the air contains this maximum amount it is saturated. In the case of water the following table gives this maximum amount at different temperatures, together with the vapour pressure which is exerted.

° F.	Maximum Vapour Pressure, Inches, Hg.	Weight of Vapour in 1 C. Ft. of Saturated Air, Lb.	Weight of 1 C. Ft. of Dry Air, Lb.	Weight of Air in 1 C. Ft. of Saturated Air, Lb.
0	0·045	0·00008	0·0863	0·0862
20	0·109	0·00019	0·0827	0·0826
40	0·246	0·00041	0·0794	0·0792
60	0·513	0·00082	0·0764	0·0759
80	1·02	0·00156	0·0735	0·0725
100	1·92	0·00282	0·0709	0·0692
120	3·42	0·00509	0·0685	0·0655

If the air at a given temperature t contains only w_1 lb. of water per cubic foot, while the maximum amount which it could hold in the state of vapour as given in the table is w lb. per cubic foot, then the ratio w_1/w is the relative humidity, and it is very nearly equal to p_1/p where p_1 is the partial pressure of the vapour present and p the maximum possible pressure as given in the table. If the volume of the air and water vapour is changed from v_1 to v_2 by either pressure or temperature changes, the density of the water would tend to change to w_1v_2/v_1 lb. per cubic foot, and if this is greater than the maximum density possible at the new temperature according to the table, then condensation will take place. If it is less than the maximum, the air will still be able to absorb a certain amount of moisture at the new temperature. An example of the calculations involved in a drying process is given below.

17 tons of damp material contains 60 per cent of moisture and this is to be reduced to 15 per cent in 10 hr. Thus 7·65 tons of moisture must be evaporated. The drying is to be effected at 75° F. and the external air is at 40° F. Assuming that the atmospheric air at 40° F. is saturated, it will contain 0·00041 lb. of water vapour per cubic foot, and when it is heated to 75° F.

at constant pressure the expansion of the air will reduce it to 0·00038 lb. per cubic foot. Now the maximum amount which air at 75° F. can hold is 0·00135 lb. per cubic foot, but to attain this a considerable time of contact with liquid is necessary, and it is safer to assume that only an 80 per cent relative humidity will be obtained. With this assumption each cubic foot of air will absorb an additional $0{\cdot}00135 \times 0{\cdot}8 - 0{\cdot}00038 = 0{\cdot}0007$ lb. The number of cubic feet to absorb 7·65 tons is therefore $7{\cdot}65 \times 2240/0{\cdot}0007 = 24{\cdot}5 \times 10^6$ at 75° F. or $22{\cdot}9 \times 10^6$ at 40° F., weighing $1{\cdot}8 \times 10^6$ lb. Assuming that the wet material with its water content has an average specific heat of 0·35 and an initial temperature also of 40° F., while the latent heat of water at 75° F. is 1030 B.Th.U. per pound, the heat required for the drying is:

	B.Th.U.
Heating wet substance, $17 \times 2240 \times 0{\cdot}35 \times 35$ =	500,000
Water vaporization, $7{\cdot}65 \times 2240 \times 1030$ .. =	17,700,000
Heating the air, $1{\cdot}8 \times 10^6 \times 0{\cdot}24 \times 35$ .. .. =	15,100,000
Losses about 10 per cent .. .. =	3,700,000
Total	37,000,000

The whole of this heat must first be given to the air, which must be raised to a temperature x, such that

$$1{\cdot}8 \times 10^6 \times 0{\cdot}24 \times (x - 40) = 37{,}000{,}000,$$
$$x = 126° \text{ F.}$$

The air will thus enter the drying-chamber at 126° F. and leave with the absorbed moisture at 75° F. If the available heating appliances for the air are not able to raise it from the atmospheric temperature of 40° F. to 126° F., a larger quantity of air must be used. It will be seen that the hygrometric state of the atmosphere has a considerable effect on the efficiency of the drying process when it occurs at a low temperature such as 75° F. The effect of varying atmospheric conditions when evaporation is to take place at high temperatures is relatively less. If heat is given to the air by means of low-pressure steam whose total heat is 1155 B.Th.U. per pound, while its total heat as condensed water at 75° F. is 43 B.Th.U. per pound, the heat transfer from steam to air being accomplished with an efficiency of 75 per cent, then the weight of steam required $= 37{,}000{,}000/1112 \times 0{\cdot}75 = 44{,}500$ lb.

Experiments on the evaporation of water in small quantities as in the drying of paper or plants have given a relationship

$$W = k\frac{p_1 - p_2}{P},$$

where W is the weight evaporated per hour per square foot, p_1 and p_2 are the maximum vapour pressures of water at the temperature of the surface and air respectively, P is the barometric pressure, and k is a coefficient which increases with the velocity of the air.

The curing of animal or vegetable matter is in part a drying process

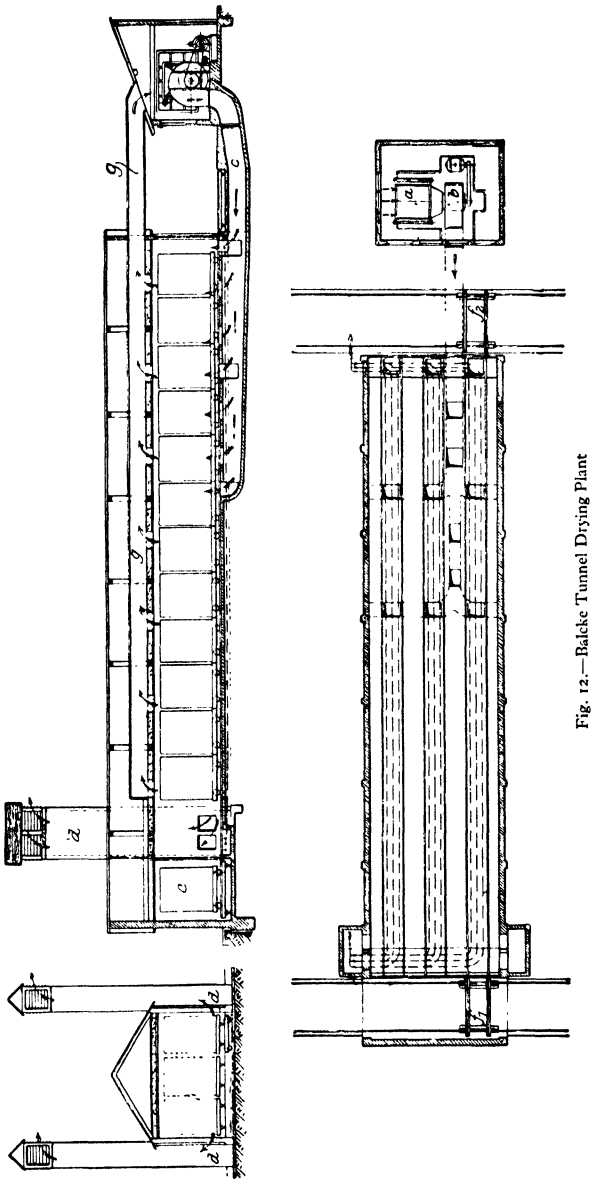


Fig. 12.—Balcke Tunnel Drying Plant

including the expulsion of water both mechanically and chemically held. For the latter the endothermic heat of decomposition must be supplied.

Fig. 12 shows the design of a tunnel drying plant for many kinds of material. The substance is placed in shallow trays, a number of which are

carried on a truck which slowly travels down the tunnel. Air heated by exhaust steam is forced by the fan *b* through the pipe *c* into the tunnel and is withdrawn, laden with moisture, by the pipe *g* to a condenser *a*. Additional ventilation for the spent air is provided by the shafts *d*. The mode of introducing the trucks *e* and withdrawing them is shown at f_1 and f_2. A design of vacuum drying-chamber is shown in fig. 13. This form is used in cases where the water must be removed without the substance being heated to a high temperature. Certain dyes and chemical preparations which would be adversely affected by too great a heat are dried in chambers of this character. The arrangement of a plant for drying large quantities of grain is given in fig. 14. There is a continuous flow of grain from the silo above through the drying-chamber and out through the hopper below. The chamber is heated by a flow of air which itself has been raised in temperature by steam flowing in pipes across the air inlet. A plant of this character reduces the water content of various kinds of grain initially 17 to 20 per cent to a value between 12 and 15 per cent. For an output of 1 ton per hour it requires 5 h.p. and 4 cwt. of steam.

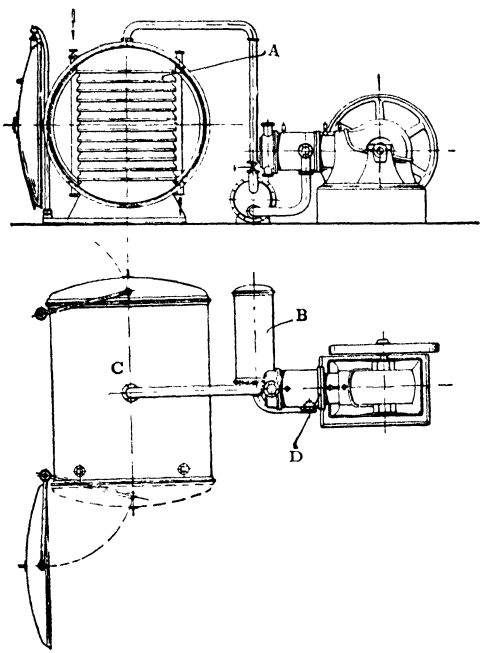


Fig. 13.—Balcke Vacuum Drying Cupboard

A, Heating Plates; B, Surface Condenser; C, Drying Cupboard; D, Vacuum Pump

8. HEATING OF ROOMS

The problems in connexion with the heating of small house rooms are of a different character to those concerned in the heating of large rooms, halls, or factories. In the former case the question of attractive appearance and convenience of position is of great importance. In England the use of open coal fires or gas fires in living-rooms is almost universal, electric stoves and oil stoves being used to a very small extent. Closed stoves, such as anthracite and coke stoves, are used sometimes in houses, and in the latter case also in factories. The use of closed coke stoves which also serve for water heating in kitchens is increasing rapidly, while the use of kitchen stoves for the three-fold purpose of room heating, water heating, and cooking is

dying out. Closed stoves for heating are used to a large extent on the Continent. Central heating is used very rarely in England in small houses, although it is used abroad as well as in large buildings everywhere.

(*a*) Open Fires

The efficiency of fuel consumption in open fire grates with coal fires has been investigated very thoroughly by Dr. Margaret Fishenden under the

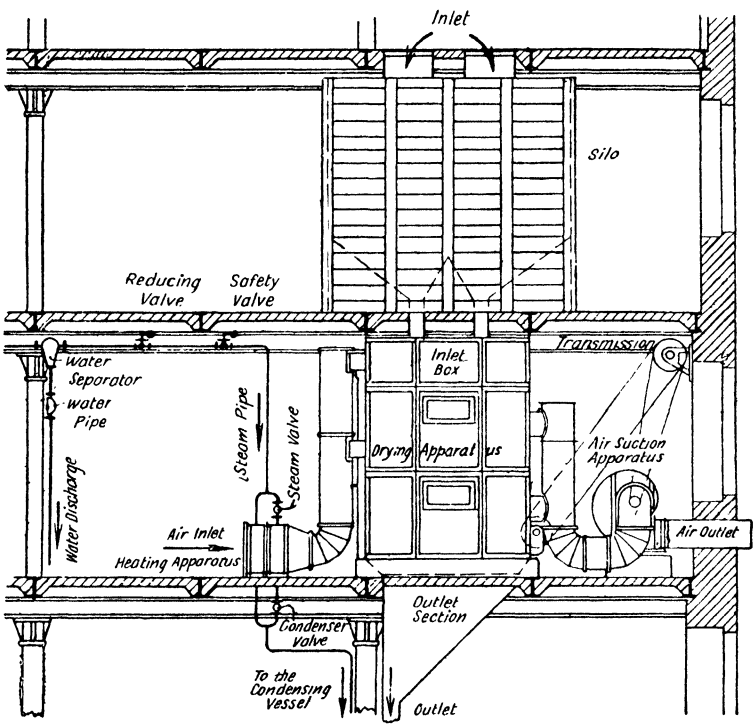


Fig. 14.—Topf Drying Plant for drying Grain

Fuel Research Board. Heat is distributed throughout the room containing the fire principally by radiation, and bodies receiving the radiation are heated while the air may receive very little increment of temperature. After deducting the heat lost by unburnt fuel in the ash, the amount of heat radiated varies between 19 and 25 per cent of the calorific value of the burnt fuel. The grates of modern design show no advantage in this respect over those of older make. The former, however, being usually without bars and having a radiating surface inclined at an angle not greater than 30° from the horizontal, radiate the heat at a greater angle than the older designs which have bars and some vertical radiating surface. The newer grates are thus advantageous to a person near to the fire but disadvantageous to a person some

distance away. In all grates a large amount of heat is carried from the room by the hot gases. With a normal draught its value, measured at ceiling-level, reaches 50 per cent of the calorific value of the burnt fuel, but this can be reduced by a reduction of the draught where that is possible. The partial closing of the chimney reduces the fuel consumption much less than it reduces the air flow, as the latter is always in great excess of that required for combustion, and hence the flue gas temperatures are higher with reduced consumption. It follows, therefore, that the brickwork of the chimney is hotter with reduced consumption and the heat which finds its way into the room from the chimney is increased. That proportion of the radiated heat which is absorbed by the air results in a higher air temperature with the reduced flow. The experimental figures would therefore lead to the deduction that the advantages of modern designs of open coal fires lie in the reduction of draught. Part of the 50 per cent or less of heat which is contained in the flue gases will be given up to the rooms above, where such exist. Of the remaining 25 per cent not yet accounted for, some will pass through the wall behind the fire either into a room there or to the outer air where the fire is on an outer wall. Some will give rise to convected heat in the room from the hot brick and metal surroundings of the grate. The total amount of heat from an open coal fire which is given to a room containing the fire is estimated at 30 per cent.

It is possible to use low-temperature coke containing 10 to 15 per cent of volatile matter in an open fire grate, and the proportion of radiated heat is found to be somewhat greater.

It must be pointed out that these efficiency figures relate solely to the amount of fuel burnt, and that the ease with which it is possible to make and to keep the fuel burning, the rate of burning, and the amount of unburnt fuel are additional considerations, and that in these respects modern grates are improvements.

The gas fire may have either a luminous flame or a Bunsen flame which heats fireclay material to a radiating temperature. The latter with its higher temperatures and greater cleanliness has practically superseded the former. The amount of heat radiated from many types of fireclay gas stoves are all in the neighbourhood of 50 per cent of the calorific value of the gas, and with the addition of some convection heat a total of about 65 per cent may be given to the room containing the fire.

As electric heaters can be placed in the middle of a room and require no chimney they may be said to have an efficiency of 100 per cent. The " lamp " radiators give a low proportion of radiant heat and distribute the remainder by low-temperature convection. The bare high resistance wire heaters, on the other hand, give a high proportion of radiant energy.

Although electrical energy can be consumed in a house in this manner without waste while gas and coal are burnt with 35 per cent and 60 per cent waste respectively, there has been waste at the generator works which is reflected in the price. The relative cost of the various methods based upon a delivery into a room of 200,000 B.Th.U. (a day's requirements) is as follows:

	Efficiency.	B.Th.U. Required.	Fuel Required.	Price.	Cost.
Coal ..	30 per cent	667,000	47·5 lb.	45*s.* per ton	11½*d.*
Gas ..	65 ,,	308,000	616 c. ft.	4*s.* 6*d.* per 1000 c. ft.	33½*d.*
Electricity	100 ,,	200,000	58·5 units	1½*d.* per unit	87½*d.*

The advantages of electricity over gas and coal therefore lie in convenience, cleanliness, and elimination of domestic labour.

The method of measuring the most important part of the heat received, i.e. the radiant heat, is as follows. An imaginary spherical surface is drawn with centre at the centre of the fire and a radius of about 3 ft., and is divided into a large number of areas of 1 sq. ft. A thermopile is placed at the centre of each of these areas in succession to obtain the relative radiation through them, and the average reading for the whole area is obtained. A radiometer is then placed at the point where the maximum radiation is transmitted and the absolute value of the heat radiated to that point in a given time is found. The total radiation for the whole area can then be calculated by multiplying by the ratio of mean to maximum thermopile readings and the number of areas involved. An illustration of one type of radiometer is given in fig. 15. The heat is absorbed by the suspended block of brass, the face towards the radiation being corrugated and coated with lamp black. Except for this side, the block is surrounded by a double-walled metal box, through the walls of which a stream of water at a constant temperature is flowing. The exposed side can be cut off from the radiant heat by a double screen to take readings of the heat lost from the block to its surroundings. The heat gain when exposed to the radiant heat is measured by its rise in temperature and its heat capacity. The difference between the distribution of radiant heat for an old-fashioned bar grate and a more modern barless grate is shown in figs. 16 and 17. In the former there is very little difference in the intensity of radiation in a vertical plane from the horizontal to an inclination of 60°, while in the latter there is a marked maximum at 60° and only half the intensity in a horizontal direction.

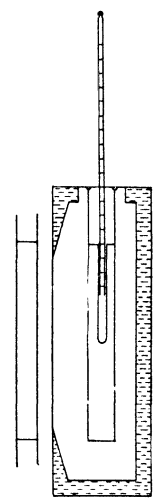


Fig. 15.—Richmond Radiometer

With ranges that serve for combined room-heating, cooking, and water-boiling purposes, the heating efficiency is considerably lower even when the range is not in use for cooking or water heating. When mineral coal is used the radiant efficiency varies between 9 per cent and 12 per cent for kitchen ranges, while for certain special types which simulate the single-purpose fire but are very inefficient for cooking purposes, the radiant efficiency was also only 10 per cent.

(*b*) Closed Stoves

In England the closed stove as a method of house heating is not popular, but as it can be removed from the walls, heat losses can be reduced and its efficiency increased. The draught in the room need not exceed that required for combustion, while with the open fire much air passes up the chimney without being in contact with the fire. High temperatures of the flue gases can thus be obtained and the heat received from the flues increased. Iron stoves only are used in England, but on the Continent very large tiled stoves made of firebrick are also used which have a large heat capacity and act as heat accummulators. In most cases the iron stove has little heat capacity,

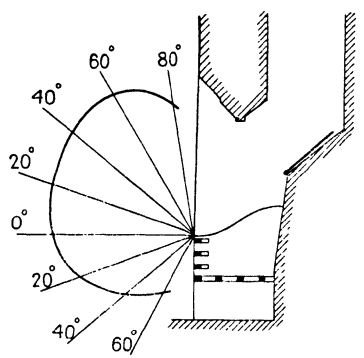


Fig. 16.—Open Grate with Bars: Distribution of Radiant Heat

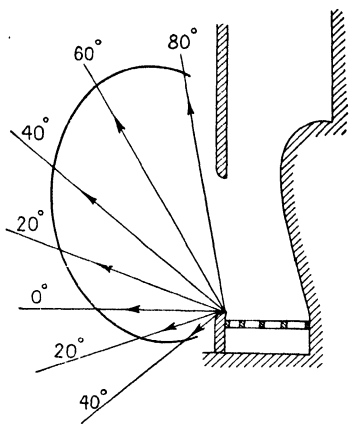


Fig. 17.—Modern Barless Grate: Distribution of Radiant Heat

and the heat supply to the room comes to an end soon after the combustion ceases. The temperature conditions and the heat delivery thus vary greatly.

In many cases there is a possibility that part of the coal in the stove burns only to carbon monoxide. This may be avoided by regulation of the air supply or by reducing its velocity. In stove heating it often happens that as a result of a badly arranged path for the gases in the stove they leave it at the entrance to the chimney at too high a temperature. This is caused either by wrong stove dimensions, a bad adjustment of the air required for combustion, or a sudden change in the weather. It can be remedied by the insertion of heat economizers, which consist either of enlarged stove pipes or of extended pipes in which the gas is made to ascend and descend. These economizers present a large cooling surface, so that the hot gases enter the chimney at a low temperature. They are, however, superfluous in properly constructed stoves with an adjustable air supply. Another aid to economical use which is often omitted is a convenient arrangement for decreasing the size of the fire space in warmer weather, either by the insertion of fireproof bricks or a change in position of the grate.

The efficiency of an iron stove is open to the same influences as in other furnaces, the losses being similar. Owing to the smaller dilution with excess air it is possible to measure any incomplete combustion more accurately than with an open range. Allowing for this and for unburnt material in the ash, the firing efficiencies of iron stoves vary between 78 per cent and 84 per cent. The radiant heat can be measured as for open fires, and is smaller owing to the lower temperatures of the radiating surface, but the convection heating with a stove in the centre of a room and free space above it will be considerably greater. To obtain a required value for the heating of a room a small outer stove surface with a high temperature or a large outer stove surface with a low temperature can be chosen. In the former case it should be noticed that the *total* radiated heat is small while the heat distributed in the room through conduction as a result of air flow is quite considerable. Though the external temperature of the large stove is lower there is a more appreciable amount of heat radiated in the neighbourhood, while less heat is distributed by air flow.

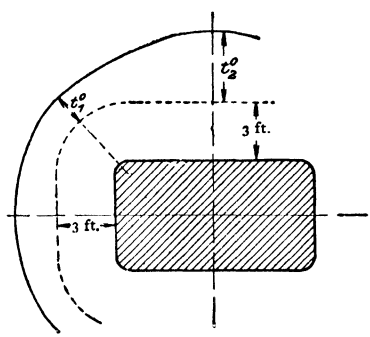


Fig. 18.—Temperature Distribution

The greater regularity of the heat emission in all directions from an isolated closed stove permits of the use of a coefficient of uniform heating to express the degree of regularity. At the edges of the stove the temperature is considerably lower than at other parts, and hence the radiation from a circular form of stove is more uniform than from a rectangular stove, which should be well rounded and without sharp edges. The non-uniform temperature distribution is caused by accumulation of material in the corners, and also by the combustion gases leaving a dead isolated space there. The value of the coefficient may be obtained by measuring the temperature with a shielded thermometer at a standard distance from the stove (say 1 yd.) for a number of points in a horizontal plane when the stove is in a large room or space. The diagram in fig. 18 shows the character of the results obtained. If the minimum temperature is t_1 and the maximum t_2 while the temperature of the unheated room is t_0, then the coefficient is given by $(t_1 - t_0)/(t_2 - t_0)$. Measurements should be made in different horizontal planes to obtain the temperature distribution and the coefficient at various levels from foot- to eye-level.

The overall efficiency of a stove involving heat losses during combustion, heat losses after production, and uneven distribution can only be found by comparison with a standard stove. It varies with the room in which it is placed and the standard stove must be tested in a similar room. An electrically heated stove is the most suitable standard, as its energy can be measured accurately and its heat output is not dependent on variations in fuel or weather.

Coke and anthracite are the most suitable fuels for closed iron stoves. They contain very little volatile constituents and burn almost without flames. Care must be taken, however, to ensure complete combustion so that no carbon monoxide is allowed to escape unburnt to dioxide. Non-bituminous flame and bituminous coal should have larger combustion chambers which are not completely filled, so that the combustion in the flames can be complete.

In house furnaces sufficient attention is not paid to the cross section and height of the chimney. If the chimney is not high enough there is insufficient draught in the upper stories, while an excessively large cross section causes too small a draught. The insertion of heat accumulators in the furnaces is also important. These are usually made of firebrick, and are used partly for taking up the temperature variations during the firing period and partly for preventing the room from becoming rapidly cooled after the fuel has been burnt up.

(*c*) Small Boilers and Cooking Stoves

The small boilers which are now being extensively used for providing domestic hot water in houses have a water-heating efficiency of 30 per cent with bituminous coal and 40 per cent with coke, which can be increased somewhat by suitable protection of the hot-water cistern and pipes. They consist essentially of an iron double-walled cylinder, the water being contained within the double walls and the fire in the centre. The larger part of the cylinder at one side is cut away, the opening being traversed by bars, while a door can completely close it (see fig. 15, Chapter III). The efficiency figures above were obtained with the door open, so that in addition radiant heat was distributed into the room equal to 7 or 8 per cent of the calorific value of the fuel. The hot-water efficiency of the three-purpose ranges already mentioned, i.e. kitcheners with boilers and cooking ovens, varies between 13 and 19 per cent. It should be remembered again that these figures are in terms of the fuel burnt, and that if a design allows much unburnt fuel to fall into the ash economic efficiency is proportionately reduced.

It is difficult to estimate the cooking efficiency of a stove with an oven or with a hot plate for saucepans. As estimated by the amount of boiling water produced by a certain amount of fuel, the efficiency is only 8 to 15 per cent. In some modern stoves a great improvement in economy has already been obtained by preheating the combustion air, and varying the height of the grate for different substances in order to adjust the fuel charge. The design of the stove should also be such that a good combustion is obtained automatically without an excessive amount of attention.

In stoves used for cooking which use solid fuel, it should be arranged that the flames and the burnt gases meet the cooking vessels in a perpendicular direction and not simply flow past as is often the case. Coke cannot be used effectively in cooking stoves of the usual design. It requires a high layer of fuel for complete combustion. When the flames from a mineral

coal first come into contact with a cold metallic surface, the cooling lowers the temperature to below the ignition temperature, so that some gases escape unburnt. This causes hydrocarbons to be decomposed and the surface to be covered with soot; it also makes heat conduction more difficult. To avoid this in the case of saucepans special care should be taken to obtain a proper combustion at the beginning of the cooking. The pans should be first heated by placing them on the covered portions of the stove before they are placed in direct contact with the flames.

The use of liquid fuel such as paraffin either for heating or cooking is very often rendered inefficient owing to incomplete combustion and the formation of soot. For small quantities of heat the high combustion temperatures required to avoid this are obtained with the high-pressure stoves, in which a fine jet of fuel is vaporized before combustion by impinging on a surface maintained at a red heat by the combustion. A preliminary heating with some more volatile fuel is necessary to start the action. Gas cooking stoves with the burners within the oven are much more economical than coal stoves. A decrease in the fuel used when lower temperatures are required is accompanied by a large decrease in efficiency, as the air supply is not adjustable.

(*d*) Indirect Heating of Rooms

The indirect heating of rooms may be by means of water or steam at high or normal pressure, or by means of air. High-pressure steam heating is carried out with either saturated or superheated steam. It has been found that superheated steam has few advantages over saturated steam as its heat contents are only slightly greater. The total heat of saturated steam at atmospheric pressure is 1150 B.Th.U. per pound, that of saturated steam at 10 atmospheres 1200 B.Th.U., and that of steam at 10 atmospheres and 100° F. superheat is 1255 B.Th.U. High-pressure steam has an advantage over low-pressure steam in that a proportionately larger amount of heat can be contained in a given volume. (It is necessary where there is some distance between the place where the steam is produced and the building that is to be heated.) Even in these cases, however, low-pressure steam is often used with suitable heat insulation in order to avoid the necessity of using heavy pipes and to reduce leakage.

The pressure difference lost in forcing the steam through the piping is very largely affected by the bends and valves. If the length l is in feet, the internal diameter d in inches, the flow of steam w lb. per hour, the condensed water w_1 lb. per hour, ρ the density in pounds per cubic foot, then the difference in pressure can be expressed by the empirical formula in pounds per square inch,

$$p = (l + 3{\cdot}5dk)\frac{(w + \frac{1}{2}w_1)^2}{\rho d^5}10^{-7}$$

where k is the sum of a number of constants k_1, k_2, . . . and represents the friction of each bend or obstruction. For a smooth bend $k = 0{\cdot}3$ to $0{\cdot}5$

for a rectangular bend 1·0, for an open valve 0·1 to 3·0 according to its character, and 2·0 for a sudden alteration in the cross section.

Low-compression steam heating is either carried out directly with a special boiler or takes the form of exhaust steam heating. In the former case the boiler works at an excess pressure of 0·04 to 0·10 atmospheres. In the latter case the pressure may vary between 5 atmospheres excess pressure and a very considerable vacuum, according as the steam plant is condensing or non-condensing. A steam-engine supplies only a limited amount of steam, and if large amounts of heat are required a special boiler with the exhaust steam as additional may be used. The formula for the pressure required to force the steam through the pipes may be simplified for cases where the pressure is not much different from atmospheric. If the heat required per hour in B.Th.U. is H_1, and 1 lb. of steam in condensing gives up 970 B.Th.U., while the density is 0·037 lb. per cubic foot, then for a straight pipe

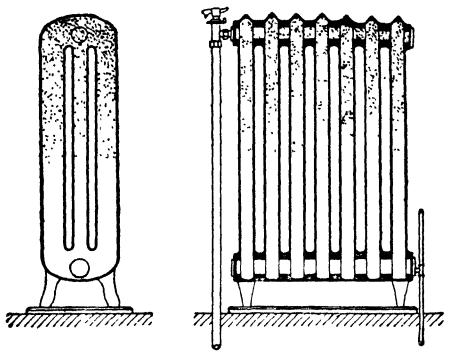


Fig. 19.—Steam Inlet at the Top

$$p = 2{\cdot}7 \times 10^{-12} \frac{H_1}{d^5} l.$$

The steam passes through pipes which transmit its heat by radiation and convection to the room. The amount transmitted from a given length of pipe can be increased by the use of ribs or fins although the heat per unit area is less. The heat transfer increases a little with the velocity. The use of radiators enables a large amount of heat to be transferred in a given space, though the reduced velocity in certain parts diminishes the amount per unit area. To promote the circulation and to render it more uniform throughout a radiator, a system of injecting air into each unit of the radiator has been proposed, its action being shown by the diagrams in figs. 19 to 21.

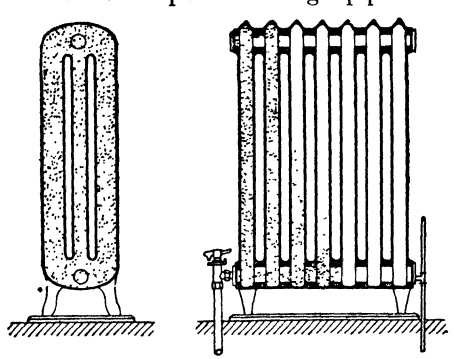


Fig. 20.—Steam Inlet at the Bottom

High-pressure hot-water heating has fallen out of favour owing to the difficulties of avoiding leaks and bursts. The heat content of the water is nearly proportional to its temperature. At atmospheric pressure it can be

heated to its boiling-point, 212° F., and if 1 lb. is cooled to 100° F. it would give up 112 B.Th.U. Under an absolute pressure of 10 atmospheres it could be heated without vaporizing to 356° F., and would transmit 256 B.Th.U. if cooled to 100° F. The usual system of hot-water heating is at atmospheric pressure, and the circulation is induced by the reduction of density with temperature increase. The hot water hence rises, gives off its heat where required, and returns to the boiler, placed at the lowest point, when cooled. The circulation is automatically such that the working pressure produced by the difference in density in the hot and cold pipes is equal to the sum of the resistances experienced.

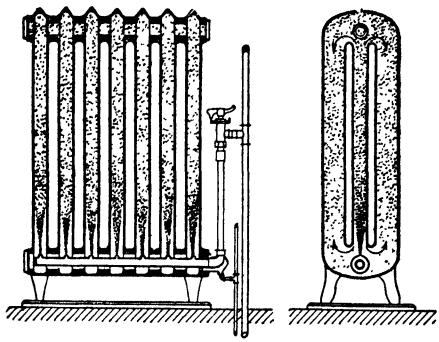


Fig. 21.—Steam Inlet with Air Circulation

Experimental figures for the heat given off in various cases are quoted below for a temperature difference of 1° F. between the steam or water and the room to be heated, in B.Th.U. per square foot.

	High-pressure Steam.	Low-pressure Steam.	Low-pressure Water.
Smooth pipes	2·6 to 2·9	2·3 to 2·6	1·8 to 2·3
Ribbed pipes	1·4	1·2	1·0
Radiators ..	1·8 to 2·1	1·4 to 1·9	1·2 to 1·6

In order to increase the circulation a mixture of water and steam is used in the Reck system which is illustrated in fig. 22. A mixture by weight of 90 per cent water and 10 per cent steam at the atmospheric boiling-point of 212° F. would give up in cooling to 100° F. a quantity of heat of 209 B.Th.U. per pound in place of 112 B.Th.U. for water only. Steam produced in the low-pressure boiler N is utilized in the preheater V for producing hot water. This hot water is mixed with steam at D and rises to the closed vessel O, where steam and water are separated. The steam flows to the low-pressure

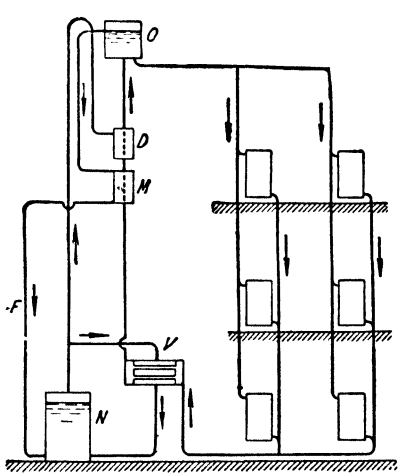


Fig. 22.—The "Reck" System of Heating

boiler N, passing on its way the condenser M, where it gives up heat to the water through condensation. The steam converted to water then flows through F back to the boiler. The velocity is thus stimulated in the upward branch of the system, the downward branches through the radiators containing only water at approximately steam temperature.

In all systems a regulating valve should be placed at the entrance to each room in order that the rooms may be uniformly heated. In addition a general regulating valve near the boiler is required to adjust simultaneously the heat to all the rooms, as regulation by means of the boiler furnace takes considerable time.

An example of a boiler for central heating with hot-water systems has been given in Chapter III, fig. 16. For direct firing, coke furnaces predominate. For efficient combustion the depth of the layer of coke should not fall below 6 in., and if it is necessary to vary the heat required owing to changes in the atmospheric temperature, it is best to adjust the heat by decreasing the grate area. This can be effected by inserting firebricks, by means of which part of the heating surface can be cut off as well as part of the grate area.

Recently briquettes have been used in addition to coke for central-heating boilers. Mineral coal, crude brown coal, and peat can also be used provided that the combustion conditions of these fuels (removal of water, distillation, and gasification) are taken into account. It must be remembered that coke only glows and burns without flames, while mineral coal and brown coal have to be gasified as well as burnt, and space for the flame is required above the fuel. The use of low-percentage fuels for central-heating boilers still requires close investigation.

In addition to water- and steam-heating systems, there are various systems of hot-air heating. The air may be heated by allowing it to flow past steam pipes or by passing the air in iron or fireclay pipes through a furnace. The efficiency of furnace heating, i.e. the ratio of the heat received by the air to the heat contents of the fuel, may reach 50 to 65 per cent. The efficiency of the room heating depends on whether the air is returned to the heating plant or is allowed to escape. It also depends, of course, on the usual factors of the room construction, the character of walls, windows, and ventilation. To heat 1 c. ft. of air measured at 60° F. through 1° F. requires 0·0184 B.Th.U. If t_0 is the temperature of the air before heating, t_1 when leaving the heating plant, t_2 on entrance to the room, and t_3 when leaving the room, while H is the heat requirements in B.Th.U. per hour, and V the volume of air required per hour measured at 60° F.:

$$V = \frac{H}{0{\cdot}0184(t_2 - t_3)}.$$

The heat H_1 to be supplied by the heating plant is

$$H_1 = 0{\cdot}0184V(t_1 - t_0).$$

The calorific value of the fuel required being H_0 and η the firing efficiency of about 50 per cent,

$$H_0 = \frac{H_1}{\eta}.$$

In estimating the heat requirements of a room H, the heat given off by human beings or by lights may be important in the case of crowded or well-illuminated rooms. The heat given off per hour by human beings averages the following values.

	B.Th.U.
Grown man in a state of rest	380
Grown man engaged in light work	470
Grown man engaged in heavy work	560
Old persons	360

Children give off about half this amount. One-third to one-quarter of the heat is used for evaporating the moisture on the skin, which for a grown man amounts to 1·5 oz. per hour in a sparsely occupied room and twice this amount in a crowded room. The production of heat accompanying various kinds of illumination is given in the following table for 1 candle-power hour.

	Fuel Consumption per hour.	Heat Production per hour.
Paraffin oil	0·0073 lb.	145 B.Th.U.
Argand gas burner	3·5 c. ft.	200 ,,
Bray gas burner	4·6 ,,	265 ,,
Incandescent gas mantle	0·75 ,,	25 ,,
Acetylene light	0·2 ,,	20 ,,
Carbon-filament bulb	4·5 watts	15 ,,
Metal-filament bulb	1·2 ,,	5 ,,
Arc light	1·1 ,,	5 ,,

When a room is heated by hot air additional services may be rendered in the removal of carbon dioxide and moisture produced. The amount of carbon dioxide produced by human beings and by illuminating agencies is as follows:

	C. Ft. per hour.
Grown man engaged in manual work	1·25
Grown man at rest	0·7
Youth at rest	0·55
Child at rest	0·35
	C. Ft.
Combustion of 1 c. ft. of coal gas	0·57
Combustion of 1 lb. of paraffin	25·0
Combustion of 1 lb. of stearine	23·0

Some idea of the heat losses from a room can be obtained from the following experimental values for radiation and conduction losses, using walls and floors of different materials. If h is the heat lost in this way, a is a portion of the wall surface in square feet, t_i and t_e the internal and external tempera-

tures to be maintained in degrees Fahrenheit, and k the coefficient of heat transfer,

$$h = \Sigma ka(t_i - t_e) \text{ B.Th.U. per hour.}$$

The coefficient k increases with a reduction in wall thickness but is not inversely proportional to it.

EXTERNAL WALLS

Brick ..	Thickness, inches	4·7	10·0	15·0	25·3	35·4
	k	0·50	0·35	0·27	0·19	0·15
Sandstone	Thickness, inches	11·8	15·7	19·7	27·6	35·4
	k	0·46	0·39	0·35	0·29	0·25
Concrete	Thickness, inches	2·0	3·9	5·9	7·9	11·8
	k	0·71	0·56	0·48	0·42	0·31

INTERNAL WALLS

Brick ..	Thickness, inches	4·7	10·0	15·0	25·3
	k	0·46	0·31	0·25	0·17
Plaster ..	Thickness, inches	1·2	2·0	2·7	3·5
	k	0·66	0·60	0·56	0·50

For floors and ceilings the value of k varies between 0·07 and 0·37 according to the material and the construction, whether solid or hollow. For a tiled roof its value is 1·0, for a board roof 0·46, for a tiled roof with timber underneath 0·32, for a concrete roof asphalted 0·54, and for a corrugated iron roof 2·2. The value of k for doors is about 0·4, for single windows 1·0, and for double windows 0·5. An addition varying between 5 and 20 per cent must be made for outer surfaces in exposed positions. The total heat loss for the whole room can be obtained by adding together that for each portion of the walls. The difference of temperature to be assumed is that between the desired internal temperature and the lowest probable external temperature, and the heating plant must have a capacity equal to this after allowing for the other factors mentioned. Thus the heat requirements of the room $H = h - h_1 - h_2$, where h_1 and h_2 are the heats received from human beings and illumination or engines respectively.

Returning to the supply of this heat by hot air, when a flow within pipes is used, the heat transfer from the hot air through the walls of the pipes to the room air follows the same law as that for the heat loss through the walls of the room, but k depends both upon the air velocity and the temperature difference. If t is the temperature of the hot air within the pipe and t_i the temperature of the room, the following table illustrates the variation in k.

v, ft./sec.		2	4	6	10	20	30
$t - t_i = 36°$ F.	k	0·28	0·46	0·60	0·75	0·98	1·08
$t - t_i = 72°$ F.	k	0·37	0·58	0·73	0·87	1·10	1·21
$t - t_i = 108°$ F.	k	0·41	0·62	0·77	0·92	1·15	1·23

Where the air is allowed to escape into the room, being used also for ventilation, there are the same heat requirements, but it is necessary also to ensure that the air is adequate for the removal of the carbon dioxide and moisture. If the air supplied has a CO_2 content by volume of V_1 per cubic foot (say 0·04 per cent), and it is desired that the CO_2 content in the room should not exceed V_2 per cubic foot (say 0·1 per cent), while the production of CO_2 by human beings and lights is V_3 c. ft. per hour, then the volume of the ventilating air, V c. ft. per hour, must be greater than $V_3/(V_2 - V_1)$. A similar formula will give the volume of air necessary for the adequate removal of the moisture produced, and if the air flow as calculated for the heating requirements is not equal to or greater than either of these volumes a larger flow at a lower temperature must be used. In actual practice the air required for the adequate ventilation of various kinds of rooms is as follows:

Living-rooms	1 to 2 times the volume per hour.
Restaurants	3 to 5 ,, ,, ,,
Kitchens	4 to 5 ,, ,, ,,
Ships cabins	4 to 10 ,, ,, ,,

9. HEATING OF WATER FOR STEAM PRODUCTION

Boilers for the production of steam vary greatly in design according to the amount of steam required, its pressure and temperature, and the

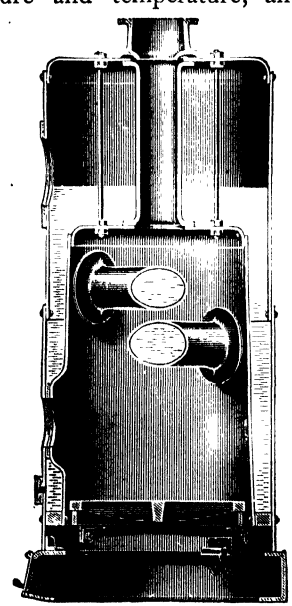


Fig 23 —Vertical Cross-tube Boiler

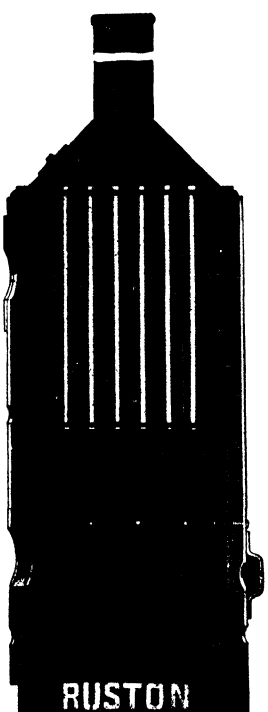


Fig. 24.—Vertical Boiler with Fire Tubes

availability of space and skilled manipulation. They may, however, be grouped into a number of classes for general consideration.

(*a*) Small Steam Boilers

For an output of steam such as is required for a single small steam-engine, boilers of a simple character such as those shown in figs. 23 and 24 are used.

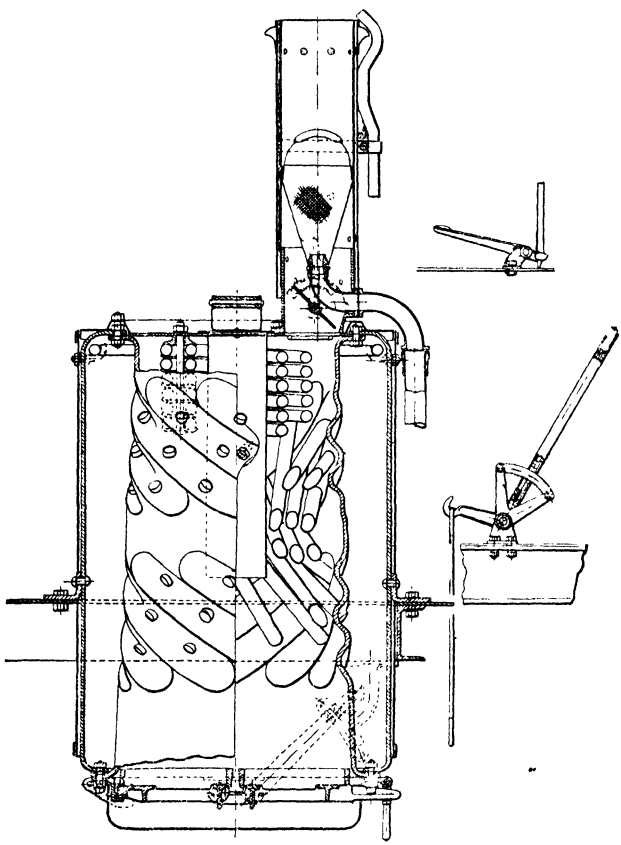


Fig. 25.—Boiler of " Super-Sentinel " Steam-wagon

The former is of the cross-tube type, in which the combustion chamber is surrounded by the water space and water-filled tubes cross the chamber to increase the heating area. In fig. 24 the increase of heating area is produced by fire tubes in considerable number, through which the hot gases pass from the combustion chamber through the water to the flue. An efficient small boiler for a steam-lorry is given in fig. 25. In this design wide fire tubes at a steep inclination pass from the fire-box to the flue. Their arrangement

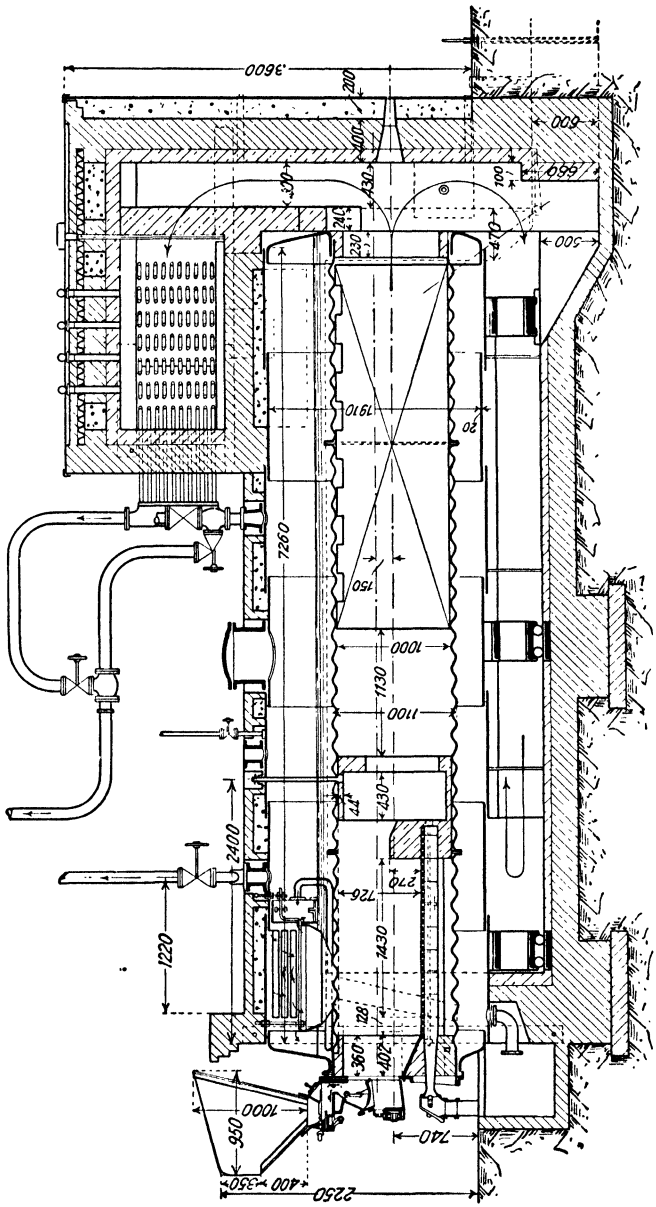


Fig. 26.—Cornish Boiler with a Heating Surface of 560 sq. ft. and a Superheater of 380 sq. ft.; pressure 175 lb. per square inch

tends to assist the water circulation, while they are sufficiently wide to be easily cleaned. The steam is superheated in passing through the horizontal coil of piping shown at the upper portion of the com-

bustion space. The boiler works at a pressure of 230 lb. per square inch.

Small boilers are also constructed of a horizontal type. The efficiency of all small boilers is low, the combustion gases usually escaping at a fairly high temperature and carrying away considerable heat.

(*b*) Steam Boilers with Large Water Capacity

For use in factories and works where considerable space is available and where the steam pressure required is not high, Lancashire and Cornish

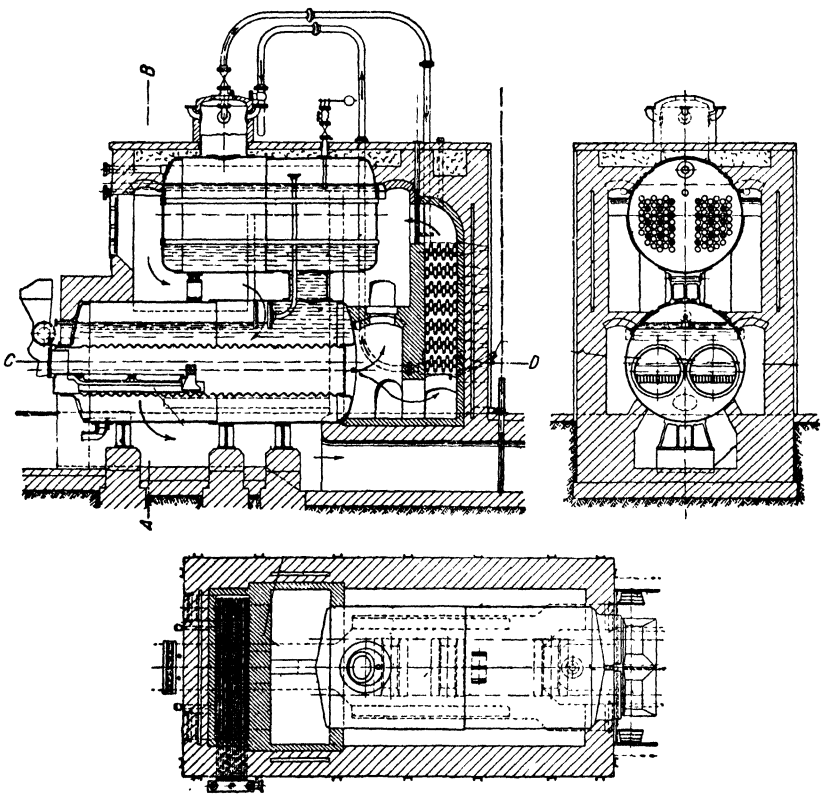


Fig. 27.—Double Boiler

boilers are used. These are simple in design and require little attention in the way of repairs and replacements. Plants of a more efficient character are shown in figs. 26, 27, and 28, in which economizers are attached. These consist of a series of tubes which are heated by the exhaust gases before passing to the flue, and in which the feed water receives a preliminary heating. Boilers of this character are usually worked with a large reserve of yield, so

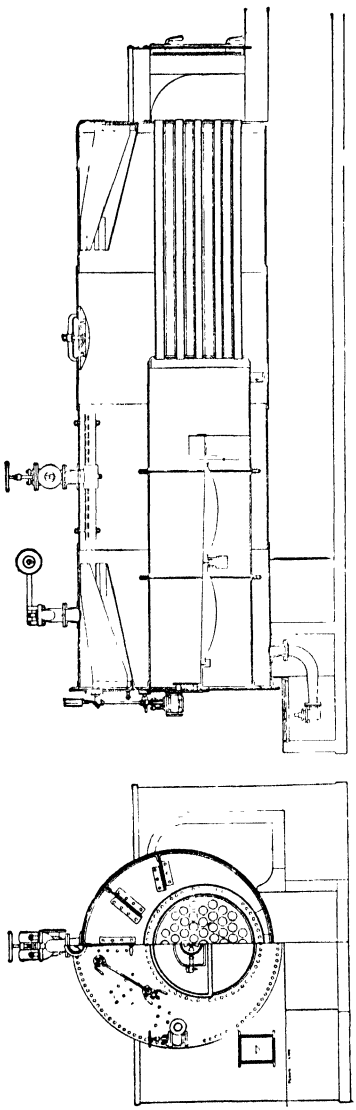


Fig. 28.—Cornish Multitubular Boiler

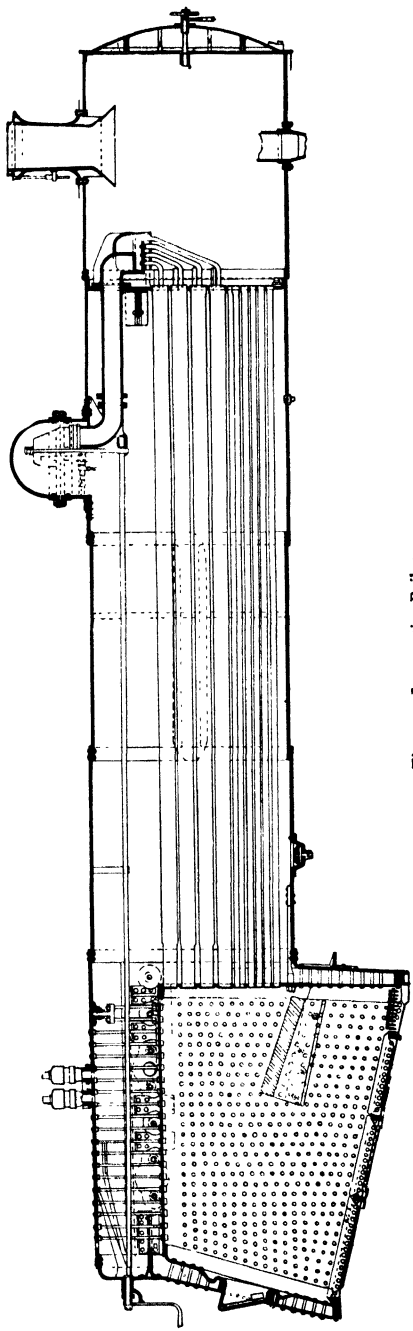


Fig. 29.—Locomotive Boiler

that they are able to supply any sudden increase in steam consumption. Owing to their large dimensions the waste gas temperature can be reduced to a low value, efficient combustion can be maintained, and a high efficiency produced. The furnaces are normally internal, but if low-value fuels are used the combustion space should be extended by using a furnace grate in front of the boiler. With forced draught or steam jets the grate can extend within the boiler.

(*c*) Steam Boilers with Fire Tubes

Some increase in the efficiency of boilers and a reduction in the weight and volume for a given area of heating surface is obtained by the use of

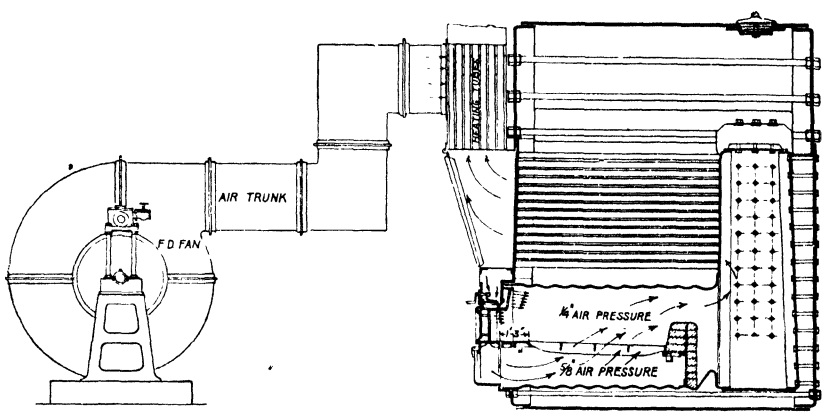


Fig. 30.—Sectional View of Marine Boiler with Regular Forced Draught

From *Engineering*

fire tubes passing through the water space. With these boilers the water capacity in comparison with the total volume remains large. A common example is the locomotive boiler, a recent design being illustrated in fig. 29. Marine boilers for smaller steamers are also of this character. The arrangement of an efficient ship's boiler of this type is shown in fig. 30. Air at a pressure between $\frac{1}{2}$- and $\frac{3}{4}$-in. water pressure is delivered by the fan under the fire grate, the air having been preheated by the exhaust gases. The boiler shown produces steam at a pressure of 10 atmospheres, and an efficiency of 76 per cent is obtained. Of the lost heat 14 per cent is contained by the waste gases as sensible heat, 3 per cent by the ash, and the remaining 7 per cent is due to radiation and conduction losses and to incomplete combustion. High efficiencies can be obtained with these boilers, but the limited combustion space renders the use of high-grade fuels necessary.

(*d*) Water-tube Steam Boilers

For the highest efficiency for high pressures and for plants where space is limited, water-tube steam boilers are used. The water circulates within

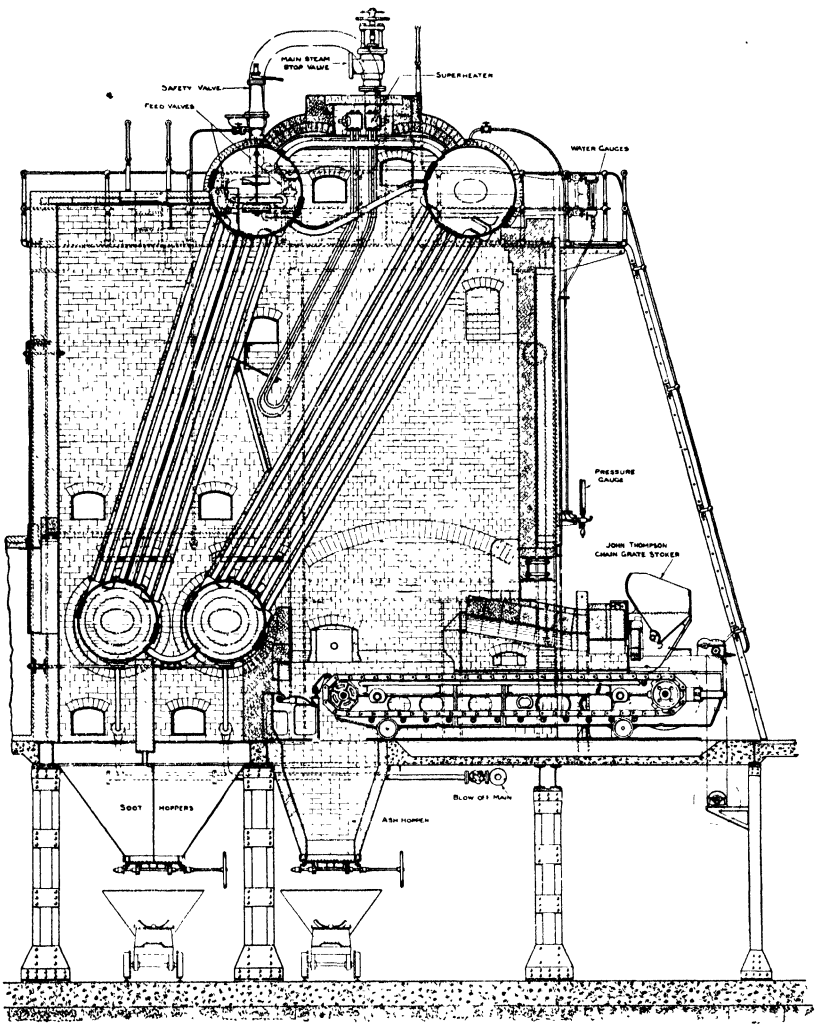


Fig. 31.—Thomson Straight Tube Boiler

Heating surface, 8610 sq. ft.; working pressure, 250 lb. per square inch; evaporation, 45,000 lb. per hour.

the tubes and the combustion gases are without. The tubes are fixed between small cylindrical boilers, but the total water content is small in comparison with the heating area. Hence it is necessary to have a continuous well-

that they are able to supply any sudden increase in steam consumption. Owing to their large dimensions the waste gas temperature can be reduced to a low value, efficient combustion can be maintained, and a high efficiency produced. The furnaces are normally internal, but if low-value fuels are used the combustion space should be extended by using a furnace grate in front of the boiler. With forced draught or steam jets the grate can extend within the boiler.

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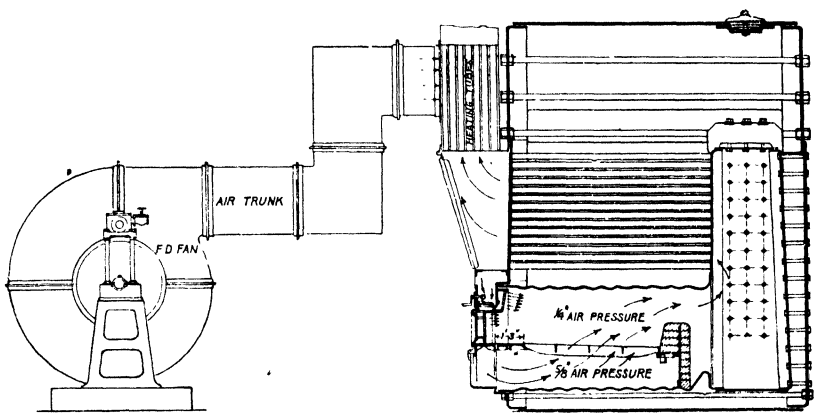


From *Engineering*

Fig. 30.—Sectional View of Marine Boiler with Regular Forced Draught

fire tubes passing through the water space. With these boilers the water capacity in comparison with the total volume remains large. A common example is the locomotive boiler, a recent design being illustrated in fig. 29. Marine boilers for smaller steamers are also of this character. The arrangement of an efficient ship's boiler of this type is shown in fig. 30. Air at a pressure between $\frac{1}{2}$- and $\frac{3}{4}$-in. water pressure is delivered by the fan under the fire grate, the air having been preheated by the exhaust gases. The boiler shown produces steam at a pressure of 10 atmospheres, and an efficiency of 76 per cent is obtained. Of the lost heat 14 per cent is contained by the waste gases as sensible heat, 3 per cent by the ash, and the remaining 7 per cent is due to radiation and conduction losses and to incomplete combustion. High efficiencies can be obtained with these boilers, but the limited combustion space renders the use of high-grade fuels necessary.

(*d*) Water-tube Steam Boilers

For the highest efficiency for high pressures and for plants where space is limited, water-tube steam boilers are used. The water circulates within

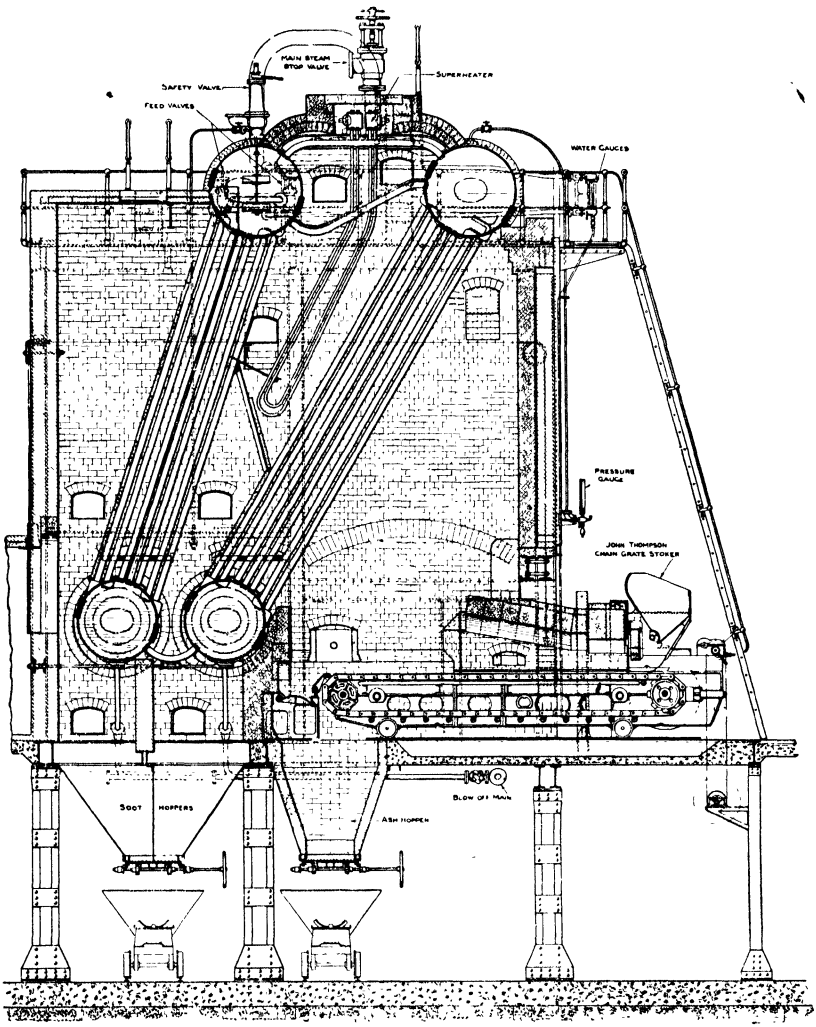


Fig. 31.—Thomson Straight Tube Boiler

Heating surface, 8610 sq. ft.; working pressure, 250 lb. per square inch; evaporation, 45,000 lb. per hour.

the tubes and the combustion gases are without. The tubes are fixed between small cylindrical boilers, but the total water content is small in comparison with the heating area. Hence it is necessary to have a continuous well-

regulated water feed. The combustion space may be large and hence can be adapted for fuels of all kinds, while furnace grates of the most efficient character with mechanical stoking can be applied.

The designs of water-tube boilers for pressures up to 200 lb. per square inch have already been illustrated in Chapter III, figs. 17, 26, 35, 68, and 69.

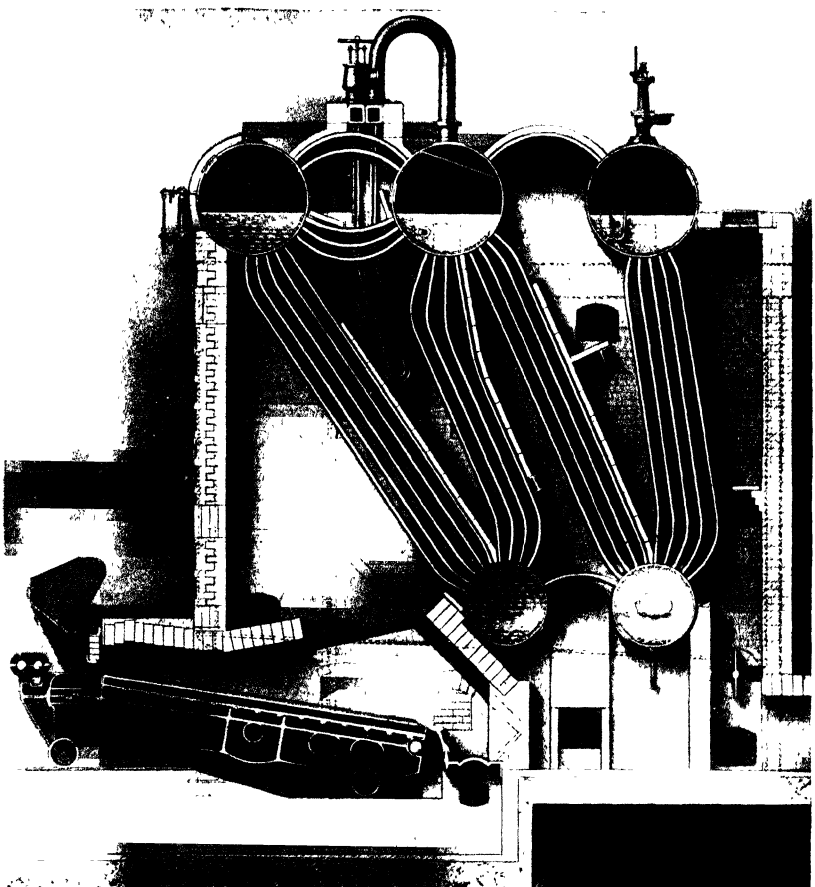


Fig. 32.—Stirling 5-drum Boiler, Superheater and Stoker

Other designs are given in figs. 31 to 34. Considerable variation is exhibited in the general arrangements and the position and shape of the tubes.

(*e*) High-pressure Steam Boilers

Boilers for the production of steam at high pressures up to 1500 lb. per square inch are usually of the same type as those described in the preceding section and differ only in details of construction. To withstand these pressures

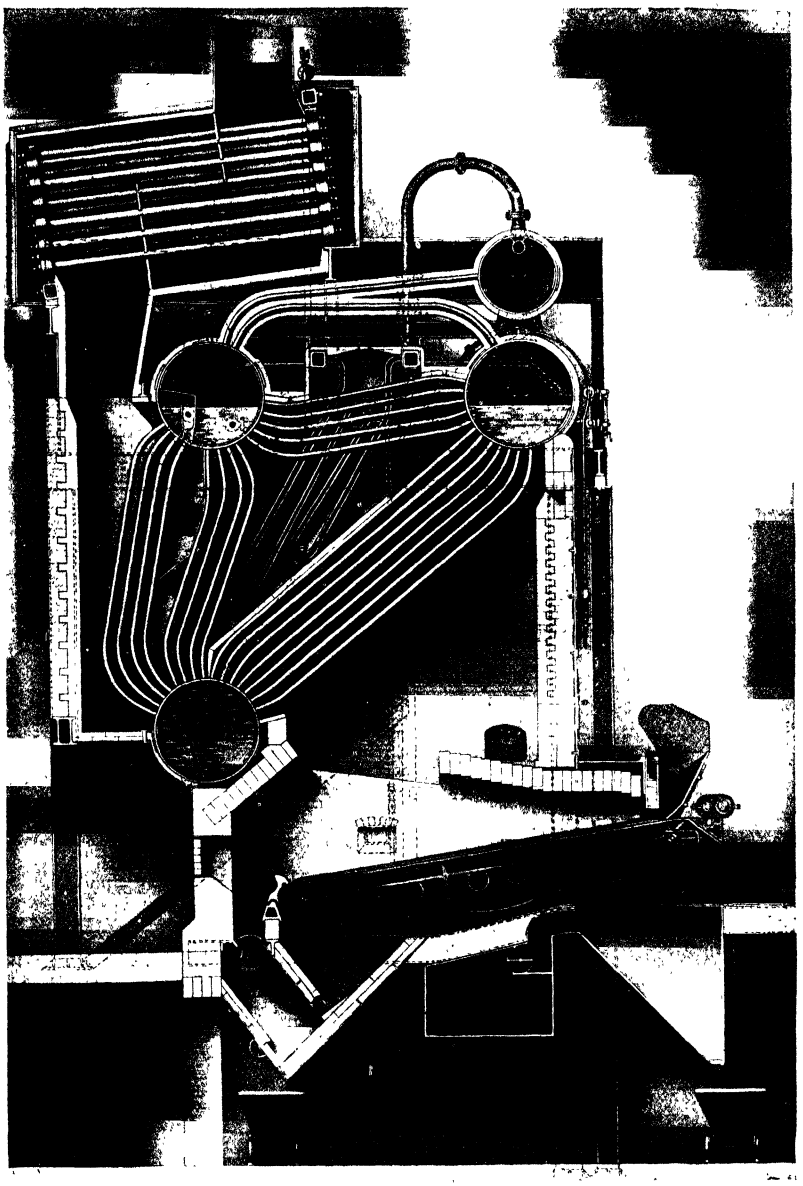


Fig. 33.—Stirling High-duty Tri-drum Boiler, Economizer, and Stoker

it is necessary that there should be no riveted seams. The cylindrical drums are wrought or pressed and of small dimensions. Fig. 35 shows a boiler for steam at 900 lb. per square inch. Its heating surface is 3200 sq. ft., the output

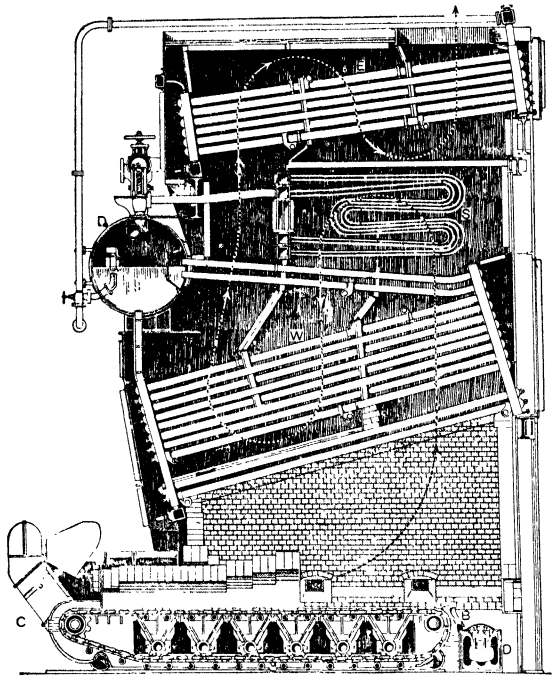


Fig. 34.—Babcock & Wilcox Boiler

B, Dumping-pan. C, Chain-grate stoker. D, Steam-and water-drum. E, Economizer. S, Superheater. W, Water-tubes.

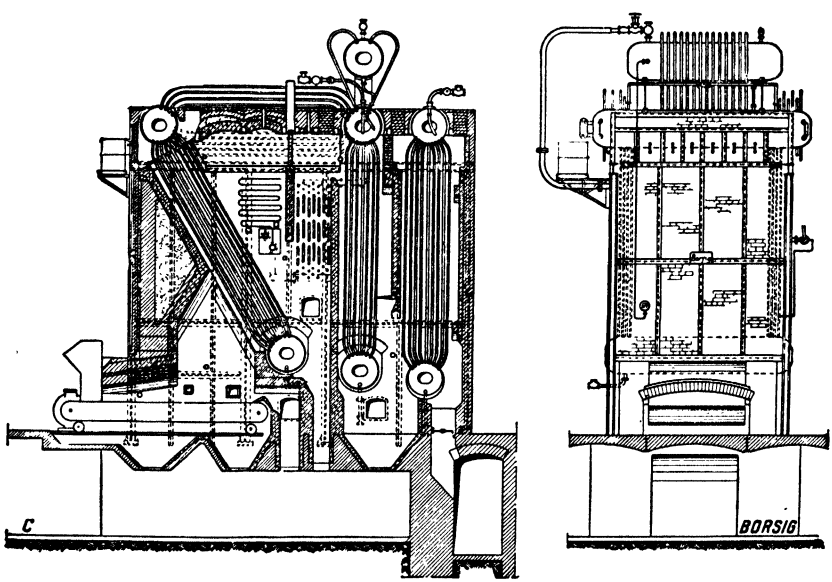


Fig. 35.—Schmidt-Borsig High-pressure Boiler

Heating surface, 3180 sq. ft. Working pressure, 900 lb. per square inch with superheater.

of steam 15,000 lb. per hour, and its capacity 565 c. ft. Fig. 36 shows a boiler for pressures of 1500 lb. per square inch. Another German design is shown in fig. 37, and an American Babcock boiler for pressures of 1250 lb. per square inch is given in fig. 38. The steam output of the latter is 100,000 lb. per hour and its water content 45,000 lb. The boiler heating surface is 15,500

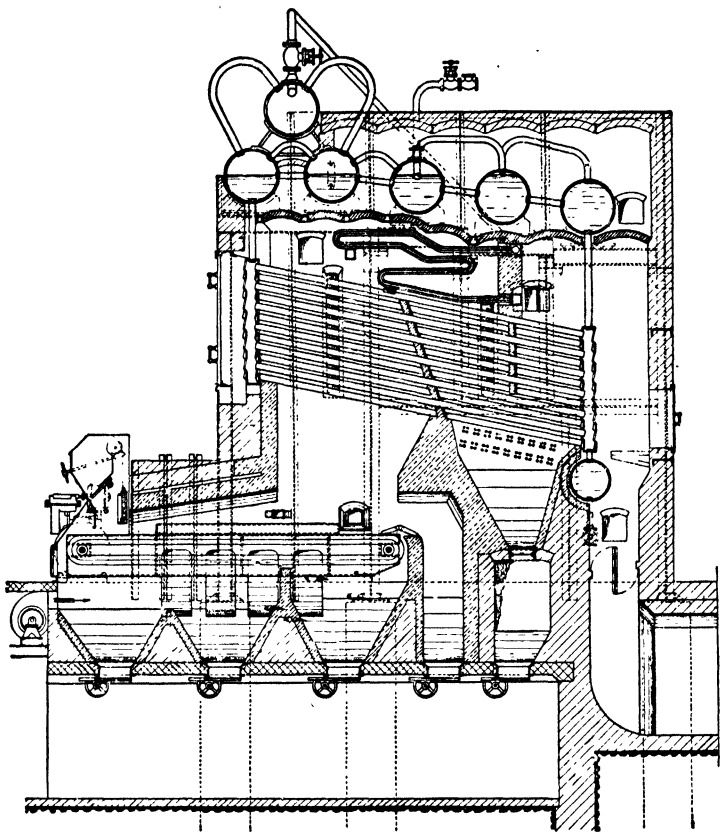


Fig. 36.—Borsig High-pressure Boiler. Working pressure 1500 lb. per square inch.

sq. ft. and the superheater 5300 sq. ft. A counter-current economizer of 9000 sq. ft. is inserted behind the boiler.

A new feature in the construction of high-pressure boilers has been introduced in the Atmos boiler shown in fig. 39 *a* and *b*. The boiler consists of tubes only, 12 in. in diameter and $\frac{3}{4}$-in. thick, made of steel and kept in rotation at 300 r.p.m. by an electric motor. The rotation assists the transfer of the heat to the water owing to the increase in scrubbing action, and it also serves to keep the temperature of the tubes uniform, thus preventing unequal expansion. The water is preheated in the lower banks of tubes shown in fig. 39*b*. Pressures up to 1500 lb. per square inch are used and plants have

been constructed for outputs up to 20,000 lb. of steam per hour. The overall efficiency is over 80 per cent, the chimney losses being 13 per cent. The steam production is about 40 lb. per hour per square foot of heating surface.

The Benson boiler, which is at present only in the experimental stage,

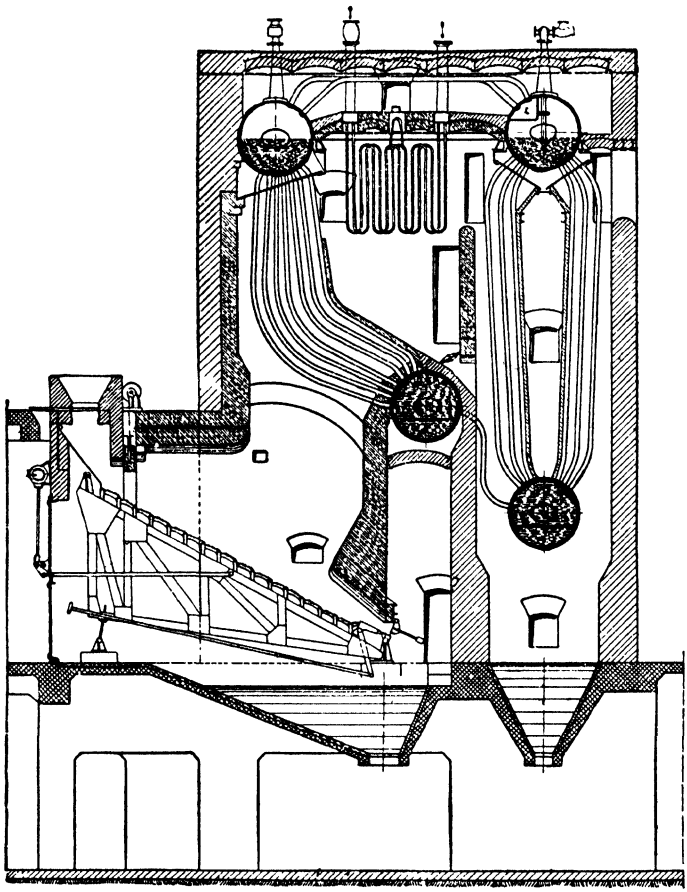


Fig. 37.—Steinmuller High-pressure Boiler

involves a new idea in steam utilization. The steam is produced at a pressure of 3200 lb. per square inch, which is greater than the critical pressure, and hence there is no ebullition and no steam chest is required. Before use in a high-pressure steam turbine it is throttled down to 1500 lb. per square inch.

(*f*) Dimensions and Efficiency of Steam Boilers

In the following table average data are given for boilers of various types. The figures refer to 1 sq. ft. of heating surface.

V is the volume in cubic feet;
V_1 the water contents in cubic feet;
V_2 the steam contents in cubic feet;
A the water surface in square feet; and
W the steam production in pounds per hour.

Type of Boiler.	V.	V_1.	V_2.	A.	W.
Small vertical boiler with cross tubes (water)	0·5 to 1·0	0·05 to 0·3	0·16 to 0·25	0·1 to 0·15	3
Small vertical boiler with many fire tubes	0·2 to 0·3	0·15 to 0·3	0·07	0·05 to 0·1	2·5
Lancashire boiler (single flue) ..	1·6 to 2·3	0·65 to 0·8	0·25 to 0·31	0·25 to 0·3	4 to 5
Cornish boiler (double flue) ..	1·5 to 1·7	0·6 to 0·7	0·25 to 0·33	0·22 to 0·3	4·5 to 6
Large fire-tube boiler	0·6 to 1·0	0·3 to 0·4	0·06 to 0·2	0·12 to 0·15	3·5 to 4·5
Water-tube boiler for ordinary work	0·2 to 0·5	0·15 to 0·3	0·08 to 0·16	0·08 to 0·15	4 to 5
High-performance water-tube boiler	0·2 to 0·5	0·1 to 0·2	0·05 to 0·07	0·02 to 0·03	5 to 7
High-pressure boilers	1·5	0·05 to 0·08	0·1 to 0·13	—	8 to 40

If the heat content of W lb. of steam is H, while that of the same amount of water as supplied to the boiler is h, while η_1 is the boiler efficiency, then the heat to be supplied to the boiler per hour per square foot of heating surface is $\frac{H - h}{\eta_1}$. The efficiency of a modern boiler varies between 78 and 90 per cent, this figure representing the ratio of the heat received by the steam to that contained in the hot gases. It is dependent on the temperature of the feed water, the attainment of which may be considered separately, and the steadiness of the steam production and removal. If η_2 is the furnace efficiency, i.e. the ratio of the heat contained in the gases to the calorific value of the fuel supplied to the grate, then the heat content of the necessary fuel is $\frac{H - h}{\eta_1\eta_2}$. The value of η_2 is also 78 to 90 per cent. If the grate has an area of a_2 sq. ft. with a loading of w_2 lb. of fuel per square foot per hour, the calorific value of which is h_2 B.Th.U. per pound, while the heating area of the boiler is a_1 sq. ft.,

$$w_2 a_2 h_2 = \frac{H - h}{\eta_1\eta_2} a_1.$$

The fuel burnt per hour, which for a given production must obviously be inversely as the calorific value, is given by

$$w_2a_2 = \frac{(H - h)a_1}{\eta_1\eta_2h_2}.$$

A comparison of the performance of different boilers necessitates considera-

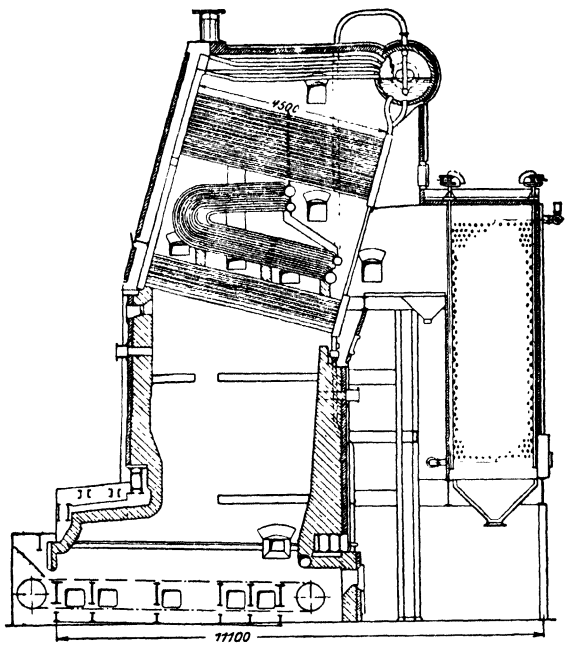


Fig. 38.—An American Babcock Boiler
Working pressure, 1250 lb. per square inch. Steam output, 100,000 lb. per hour

tion of the kind of steam produced. A conventional method of comparison is to introduce a factor proportional to the heat of production of 1 lb. of steam. Thus to transform 1 lb. of water at 32° F. to steam at atmospheric pressure and 212° F. requires 1150 B.Th.U. The production of steam per pound of fuel is

$$\frac{Wa_1}{w_2a_2},$$

and as the heat required per pound is $(H - h)/W$, the performance of the plant is represented by

$$\frac{Wa_1}{w_2a_2} \cdot \frac{H - h}{1150W} = \frac{(H - h)a_1}{1150w_2a_2}$$
$$= \frac{\eta_1\eta_2h_2}{1150}.$$

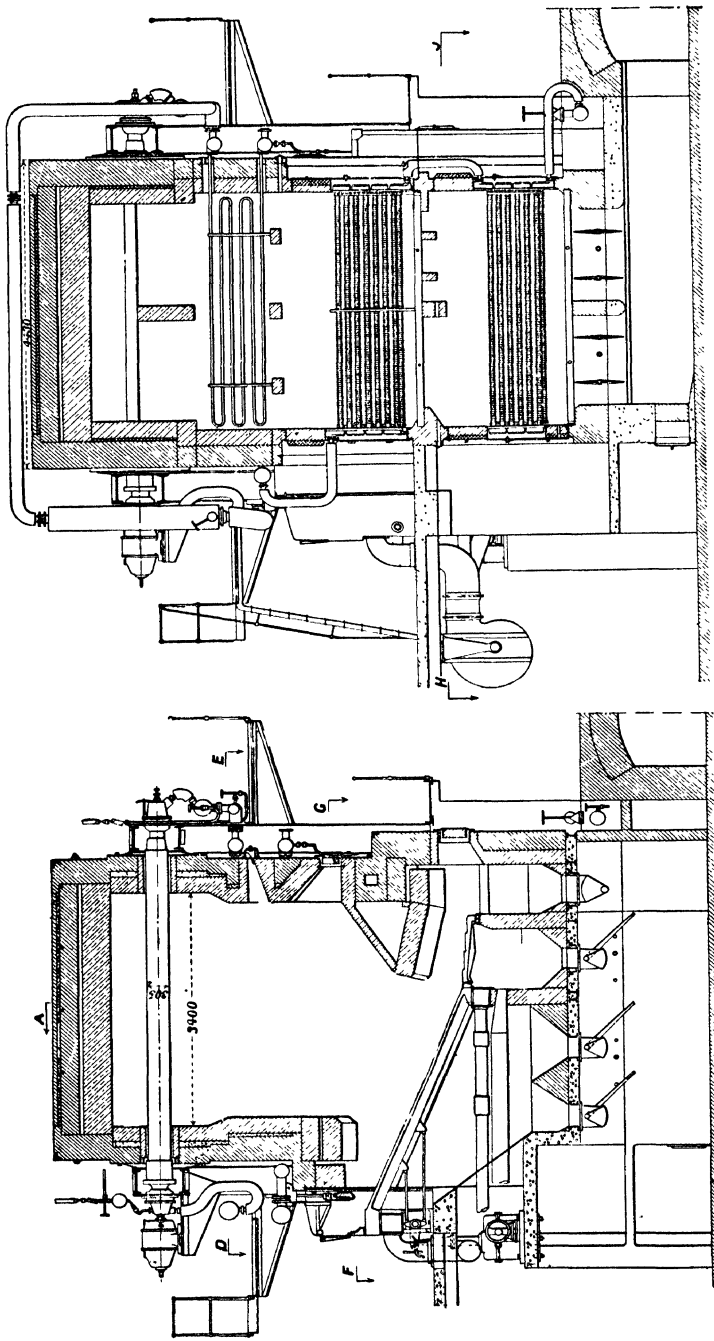


Fig. 39.—The "Atmos" High-pressure Boiler

The amount of steam produced per pound of fuel varies from 8 lb. with mineral coal of 13,500 B.Th.U. per pound to 2·5 lb. with crude brown coal of 4000 B.Th.U. per pound. The fuel burnt per square foot of heating surface of the boiler varies from 0·2 lb. per hour with very slow combustion to 1 lb. per hour with vigorous production.

The feed water may be preheated either by the exhaust gases or by exhaust steam from the steam plant after its utilization. If S is the heating surface of the preheater in square feet, while Wa_1 is the feed water required in pounds per hour, t_1 and t_2 the initial and final temperatures of the heating medium, and t_3 and t_4 those of the feed water in degrees Fahrenheit,

$$\frac{S}{Wa_1} = \frac{1}{K[(t_1 - t_3) - (t_2 - t_4)]} \log_e \frac{t_1 - t_3}{t_2 - t_4},$$

$$\text{or approximately } \frac{2(t_4 - t_3)}{K[(t_1 - t_3) - (t_2 - t_4)]},$$

where K is a constant varying between 0·5 and 1·5. The heat yield of preheaters or economizers represents a saving of fuel, as the heat would otherwise have to be supplied by the direct heat from fuel, while the exhaust gases or steam would pass to waste at a higher temperature.

The efficiency of the heat transfer in boilers depends upon the cleanliness of the surfaces both inside and out. The inside must be kept free from slime and scale. In water-tube boilers the former may collect in the lower drums, which can be easily cleaned, but purification of the water is a more satisfactory method for the elimination of both deposits. The heat conductivity of scale is only $\frac{1}{30}$ to $\frac{1}{50}$ that of iron, and it is very essential that it should not be allowed to accumulate. Deposits of soot and ash on the outer surface must also be avoided. The continuous and automatic feed of water according to the steam withdrawal is also of importance.

Purification of the water according to the character of the water supply consists of both mechanical and chemical processes. The former includes the removal of suspended impurities by settling tanks and filters and the removal of oil by separators and filters. Removal of salts in solution is a chemical process and is very necessary. Unless removed, sulphates are deposited as a scale within boilers owing to the evaporation of the solvent water. Carbonates of calcium and magnesium are precipitated when the carbon dioxide is expelled from solution. Chlorides are decomposed at high pressures and temperatures and give hydrochloric acid, which corrodes the boiler. Carbonate of lime reacts at high steam pressures with magnesium sulphate and magnesium chloride as follows:

$$CaCO_3 + MgSO_4 = MgO + CaSO_4 + CO_2,$$
$$CaCO_3 + MgCl_2 = MgO + CaCl_2 + CO_2.$$

The MgO developed from $MgSO_4$ is insoluble and cakes, while the MgO from $MgCl_2$ is separated in flakes which do not cake.

The removal of the dissolved substances is accomplished by many

methods of precipitation, the following processes being the most common:

(*a*) With slaked lime for the removal of carbonates:

$$CaH_2(CO_3)_2 + Ca(OH)_2 = 2CaCO_3 + 2H_2O.$$

(*b*) With soda for precipitation from sulphates and for bicarbonates:

$$CaSO_4 + Na_2CO_3 = CaCO_3 + Na_2SO_4.$$

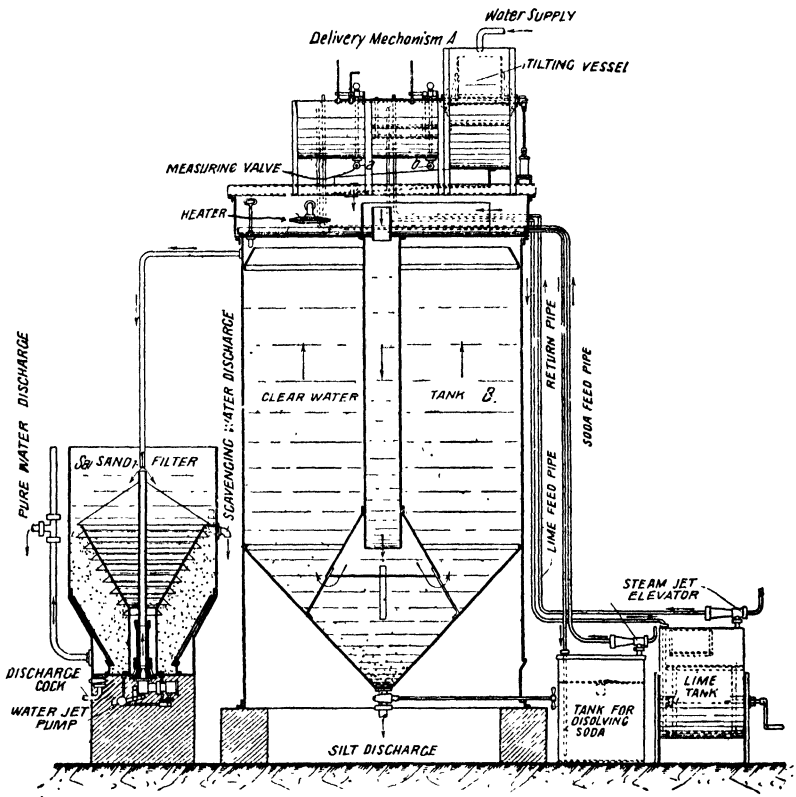


Fig. 40.—Feed Water Purification

(*c*) With caustic soda only or with caustic soda and slaked lime. The caustic soda, however, is more expensive than the other chemicals:

$$CaH_2(CO_3)_2 + 2NaOH = CaCO_3 + Na_2CO_3 + 2H_2O.$$

(*d*) Precipitation with zeolites (basic silicates of aluminium) or permutite, as the artificial compound is named:

$$Na_2Al_2Si_2O_8 + CaH_2(CO_3)_2 = CaAl_2Si_2O_8 + 2NaHCO_3$$
$$Na_2Al_2Si_2O_8 + CaSO_4 = CaAl_2Si_2O_8 + Na_2SO_4.$$

The permutite can be regenerated in a very simple way by treatment of the calcium compound with a strong solution of common salt.

$$CaAl_2Si_2O_8 + 2NaCl = Na_2Al_2Si_2O_8 + CaCl_2.$$

Similar reactions occur with magnesium salts.

Apparatus for the purification of boiler feed water is shown in figs. 40 and 41. In the latter a measuring apparatus (1) is inserted for metering the water that is to be purified. A water wheel (2) works the stirring apparatus in the lime saturator (3). The soda bank is shown at (4) and the mixing vessel for lime, soda, and water at (5). The water flows through (6) to (7) and rises upwards between spiral surfaces (8) where the solids are deposited.

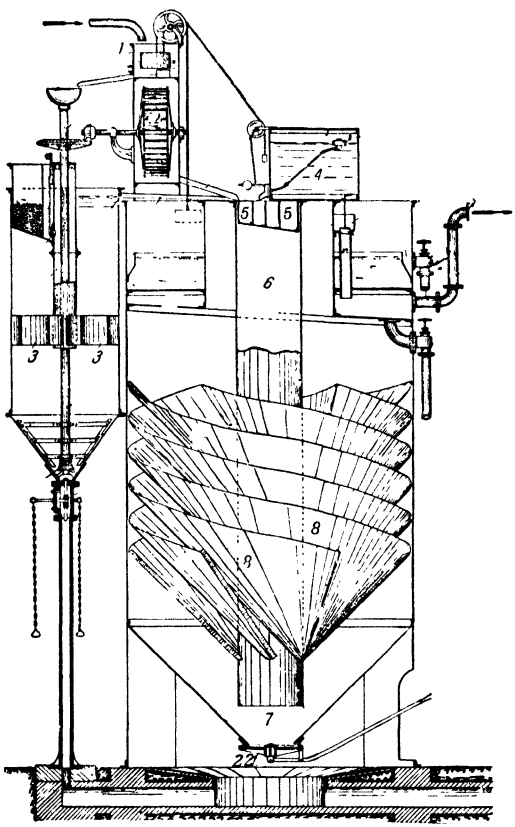


Fig. 41.—Apparatus for Water Purification

The object of the superheater is to vaporize any water with the steam which has been carried over mechanically or been formed by condensation, to ensure sufficient heat to prevent any condensation between the boiler and the place of use, and in cases where the steam is used for steam-engines or turbines to obtain a higher engine efficiency. This latter purpose, which is now of great importance, is considered in a later chapter. The velocity of the steam within the superheater and the dimensions must be arranged to ensure the amount of superheat required without any excess. If the heat is not withdrawn with sufficient rapidity and the steam is at a higher temperature than that for which the superheater was designed, strains and breakages may arise in the tubes due to overheating. The superheaters are placed in the combustion gases where the temperatures are 800° F. to 1200° F. The steam velocity within the tubes ranges from 30 to 50 ft. per second, and the heat transfer for 1° F. between the hot gases and steam ranges between

8 and 12 B.Th.U. per square foot per hour. The situation of superheaters has been shown in several of the preceding diagrams. Fig. 42 represents a boiler heated by the exhaust gases from a generator furnace. The steam is carried from the top of the boiler to a superheater at the right, where the exhaust gases at their initial and maximum temperature serve for the super-heating.

If Wa is the steam required in pounds per hour;
x the wetness;
L the latent heat;
C_p the specific heat;
a_3 the area of superheater in square feet;
H_1 the heat required for superheating in B.Th.U. per hour;
t_0 the saturation temperature;
t the temperature of superheated steam;
k the mean coefficient of heat transfer; and
Δt the mean temperature difference between steam and gases;

$$H_1 = Wa_1[xL + C_p(t - t_0)] = ka_3\Delta t.$$

Allowing for conduction and radiation losses the heat to be supplied by the exhaust gases is

$$H_2 = \frac{H_1}{\eta},$$

where η is the superheater efficiency. The fall in temperature of the exhaust gases will be inversely as the quantity and will depend also upon the calorific value remaining in the gases as well as their sensible heat. The calorific value decreases with the CO_2 content, i.e. with the completeness of the previous combustion.

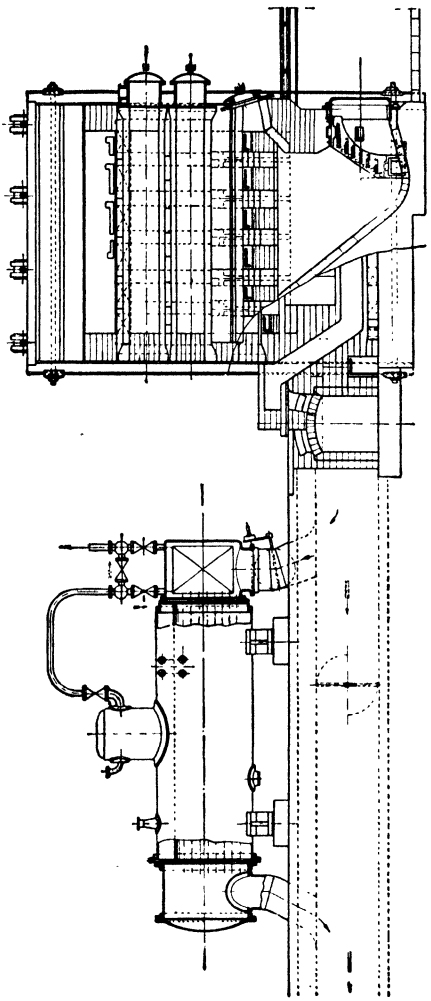


Fig. 42.—Exhaust Heat Boiler Plant placed behind a Semi-generator Furnace

As an example of the data involved in the construction and working of an average-sized large boiler, the following table is given showing the details and results of a heat balance test.

Boiler system, water-tube boiler	
Boiler heating surface	3240 sq. ft.
Normal boiler pressure	190 lb./sq. in.
Grate surface	86·5 sq. ft.
Superheater surface	1060 „
Economizer surface	1960 „

I. *Test Data*

1.	Duration of experiment	593 min.
2.	Total coal consumption	24,750 lb.
3.	Residue in ash and slag	9·5 per cent.
4.	Total consumption of feed water	169,500 lb.
5.	Mean steam pressure	184 lb./sq. in.
6.	Mean temperature of feed water entering economizer	52° F.
7.	„ „ „ of feed water leaving economizer	190° F.
8.	„ „ of superheated steam	579° F.
9.	„ „ in boiler house	68° F.
10.	„ „ of gases in fire chamber	2010° F.
11.	„ „ in front of superheater	1078° F.
12.	„ „ behind superheater	907° F.
13.	„ „ in front of economizer	665° F.
14.	„ „ behind economizer	474° F.
15.	Mean CO_2 contents of gases at the damper	12·6 per cent.
16.	„ O_2 „ „ „	6·8 per cent.
17.	Strength of draught over the grate	0·2 in. water.
18.	„ „ at the damper	0·7 in. water.

II. *Performance and Consumption*

1.	Coal consumption per hour			2500 lb.
2.	Coal consumption per hour for 1 sq. ft. grate surface			29·0 lb.
3.	Production of steam per hour			17,400 lb.
4.	Production of steam per hour for 1 sq. ft. heating surface			5·32 lb.
5.	Gross vaporization: 1 lb. coal produces following steam			6·96 lb.
6.	Total heat per 1 lb. saturated steam			1200 B.Th.U.
7.	Temperature of the saturated steam			377° F.
8.	Superheating			202° F.
9.	Specific heat of the steam			0·54
10.	Total heat for 1 lb. superheated steam			1310 B.Th.U.
11.	Useful heat yield per 1 lb. steam by the	Economizer (190−32)−(52−32)	=	138 B.Th.U.
12.		Boiler, 1200−(190−32)	=	1042 „
13.		Superheater, 1310−1200	=	110 „
				1290 „
14.	Total steam heat			1290 „
15.	Useful heat yield per 1 lb. coal, 6·86 × 129		=	8850 „
16.	Calorific value of 1 lb. coal			11,000 „
17.	Furnace efficiency			90·0 per cent.
18.	Boiler efficiency			89·5 „
19.	Thermal efficiency			80·5 „
20.	Thermodynamic efficiency			93·0 „

III. *Heat Balance*

1. Useful heat yield in steam	8850 B.Th.U.	= 80·5	per cent.
2. Chimney loss	1300 „	= 11·8	„
3. Remaining losses	850 „	= 7·7	„
4. Calorific value of coal	11,000 „	= 100	„

As a general rule experiments for determining the efficiency of a boiler plant should extend over several days, while the stoker should be allowed to attend to the furnace as usual. This is the only means of obtaining the true efficiency for a continuous running. In order to detect faults, a special set of experiments should be made extending over 8 to 10 hr. for comparing single values. In these experiments it is important that the type of coal, the grain of the coal, and the calorific value of the coal should be the same.

(*g*) Electric Boilers

Steam can be generated by electric heating on a small scale. The plant requires very little attention, is clean and very efficient, but also very costly.

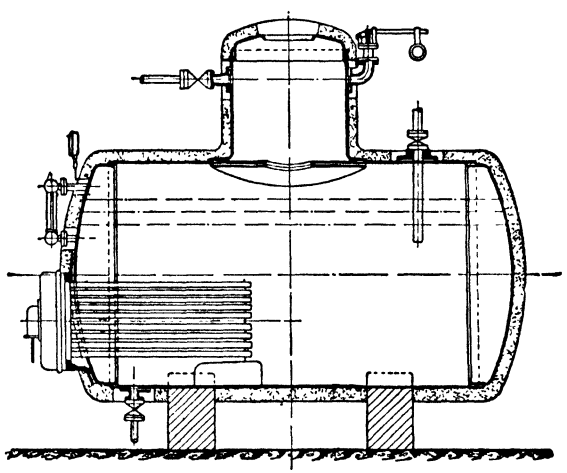


Fig. 43.—Electric Steam Boiler with Immersed Resistances

Two methods are used. The more common method is by high-resistance coils or grids through which either direct or alternating current can be passed. If the coils are immersed in the water practically 100 per cent of the heat may be transferred to it, and the heat losses are those from the boiler surfaces and connexions only. In some cases the heating resistances are outside the boiler, and losses occur similar to those of furnaces but much smaller in amount with an efficient design. The resistances are usually made of alloys of nickel and chromium, which can be heated continuously for lengthy periods without oxidation. They may either be bare or may be

embedded just below the surface of the fireclay on which they are mounted. Bare wires and high temperatures are necessary where radiant heat is utilized. Recently rods similar in appearance and fragility to graphite and composed of a combination of carbon and silicon have been used. They have a very high specific resistance, and a rod of short length and appreciable thickness can dissipate a large amount of heat. A boiler heated by high resistances immersed within the water is shown in fig. 43.

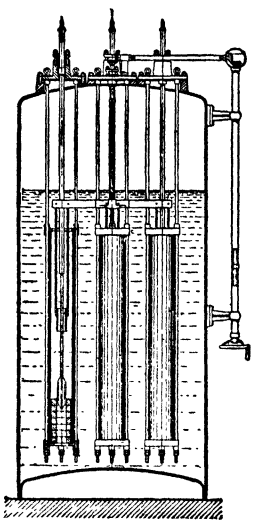


Fig. 44.—A.E.G. Electrode Boiler for Alternating Current; Adjustable Electrodes

A second and less usual method is that of electrode heating. Alternating current is used and is conducted into and out of the water by large electrodes. A design of this character is shown in fig. 44. As with the immersed resistances, the losses are those from the boiler only and amount to between 6 and 10 per cent.

10. REFRIGERATION

In the process of refrigeration, heat is conducted from a colder to a warmer medium and mechanical work must be performed. In many cases the heat is first abstracted from a salt solution which is circulated, and in turn cools the chamber or body requiring to be cooled. The specific heat of a salt solution at 32° F. is as follows:

Strength of solution	0%	4%	8%	12%	16%	20%	24%
Specific heat ..	1·006	0·944	0·904	0·869	0·840	0·814	0·789

The principal types of refrigerating plants are: (*a*) compression machines, (*b*) absorption machines, and (*c*) evaporative refrigerators.

(*a*) Compression Refrigerating Machines

If a gas is compressed the work of compression results in an increased temperature, and heat must be abstracted to restore it to its original temperature. If now the gas is allowed to expand, it will fall in temperature, and by the withdrawal of heat from its surroundings it can be made to cool them. Refrigerating machines using air are built on this principle, but as specific heat only is involved large volumes of air are required and bulky compressors. If the working medium is carbon dioxide, ammonia, or sulphur dioxide, liquefaction may be produced during compression and vaporization during expansion, and the heat transfer for a given weight of vapour is greatly increased. The working cycle is shown diagrammatically on the temperature-entropy diagram, fig. 45. The vapour drawn from the evaporator may contain some liquid, as shown at the point *a*, and is compressed adiabatically along

the line *ab*, being dry and superheated at *b*. It is then cooled and liquefied at constant pressure along the line *bcdf*. Passing through a throttle valve to a region of lower pressure, it falls in temperature but is evaporated by the heat of its surroundings along the line *fga*. In flowing through the throttle a certain amount of work is lost and the effective refrigerating output is reduced, which could be avoided by obtaining a fall in pressure by expansion only. The use of the throttle, however, simplifies the mechanism. If an expansion chamber is used and the liquid is allowed to fall to the lower temperature by adiabatic expansion, some will be vaporized as shown by the line *fe*, and the remainder by abstraction of heat from the surroundings along the line *ea*. In this ideal case the heat given up during compression and condensation is represented by the area *bfmq* = H_1, the work done is *abcdfea* = W, and the heat absorbed during expansion and evaporation is *aemq* = H_2 = H_1 − W. The output factor

$$\xi = \frac{H_2}{W} = \frac{H_1}{W} - 1.$$

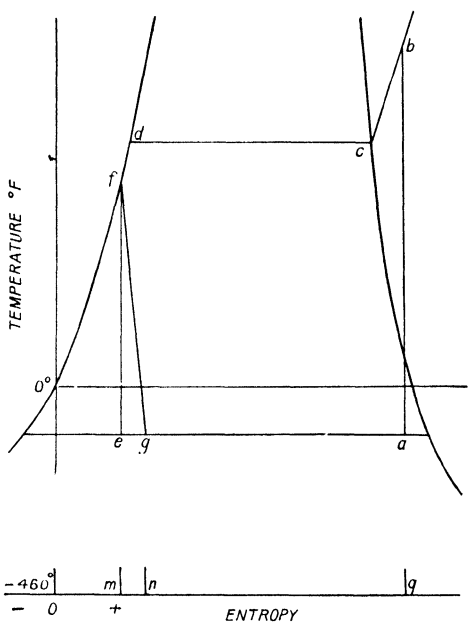


Fig. 45.—Working Diagram

With the use of the throttle the heat absorbed during vaporization is given by the area *agnq* = H_3, while the work done is *abcdfga* = W_1,

$$\xi_1 = \frac{H_3}{W_1}.$$

Now the ratio H_3/H_2 = *ag*/*ae*, while the ratio of W_1/W is greater than this. Hence the actual output factor is less than the ideal output. Furthermore, for a cycle working with a small pressure difference, both H_2 and W are practically proportional to the latent heat represented by the length of the horizontal lines. The effect of expanding through a throttle depends on the specific heat, and would hence be proportionately greater where the latent heat is small, i.e. near the critical point.

As an example of the quantities involved, the following figures refer to the cooling of a room to 0° F., the available water for condensing the ammonia vapour being at 60° F. The water itself will rise in temperature, and furthermore at the exit from the condenser the ammonia will still be, higher in temperature than the water unless the condenser is extremely large. Hence a temperature of 80° F. may be taken as the temperature of the ammonia

when compressed and condensed. Similarly it will be necessary to reduce the vaporized ammonia to −25° F. if the room is to be reduced to 0° F. If the vapour as drawn from the vaporizer for compression contains 5 per cent of liquid:

At *a*, temperature = −25° F.; pressure = 15·6 lb. per square inch; entropy = 1·163; total heat = 501·7 B.Th.U. per pound.
At *c*, temperature = 80° F.; pressure = 154 lb. per square inch; entropy = 1·035.
At *b*, temperature = 199° F.; pressure = 154 lb. per square inch; entropy = 1·163; total heat = 632·7 B.Th.U. per pound.
At *d*, temperature = 80° F.; pressure = 154 lb. per square inch; entropy = 0·102; total heat = 53·6 B.Th.U. per pound.
At *f*, temperature = 70° F.; entropy = 0·081; total heat = 42·1 B.Th.U. per pound.
At *e*, temperature = −25° F.; pressure = 15·6 lb. per square inch; entropy = 0·081; total heat = 31·4 B.Th.U. per pound.
At *g*, total heat = 42·1 B.Th.U. per pound; entropy = 0·106.
With the expansion chamber, H_2 = 470 B.Th.U. per pound; W = 120·4 B.Th.U. per pound; ξ = 3·91.
With the throttle, H_3 = 460 B.Th.U. per pound; W_1 = 119·3 B.Th.U. per pound; ξ_1 = 3·86.

Now the work which could be obtained in the expansion is usually wasted, so that the actual work required is that of the compression. In the compressor there is waste due to friction and, assuming a compressor efficiency of 90 per cent, the heat required per pound of ammonia is

$$\frac{632{\cdot}7 - 501{\cdot}7}{0{\cdot}9} = 145{\cdot}5 \text{ B.Th.U.}$$

The realized value of the refrigerating factor is hence

$$\xi_2 = \frac{460}{145{\cdot}5} = 3{\cdot}16.$$

Cooling water required per pound $= \frac{632{\cdot}7 - 42{\cdot}1}{10} = 59$ lb., assuming that the water is allowed to rise 10° F.

Horse-power required for a flow of 1 lb. of ammonia per minute $= \frac{145{\cdot}5 \times 778}{33{,}000} = 3{\cdot}53.$

As only a small rise in temperature of the cooling water is permissible for an efficient condensation of the ammonia, the utilization of the heat received by the water can only be performed in an advantageous manner in conjunction with other heat production. It may be supplied as preheated water to plants in which further heating takes place. If it can be utilized in this or a similar manner the combined plant can be very efficient. If 1000 lb. of steam per hour are available for heating a room and 1100 B.Th.U. can be abstracted per pound, the heat available per hour is 1,100,000 B.Th.U.

Instead of using the steam directly it may be utilized to drive a steam-engine of about 60 h.p., which in turn would actuate the compressor of a

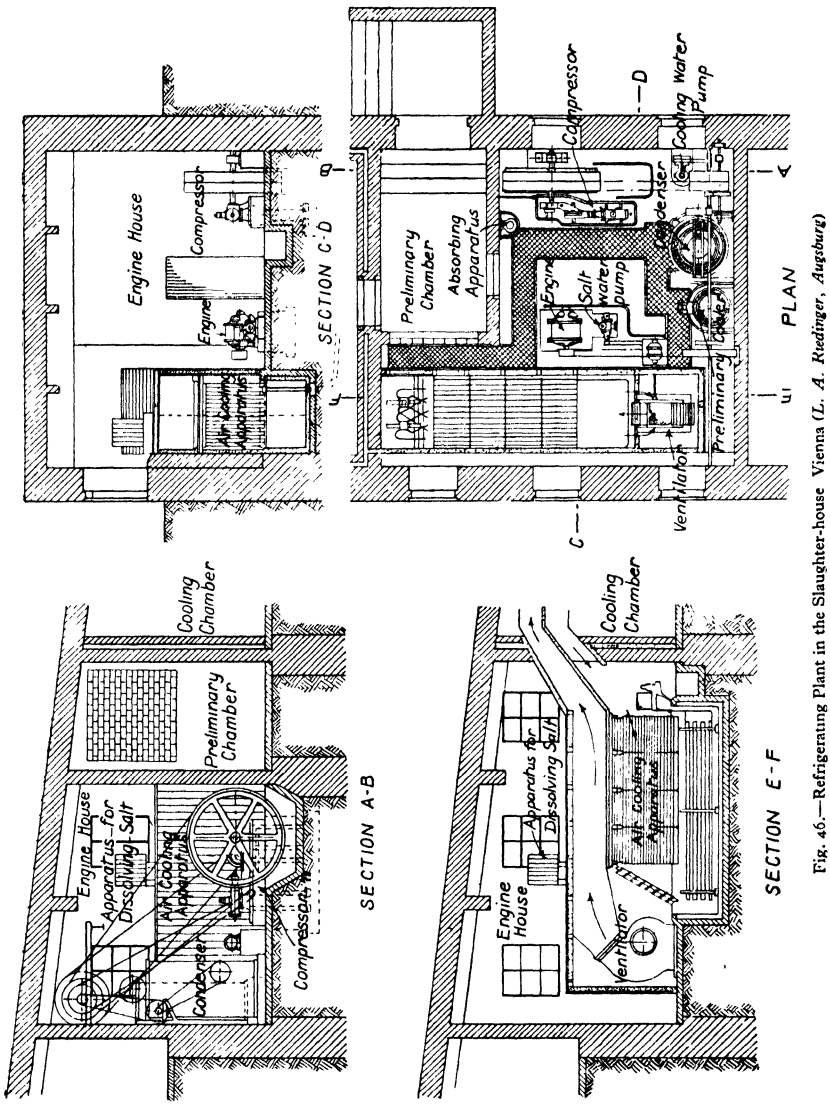


Fig. 46.—Refrigerating Plant in the Slaughter-house Vienna (*L. A. Riedinger, Augsburg*)

refrigerator. If the efficiency of the steam-engine is 12½ per cent the work done per hour on the compressor is 137,000 B.Th.U., while 800,000 B.Th.U. remain in the exhaust steam. If the refrigerating factor realized is 3·16 per cent as in the example, the heat abstracted in the condenser of the refrigerator

is 4·16 × 137,000 = 570,000 B.Th.U. From the exhaust steam and the condenser water of the refrigerator there is thus 1,370,000 B.Th.U. available. This is greater than the original amount in the fresh steam, but it is at a lower temperature and the efficiency of its utilization may be affected by that fact. In addition to increasing the heat available a refrigeration of 3·16 × 137,000 = 430,000 B.Th.U. has been obtained, this abstraction of heat being, of course, the cause of the increased heat elsewhere.

The general arrangement of a refrigerating plant is shown in fig. 46. In this case carbon dioxide is the medium used. The calculations are similar but the pressures required are higher, 60 atmospheres being commonly used as compared with 10 atmospheres for ammonia and 3½ atmospheres with sulphur dioxide. The output of refrigerating machines varies according to the temperature conditions, the following being comparative figures for a room temperature of 18° F. (i.e. 14° below freezing), using cooling water at 68° F.

	Refrigeration per Pound.	Refrigeration per Horse-power Hour.
CO_2	70 B.Th.U.	15,000 B.Th.U.
NH_3	500 ,,	21,500 ,,
SO_2	140 ,,	21,500 ,,
Air	35 ,,	3,000 ,,

(*b*) Absorption Refrigerating Machines

In the compression machines the condensation of the vapour is attained by a mechanical compressor and the abstraction of heat. In the absorption process a strong solution of ammonia is heated by steam, and the expulsion of the gas causes the necessary rise in pressure sufficient to condense it in a connected and cooled chamber. As in the preceding method, the liquid under pressure is allowed to expand through a throttle to a place of lower pressure and to evaporate there, abstracting heat from a bath of salt solution which surrounds it. The vapour can then be again dissolved in the water from which it was expelled and the process repeated. With the same requirements as in the example of the preceding section the ammonia vapour when condensed will be at 80° F. and its pressure 154 lb. per square inch. If the ammonia solution is heated by steam at a gauge pressure of 20 lb. per square inch its temperature will reach 259° F. Ammonia gas is expelled until the concentration in the solution reaches a minimum value determined by the temperature and pressure. This value is given as a percentage by weight by the following approximate formula:

$$\frac{T_s}{T} = 0{\cdot}00471x + 0{\cdot}655,$$

where T is the absolute temperature and T_s is the boiling-point of the gas

corresponding to the pressure, in this case 80° F. + 460° F. Hence $x = 19{\cdot}1$ per cent. After the condensed ammonia gas has passed through the throttle and been vaporized under the reduced pressure of 15·6 lb. per square inch it is at the corresponding boiling-point of −25° F. The reverse process of dissolving it in water at 60° F. will proceed to a maximum concentration given by the same formula:

$$\frac{-25 + 460}{60 + 460} = 0{\cdot}00471x + 0{\cdot}655,$$

$$x = 38{\cdot}5 \text{ per cent.}$$

This solution can then be returned to the boiler for renewed expulsion of half of its ammonia. Some small corrections are necessary for pressure lost in the connecting pipes. Instead of using fresh water for the solution of the ammonia the weakened liquid from the boiler is used. Some water is vaporized in the boiler at the same time as the ammonia gas is expelled, and it is necessary to separate this out by preliminary cooling.

(*c*) Evaporative Refrigerating Machines

If a portion of a liquid is caused to evaporate under such conditions that the latent heat must be supplied by the remaining liquid, then the latter

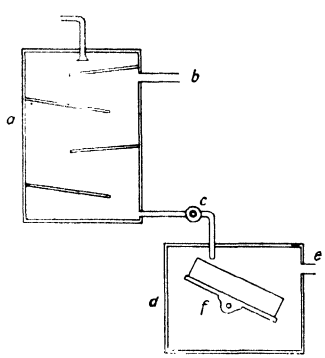


Fig. 47.—Evaporative Refrigerating Machine

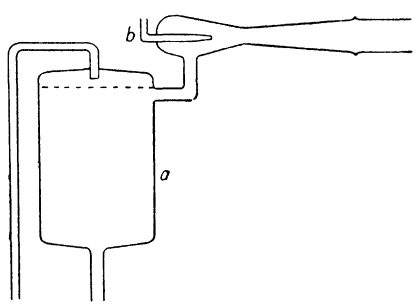


Fig. 48.—Evaporative Refrigerating Machine

may be reduced to a very low temperature. In the case of water ice may thus be made, while with a salt solution temperatures below the freezing-point of water can be obtained and the cold solution circulated for the cooling of refrigerators. To ensure the necessary conditions, evaporation is induced by reduction of pressure which must be maintained by the rapid removal of the vapour. This may be performed by condensation in a connected chamber or by absorption in strong sulphuric acid or by the use of a vacuum pump.

Fig. 47 shows the principle of one apparatus in which a pump is used. Water is sprayed into the chamber *a* which is connected to the first stage of

a vacuum pump by the pipe *b*. In its passage to the bottom of the chamber the air is removed from the water and there is sufficient evaporation to cause considerable cooling. It then passes through the valve *c* into the chamber *d*, which is connected to the second stage of the pump by the pipe *e* and which is under a much smaller pressure. The water is received on the oscillating tray *f*, in which cakes of ice are formed. The valve *c* is only opened intermittently so that the water is always received by *f* when in position for a flow across its whole surface. Another method of producing the necessary high vacuum is shown in fig. 48. A number of steam jets *b* issuing from suitably shaped nozzles withdraw the air from the chamber *a* connected to the space surrounding the jets. A spray of salt solution falling from the top of

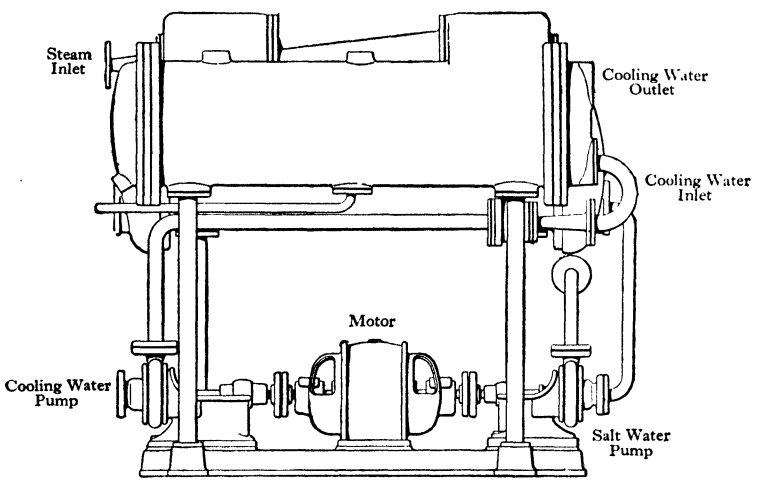


Fig. 49.—Steam Jet Refrigerator

the chamber is cooled by the evaporation of the water from it, and when collected at the bottom may be passed through the pipes of a cooling system. On its return, water must be added to take the place of that evaporated before it is again sprayed through the evaporating chamber. A plant of this type is shown in fig. 49.

(*d*) Heat Insulation for Refrigerators

In the heating of rooms and in the cooling of rooms by warm or cold pipes respectively, a rapid heat transfer is required, but from connecting pipes to the room and from the walls of the rooms it is desired to reduce the heat flow. Walls and pipes should be smooth and there should be no sharp corners in the latter nor sudden changes in the cross section. Moisture should be eliminated as much as possible as latent heat is very large in amount. Valves should not exercise any throttling effect unnecessarily.

If a room is surrounded by insulation, the thickness of the walls and in-

sulation being d_1 and d_2 ft. respectively and the temperatures inside and outside being t_1 and t_2, the heat lost per square foot per hour

$$Q = \frac{t_1 - t_2}{\frac{1}{k} + \frac{1}{k_0} + \frac{d_1}{k_1} + \frac{d_2}{k_2}},$$

where k = heat transfer coefficient for the inner surface;
k_0 = heat transfer coefficient for the outer surface;
k_1 = heat conductivity of wall;
k_2 = heat conductivity of insulation.

For a pipe with inner and outer radii r_1 and r_2 and outer radius of insulation r_3, the quantity of heat flowing through the surface of length l is given by

$$Q = \frac{2\pi l(t_1 - t_2)}{\frac{1}{r_1 k} + \frac{1}{r_3 k_0} + \frac{1}{k_1}\log_e\frac{r_2}{r_1} + \frac{1}{k_2}\log_e\frac{r_3}{r_2}}.$$

The values of k and k_0 depend upon the velocity of the flowing gas or liquid. The values of the coefficients k_1, k_2, . . . for various materials are approximately for

Iron, k_1	= 18	Asbestos, k_2	= 0·04
Brass	= 61	Sawdust	= 0·03
Wood	= 0·1	Felt	= 0·02
Brick	= 0·5	Cork	= 0·02
Zinc	= 74	Cement	= 0·4

CHAPTER VI

Utilization of Heat for Power Purposes

Heat can be utilized for power purposes either by direct combustion of the fuel in the engine or by the employment of an energy container such as air or steam. Engines of either class may be of cylindrical form with reciprocating pistons or may be turbines with continuous rotary motion only. The efficiency of the engine depends upon the temperature gradient of the medium passing through it, i.e. the difference between the highest and the exhaust temperature, and in simple power plants this difference is of primary importance. In large plants, however, the heat remaining in the exhaust can often be utilized for other purposes and it is not so essential that it should be reduced to a minimum.

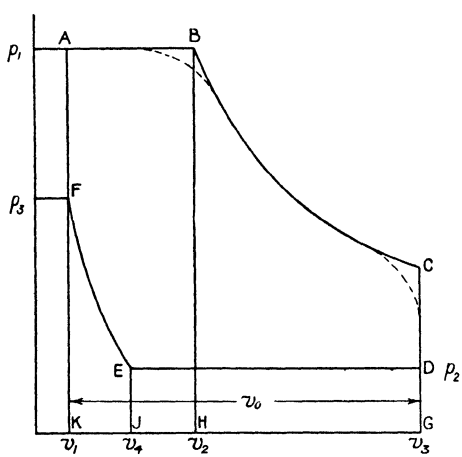


Fig. 1.—Indicator Diagram for Single-Cylinder Engine

1. STEAM RECIPROCATING ENGINE

The indicator diagram for a single-cylinder steam-engine is shown in fig. 1. If p is the mean indicated pressure in pounds per square foot, n the revolutions per minute, and v_0 the stroke volume in cubic feet, then the indicated horsepower is:

$$\text{I.H.P.} = \frac{npv_0}{33{,}000}.$$

For a double-acting single cylinder, if p_1 and A_1 are the mean indicated pressure in front of the piston and the area of the front in square feet, while p_2 and A_2 are corresponding quantities for the back of the piston, then

$$\text{I.H.P.} = \frac{nl}{33{,}000}(p_1A_1 + p_2A_2),$$

where l is the piston stroke. If the indicated pressures are given in atmo-

spheres, the piston areas in square feet, and the mean piston velocity y in feet per second,

$$\text{I.H.P.} = 1{\cdot}925y(p_1A_1 + p_2A_2).$$

To find the indicated horse-power of an actual engine, it is necessary to obtain an indicator diagram experimentally by means of one of the instruments described in a later chapter. An approximate idea of the horse-power, however, may be obtained if the initial and final pressures are known, together with the valve timing and the engine dimensions. The comparative effect of altering any factor can be obtained although the absolute values of the horse-power may be erroneous. In fig. 1:

Let v_1 be the clearance volume;
v_2 the volume at closing of inlet valve;
$v_3 = v_1 + v_0$ the volume at end of stroke;
v_4 the volume at closing of exhaust valve;
p_1 the initial steam pressure (absolute);
p_2 the exhaust pressure (absolute);
r the ratio of cut off $= v_0/(v_2 - v_1)$;
s the expansion ratio $= v_3/v_2 = r$ very nearly; and
$l = v_4/v_1$.

The expansion curve BC and the compression curve EF follow some law $pv^n =$ constant, where n does not differ greatly from unity.

$$\begin{aligned}\text{Work done per stroke} &= \text{ABCDEF}\\ &= \text{ABHK} + \text{BCGH} - \text{EDGJ} - \text{FEJK}\\ &= p_1(v_2 - v_1) + \int_{v_2}^{v_3} pdv - p_2(v_3 - v_4) - \int_{v_1}^{v_4} pdv.\end{aligned}$$

$$\text{Now} \qquad \int_{v_2}^{v_3} pdv = p_1v_2^n(v_2^{1-n} - v_3^{1-n})/(n-1),$$

$$\int_{v_1}^{v_4} pdv = p_2v_4^n(v_1^{1-n} - v_4^{1-n})/(n-1).$$

In the special case where $n = 1$, these expressions take the form

$$\int_{v_2}^{v_3} pdv = p_1v_2 \log_e \frac{v_3}{v_2},$$

$$\int_{v_1}^{v_4} pdv = p_2v_4 \log_e \frac{v_4}{v_1}.$$

The area of the cycle can thus be represented in all cases by the expression

$$W = Ap_1 - Bp_2,$$

where A and B are constants determined only by the dimensions and timing of the engine and the value of n. The mean indicated pressure p can be represented by similar constants thus:

$$p = \frac{W}{v_3 - v_1} = \frac{W}{v_0} = ap_1 - bp_2.$$

Owing to the rounding of the corners of the diagram due to wire-drawing, it is necessary to deduct a small correction c varying between 2 per cent and 5 per cent of p:

$$p = ap_1 - bp_2 - c.$$

As the clearance volume of steam-engines is small the values of a, which depend upon v_3/v_2 or s, will not be greatly affected if it is neglected. The values of b depending on v_4/v_1 are affected directly by the value of the clearance volume, and it is necessary to make full allowance for this in its calculation. The following tables give the values of a and b for various cases.

VALUES OF a

n =	1.	1·135.	1·20.	1·30.
s or r = 1·5	0·937	0·930	0·927	0·922
2	0·847	0·831	0·825	0·813
3	0·700	0·674	0·663	0·645
4	0·596	0·567	0·554	0·533
5	0·522	0·489	0·475	0·455
6	0·465	0·432	0·417	0·397
7	0·421	0·387	0·373	0·357
8	0·385	0·352	0·337	0·318
10	0·330	0·298	0·285	0·266
15	0·247	0·218	0·206	0·190
20	0·200	0·173	0·163	0·149
25	0·168	0·144	0·135	0·122
30	0·147	0·124	0·116	0·104

VALUES OF b FOR AN AVERAGE n = 1·1

	$\frac{v_4}{v_3}$ =	0·2.	0·3.	0·4.	0·5.	0·85.
Clearance $\frac{v_1}{v_0}$ =	0·025	1·376	1·696	2·067	2·476	4·143
	0·05	1·252	1·478	1·748	2·056	3·328
	0·10	1·155	1·310	1·499	1·716	2·649

The values of b may also be determined from the exhaust pressure p_2 and the pressure at the end of compression p_3. For example, with a clearance of 5 per cent, the values of b for ratios of p_3/p_2 as given in the following table are:

p_3 =	5.	6.	7.	8.	9.	10.	11.	12.
p_2 = 1·1	1·130	1·189	1·259	1·327	1·396	1·470	1·555	1·636
1·5	1·069	1·153	1·145	1·188	1·234	1·282	1·333	1·386
2·0	1·033	1·055	1·077	1·105	1·135	1·164	1·195	1·233
3·0	1·009	1·017	1·025	1·041	1·058	1·073	1·089	1·106

In the engines with double or triple expansion the exhaust steam from the first cylinder is introduced into and works the second cylinder and the exhaust from the second works the third cylinder. The volumes of the cylinders must increase successively according to the initial pressures for the cylinders. The volume ratios of the cylinders in practice vary according to the following figures.

	Initial Pressure, lb. per sq. inch.	Low-pressure Volume / High-pressure Volume
Double expansion ..	90 to 120	2·25 to 2·40
	150 to 180	2·28 to 3·0
	225	4·0
Triple expansion ..	to 180	$V_3 : V_2 : V_1$ = 5 to 7 : 2·8 to 2·25 : 1

The diagrams for the cylinders are superimposed as illustrated in fig. 2. Tangential to the expansion curve of the high-pressure cylinder is shown a broken expansion line following the law pv = constant, which represents the curve which might have been obtained if the whole of the expansion could have been produced in one cylinder. The difference in the area under the broken curve and the sum of the three actual diagrams represents the losses due to the passage of the steam between the different cylinders. The value of the coefficient a applicable to these cases is given by

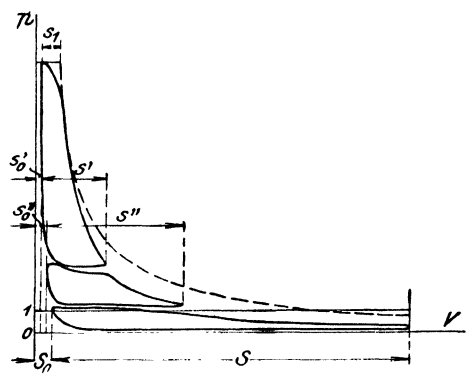


Fig. 2.—Indicator Diagram for Triple Expansion Engine

$$a_1 = \phi a,$$

where ϕ is a fraction. For engines with clearances from 5 to 8 per cent, ϕ has the following average values.

Slow-speed engines	ϕ = 0·75
Medium speed, wet steam	ϕ = 0·65 to 0·7
Medium superheat	ϕ = 0·57 to 0·67
High superheat	ϕ = 0·55 to 0·60

A rough rule which represents the performance of average engines is given by Grassman as follows:

Single-cylinder condensing engines	$p = 18 + 0{\cdot}20\,p_1$
Single-cylinder non-condensing engines	$p = 18 + 0{\cdot}25\,p_1$
Double compound condensing engines	$p = 18 + 0{\cdot}09\,p_1$
Triple compound engines	$p = 18 + 0{\cdot}05\,p_1$
Triple compound engines (ship)	$p = 22 + 0{\cdot}07\,p_1$

If the closing of the exhaust valve and the subsequent cushioning of the exhaust steam is such that the maximum compression pressure is equal to the initial pressure, i.e. A ≡ F and $p_3 = p_1$, fresh steam to fill the stroke to the point of cut off, $v_2 - v_1$, only is required. In the ideal case with no clearance the diagram reduces to fig. 3, and the maximum amount of work would be obtained. The performance of the actual engine is hence compared to this simplified case with the same initial pressure, exhaust pressure, and cut off, and with adiabatic expansion. The weight of steam required per cycle, $v_2 - v_1$ at pressure p_1, and stated temperature, can be found from steam tables as exemplified in Chapter I and compared with the actual steam used per cycle. The ratio between the two is known as the efficiency ratio.

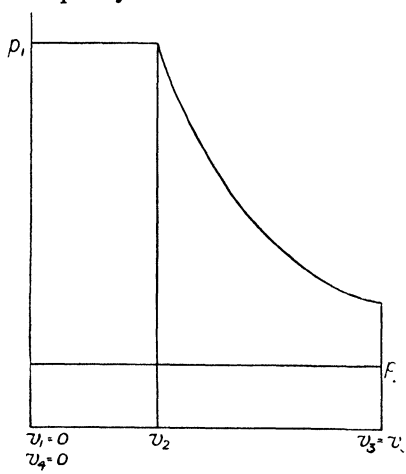


Fig. 3.—Indicator Diagram showing no Clearance

The excess of steam required is due to heat losses to the walls of the engine and the passages from the boiler, which result in a certain amount of condensation, and to leakage past the piston. This excess of steam, usually denoted the missing steam, increases with an increase of expansion ratio and diminishes with increase of speed, with superheat, and with the use of steam jacketing. The missing steam in a compound engine is less than in a single-cylinder engine of the same expansion ratio, in spite of the heat losses which occur in the passages between the cylinders of the compound engine. This result is due to the reduction in the temperature gradient between the steam and the walls in each cylinder, and to the reduction in the pressure gradient between the two sides of the piston. The leakage due to the latter cause is only a loss in the low-pressure cylinder, as the leaks in the high- and intermediate-pressure cylinders pass to the intermediate- and low-pressure cylinders respectively. The following table gives some experimental results. The efficiency ratio is merely a relative efficiency with respect to a certain more perfect cycle. The absolute thermal efficiency, i.e. the ratio of the work obtained to the heat put in, is very much smaller. The former can be obtained from the area of the cycle and the latter from the initial total heat of the steam required per cycle as given by the steam tables.

The heat equivalent of an indicated horse-power hour is 2550 B.Th.U., so that the efficiencies revealed are very low except with the highest pressures and superheat.

The influence of the various factors on the steam consumption is shown in another way in the diagrams of figs. 4 to 7. In the first two diagrams the steam consumption of a high-pressure cylinder of a compound engine is

Character of Engine and Steam.			Initial Pressure, p_1. Pounds per Square Inch.	Consumption.		Thermal Efficiency.	Efficiency Ratio.
				Pounds per I.H.P. Hour.	B.Th.U. per I.H.P. Hour.		
Single cylinder.	Non-condensing.	Saturated.	150 to 180	22 to 19	27,000 to 22,500	0·095 to 0·110	0·645 to 0·716
		Superheated to 550°–620° F.		16 to 13	21,000 to 18,000	0·119 to 0·140	0·768 to 0·810
	Condensing.	Saturated.	120 to 150	16·5 to 15·5	20,000 to 18,500	0·127 to 0·135	0·520 to 0·534
		Superheated to 550°–620° F.	150 to 180	11·5 to 10	15,000 to 13,500	0·166 to 0·186	0·636 to 0·674
Double expansion.	Condensing.	Saturated.	120 to 150	16·5 to 12	20,000 to 15,000	0·127 to 0·172	0·520 to 0·665
		Superheated to 500° F.		13 to 10.5	17,000 to 13,500	0·147 to 0·184	0·591 to 0·695
		Superheated to 550°–620° F.		11 to 9·5	14,500 to 12,500	0·173 to 0·199	0·682 to 0·722
Triple expansion.	Condensing.	Saturated.	150 to 225	13 to 11	16,000 to 13,500	0·158 to 0·185	0·606 to 0·680
		Superheated to 500° F.		11 to 10	14,500 to 12,500	0·177 to 0·197	0·667 to 0·717
		Superheated to 550°–620° F.		10 to 9	13,000 to 12,000	0·192 to 0·209	0·714 to 0·735

shown using saturated and superheated steam respectively; in fig. 7 the consumption for the intermediate-pressure cylinder; and in fig. 6 that for the low-pressure cylinder or for a single-cylinder engine exhausting either to atmosphere or to a condenser.

A simple experimental law connecting the steam consumption of an

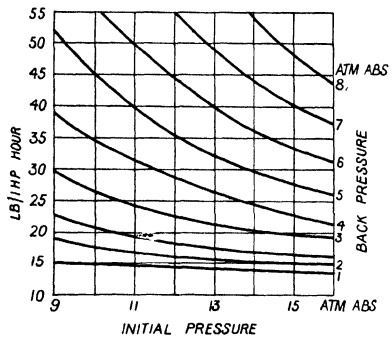


Fig. 4.—Steam Consumption (no losses) with Saturated Steam

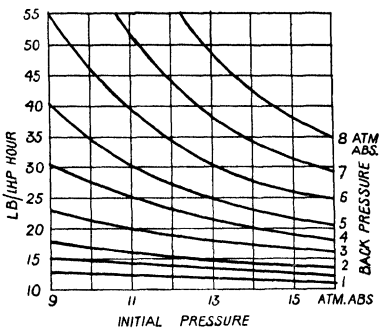


Fig. 5.—Steam Consumption (no losses) with Superheated Steam

engine when throttled (the character of the steam remaining constant), or the heat consumption of the engine if the steam undergoes slight alteration with the indicated horse-power, has evolved from trials on many engines. If w is the steam consumption in pounds and H the heat consumption in B.Th.U. per hour,

$$\text{I.H.P.} = aw - b,$$
$$\text{or I.H.P.} = a_1 H - b_1.$$

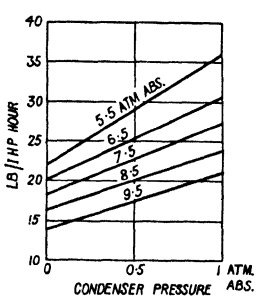


Fig. 6.—Steam Consumption Condensing Engine

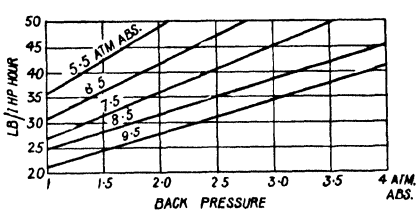


Fig. 7.—Steam Consumption Intermediate Pressure Cylinder

The constants a, b, a_1, and b_1 depend upon the type and design of the engine. The constant a also depends on the character of the steam, but for modern engines of the same type the value of a_1 appears to be almost independent of the pressure and temperature of the steam within the usual workable limits of the engine. For example, tests on a number of locomotive engines using saturated steam but at various initial pressures, yielded results which were all very near to the following average curves.

$$\text{Compound engines, I.H.P.} = \frac{H}{23{,}000} - 60.$$

$$\text{Simple engines, I.H.P.} = \frac{H}{27{,}000} - 60.$$

When using saturated steam at (say) 200 lb. per square inch and total heat 1205 B.Th.U. per pound:

$$\text{Compound engines, I.H.P.} = \frac{w}{19} - 60.$$

$$\text{Simple engines I.H.P.,} = \frac{w}{22{\cdot}5} - 60.$$

The thermal efficiency represented by these equations is low. Thus with a compound engine for 1000 h.p. the value of H is $24{\cdot}4 \times 10^6$ B.Th.U. per hour, while the thermal equivalent of the 1000 h.p.-hr. is $2{\cdot}55 \times 10^6$ B.Th.U. The efficiency is thus only 10·5 per cent, a value to be expected from non-condensing engines using saturated steam.

In order to maintain a required vacuum, heat must be abstracted in the condenser in amounts which form a very large proportion of the heat of formation of the steam. With the cycle of fig. 3 the percentage of exhaust heat to be removed would diminish with reduction of the vacuum, but owing to the rounding of the diagram the percentage shows an increase at the lowest condenser pressures. The percentage has a minimum value for an initial steam pressure of 40 atmospheres and is greater for lower or higher initial steam pressures.

Initial Pressure, Atmospheres	Condenser								
	Pres., Atmosphere.	Temp., °F.	Percentage of Exhaust Heat.	Pres., Atmosphere.	Temp., °F.	Percentage of Exhaust Heat.	Pres., Atmosphere.	Temp., °F.	Percentage of Exhaust Heat.
10	0·1	114	92·8	0·2	140	94·3	0·3	156	94·8
20			91·8			92·6			93·1
40			90·8			91·7			92·0
100			94·6			95·4			95·7

The most favourable vacuum appears to be about 85 or 86 per cent, although steam-turbines can utilize a vacuum up to 96 per cent with advantage.

The heat abstracted from the exhaust steam is usually a dead loss, as owing to its low temperature it is difficult to utilize. The exhaust heat of the waste gases of the furnace is used to preheat the feed water. Regenerators used for this purpose are termed economizers. Recently a method of heating the feed water has been introduced, in which a portion of the working steam is withdrawn from the engine cylinder after a partial expansion and while still at considerable pressure and temperature, and is used for the purpose. The whole of the heat from this portion of the steam is thus recovered. With a compound engine, if steam is thus withdrawn from the high-pressure cylinder, the volume of the low-pressure cylinder must necessarily be less than in the

normal method of working. Under favourable circumstances one-half of the steam may be withdrawn. The character of the indicator diagrams with an early and a late release of the portion of steam is shown in figs. 8 and 9.

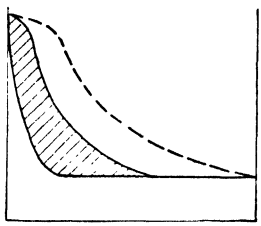


Fig. 8.—Partial Steam Release (early)

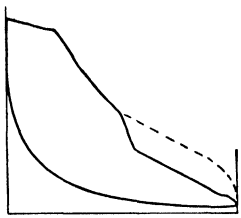


Fig. 9.—Partial Steam Release (late)

The general arrangement of a small single-cylinder engine is shown in plan and elevation in fig. 10. As examples of the performance of this class of engine, the following figures are quoted for the output with saturated steam:

	Bore.	Stroke.	R.P.M.	Steam Pressure.	Maximum H.P.
	Inches.	Inches.		Lb. Sq. In.	
Non-condensing	8	20	135	80	28
	8	20	135	100	35
	20	48	75	80	232
	20	48	75	100	293
Condensing ..	8	20	135	80	35
	8	20	135	100	41
	20	48	75	80	276
	20	48	75	100	340

The steam is supplied to the engine by means of the steam pipe A, fig. 10, and passes through the cylinder jacket before entering the cylinder. The two inlet valves for the opposite sides of the piston are shown at BB, while the two exhaust valves are underneath at CC. The exhaust steam passes out by the exhaust pipe D to the jet condenser E by which it is condensed and the vacuum maintained. The condenser pump is operated by a shaft attached to the piston and passing through the cylinder end. Instead of the older type of slide valve, drop valves similar in action to the valves of internal-combustion engines are used. The opening and closing of the valves can be performed with greater rapidity and " wire-drawing " reduced. The action of the valves is maintained by the rotating shaft M driven from the crank shaft. Details of the non-condensing engine cylinder and of the valve gear are shown in fig. 10(*a*), the piston being at the end of its stroke. The inlet valves A and the exhaust valves N are normally kept on their seatings by the springs at L and S respectively. The inlet valves are lifted alternately by levers B which are depressed at their outer ends by the bell-crank levers C.

Motion is given to the levers by the eccentrics E mounted on the shaft M. The period of opening is determined by the arm G carried by eccentrics H

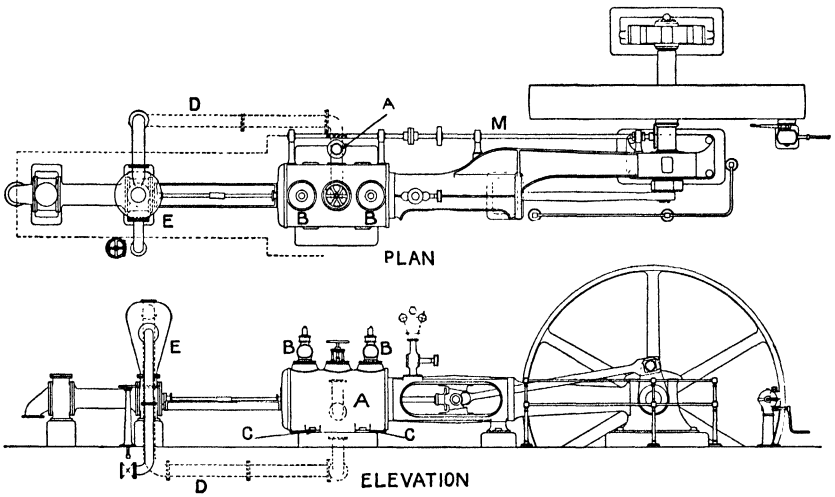


Fig. 10.—General Arrangement of Condensing Single-Cylinder Steam-Engine (Marshall & Co., Gainsborough)

on the shaft J which is connected to the governor. The springs L are enclosed in air boxes for damping which can be regulated by the valve F. The exhaust

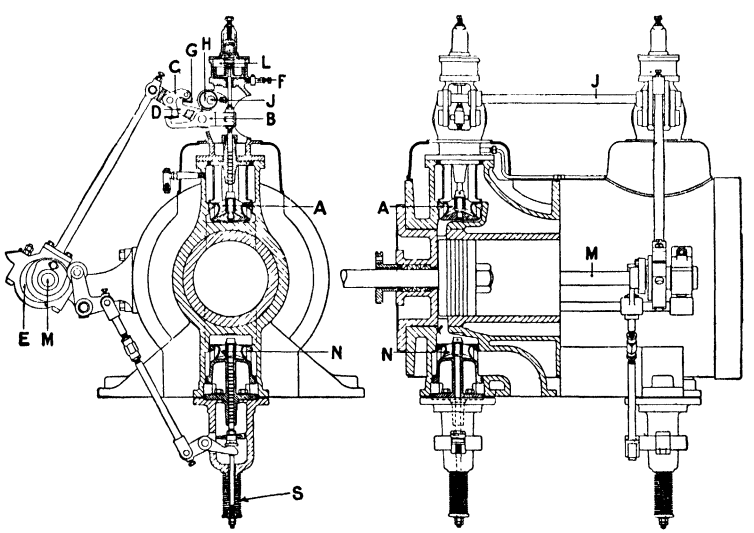
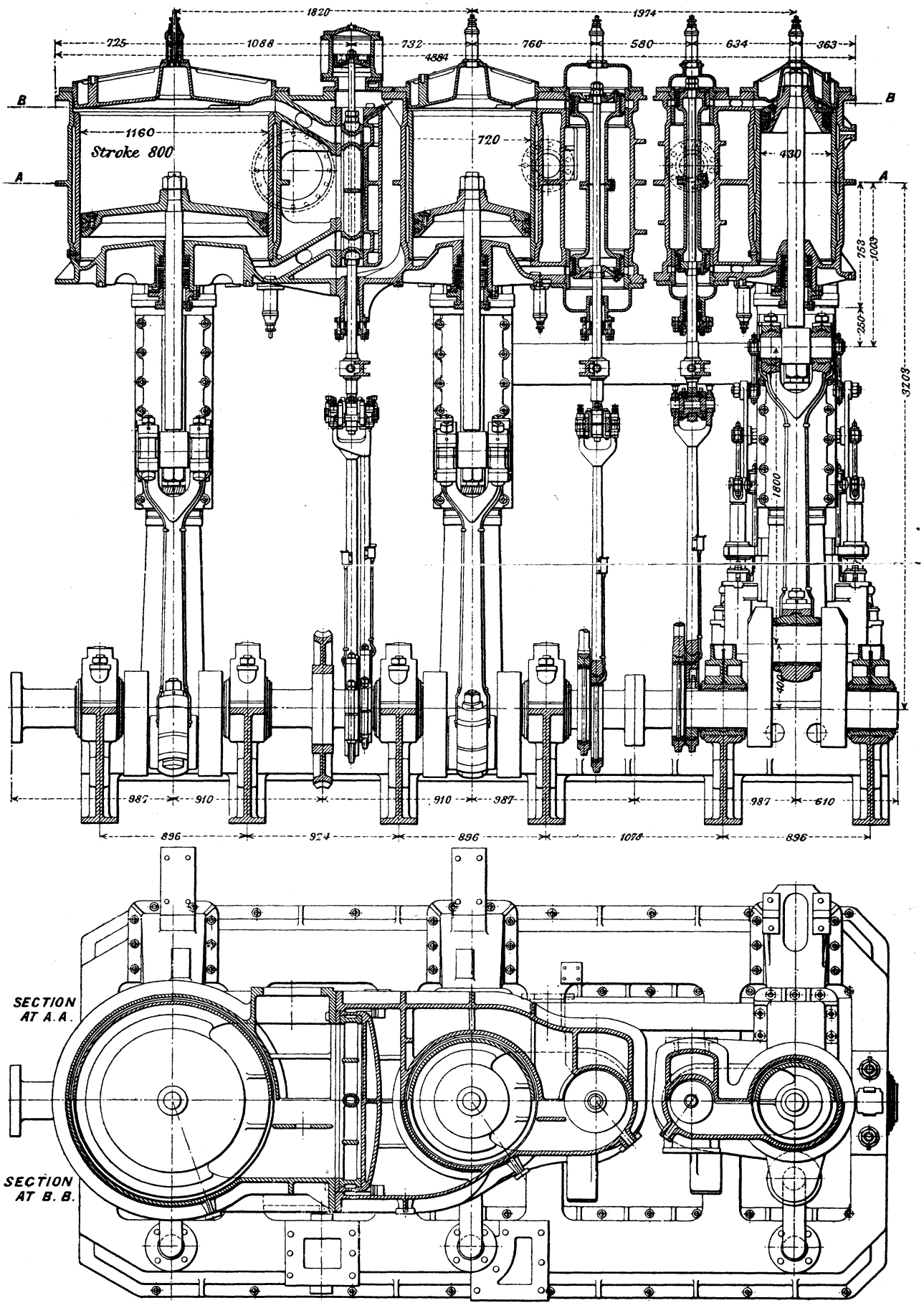


Fig. 10(*a*).—Steam-Engine Cylinder with Drop Valves

valves open by the removal of the spring pressure by similar levers operated by an eccentric on the shaft M.



From *Engineering*

Fig. 11.—Triple Expansion Marine Engine

[*Facing p. 246*

by expansion through a nozzle, and the jet impinges upon the vanes of the rotating wheel to which it imparts its energy. There is no change of pressure in the passage through the turbine. In the reaction turbine the expansion and loss of pressure take place within the turbine itself.

The two types of turbines may be represented diagrammatically by figs. 12 and 13. In fig. 12 steam expands through the nozzle, emerging with a velocity v and impinging on the blade which has a velocity u. The velocity of the steam relative to the vane is thus given by X and the direction of the blade entrance must be parallel to X in order to avoid loss of energy by impact. Neglecting friction, the velocity of the steam is constant as there is no drop in pressure, and changes only in direction. The inclination of exit is the same as at entrance and the absolute velocity of the issuing steam is given by Y. In the reaction turbine shown in fig. 13, the steam issuing from the guides with velocity v will, as before, have a velocity relative to the blades at entrance given by X. Owing to the drop of pressure during passage through the blades, the velocity of emergence relative to the blades is increased to X′ and the absolute velocity is Y′. The drop in pressure and the mechanical work obtained by action through a single set of blades cannot be very great and hence a series of alternate guides and blades are used. The angle of entrance into the next guide must be parallel to Y′. Turbines are also constructed which work partly on the impulse principle and partly by reaction.

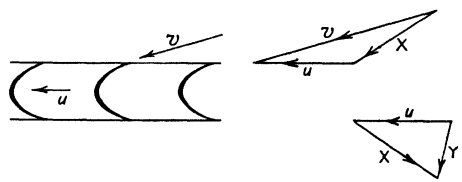


Fig. 12.—Velocity Diagram for Simple Impulse Turbine

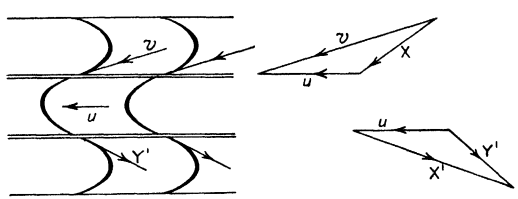


Fig. 13.—Velocity Diagram for Simple Reaction Turbine

It has already been shown in Chapter I that when a gas at rest expands adiabatically through an orifice from an original pressure p_0 to a final pressure p_1, the maximum weight flowing per second is attained where

$$\frac{p_1}{p_0} = \left(\frac{2}{\gamma + 1}\right)^{\frac{\gamma}{\gamma-1}}.$$

For superheated steam the ratio of the specific heats, γ, is 1·300 while for wet steam $\gamma = 1{\cdot}035 + 0{\cdot}1x$, where x is the dryness fraction. Thus for dry saturated steam where $x = 1$, $\gamma = 1{\cdot}135$. The types of nozzles used are represented by figs. 14 and 15, the former showing one with divergent exit and the latter with non-divergent exit. In the first case p_0, the minimum pressure, is at the throat while in the latter throat and exit coincide. For dry saturated steam the maximum flow is given when $p_1/p_0 = 0{\cdot}58$. If the

flow for any nozzle is then w_m lb. per hour, for other ratios the flow is kw_m where k is given by the following table:

$\frac{p_1}{p_0}$ =	1·00	0·98	0·96	0·94	0·92	0·90	0·85	0·80	0·70	0·58
k =	0·00	0·31	0·43	0·52	0·60	0·66	0·77	0·86	0·96	1·00

If p_1/p_0 is less than 0·58, the pressure of the steam at the minimum cross section is nevertheless 0·58 p_1, and additional expansion takes place in the divergent portion of divergent nozzles or in the space outside with non-divergent nozzles. In the latter case the resultant turbulence causes a re-creation of heat and loss of mechanical energy.

If the area of the throat is A sq. in, and V_0 the volume of 1 lb. of steam

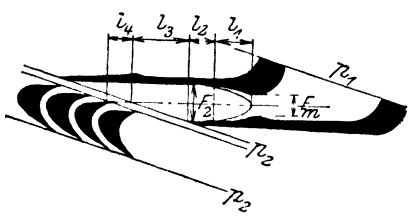


Fig. 14.—Divergent Nozzle

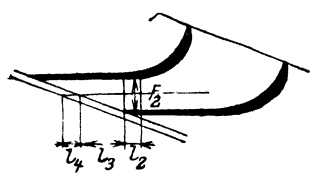


Fig. 15.—Non-divergent Nozzle

at the initial pressure p_0, the expansion still being assumed adiabatic, the maximum flow w in pounds per second for dry saturated steam is given by

$$w = 0{\cdot}304A\sqrt{\frac{p_0}{V_0}}.$$

If tables of the value of V_0 are not available an approximate value is given by the equation

$$w = 0{\cdot}2P_0 + 0{\cdot}07,$$

where P_0 is the initial pressure in atmospheres. This simple formula differs from the previous formula by less than 1 per cent for pressures between 3 and 20 atmospheres. The accurate values for this range of pressures are as follows:

P_0 =	1	5	10	15	20
w =	0·235	1·07	2·09	3·11	4·12

For superheated steam the maximum discharge is given by the similar formula:

$$w = 0{\cdot}3155A\sqrt{\frac{p_0}{V_0}}.$$

To obtain these maximum flows the pressure gradient p_1/p_0 must be equal to or greater than the critical value and the flow must be adiabatic. In actual nozzles there is friction and recreation of heat; and the kinetic energy should be somewhat less than that given by above values of

the delivery and the corresponding velocities. In practice, however, the deliveries are found to be a little greater than those calculated, owing to a delay in the precipitation of the water, a state of supersaturation occurring.

The issuing velocity could also be obtained in terms of the drop of total heat by equating the increase of kinetic energy to the loss of heat energy. Thus if H_0 is the total heat per pound of steam in B.Th.U. under the initial conditions and H_1 under the final conditions, while U is the issuing velocity in feet per second, then neglecting conduction losses

$$U^2 = 50{,}000\ (H_0 - H_1).$$

If the steam has velocity U_0 under the initial conditions, then the velocity U_1 attained is given by

$$U_1^2 - U_0^2 = 50{,}000\ (H_0 - H_1).$$

In this expression H_1 is the actual total heat of the steam at the exit, which will be greater than that expected from adiabatic expansion owing to the friction. An approximate idea of the effect of the friction can be obtained in the case of carefully made nozzles, as shown in figs. 14 and 15, from the following formulæ:

A is the area of throat;
A_1 the exit area of divergent nozzle;
l_1 the length of expanding portion;
l_2 the length of parallel portion; and
l_3 and l_4 the length of truncated end.

(1) Loss at entrance h_0 should be small in carefully curved nozzles.

(2) Loss in l_1:

$$h_1 = \frac{a(b-1)}{bc}(H_0 - H_1),$$

where $a = 0{\cdot}01$ to $0{\cdot}02$,

$$b = A_1/A,$$

$$c = (\sqrt{A_1} - \sqrt{A})/l_1.$$

(3) Loss in l_2:

$$h_2 = \frac{2al_2}{\sqrt{A_1}}(H_0 - H_1).$$

(4) Loss in l_3:

$$h_3 = \frac{a}{\tan\alpha}(H_0 - H_1),$$

whese α is the inclination of the end of the nozzle to its axis.

(5) Loss in l_4:

$$h_4 = \frac{l_4}{l_3}h_3.$$

If then H_1' is the total heat calculated according to adiabatic flow,

$$H_1 = H_1' + (h_0 + h_1 + h_2 + h_3 + h_4).$$

In addition to the allowances for friction there are, as in the reciprocating engine, heat losses from radiation and conduction and also leakage losses.

Turbines are classified according to the character of the exhaust steam. When the exhaust can be utilized for heating or other purposes, the initial pressure of the steam supply to the turbine being high while the final pressure is atmospheric or above, the turbines are known as back-pressure turbines. Where low-pressure steam cannot be thus utilized, the maximum mechanical power is obtained from it in the turbine by using a condenser and expanding the steam to a vacuum which may be as low as half an inch of mercury absolute. These turbines are termed condenser turbines. The great advantage of the turbine is that by the use of a condenser considerable work can be obtained from steam initially only at a low pressure. The exhaust from a back-pressure turbine, a reciprocating steam-engine, or other source may be utilized in turbines denoted exhaust turbines. Turbines in which provision is made for withdrawing part of the steam at some intermediate stage, if some is required for heating or other purposes, are known as reducing, intermediate, or supply turbines. Where exhaust low-pressure steam is not used to drive a separate exhaust turbine but is added at some stage to a turbine which also uses high-pressure steam, the term mixed-pressure turbine is given.

As may be expected from the variation in the steam conditions, the consumption per brake-horse-power hour also varies greatly. The following gives some idea of the values obtained.

Back-pressure Turbine:

200 lb. per square inch; 200° F. superheat—20 lb. per b.h.p. hour with atmospheric back pressure to 50 lb. per b.h.p. hour with three atmospheres back pressure.

Condensing Turbine:

200 lb. per square inch; 200° F. superheat—8·5 to 10 lb. per b.h.p. hour.
200 lb. per square inch saturated—12 to 15 lb. per b.h.p. hour.

Exhaust Turbine:

Atmospheric pressure, saturated—20 to 25 lb. per b.h.p. hour.

As will be seen from the diagram of the Parsons steam-turbine, reaction turbines are usually divided into three sections denoted the high-pressure, intermediate-pressure, and low-pressure sections respectively. The distribution of the power in the three sections is illustrated by the following results obtained by Gensecke for the heat drop in the sections of a 400-h.p. Parsons turbine.

HEAT DROP IN B.TH.U. PER POUND

H.P. =	450.	310.	170.	Running Light.
High-pressure section	88	90	95	77
Intermediate-pressure section	98	99	100	76
Low-pressure section	162	144	112	36
Total ..	348	333	307	189

With the normal load, the power derived from the three sections is thus approximately $\frac{1}{4}$, $\frac{1}{4}$, and $\frac{1}{2}$. The low-pressure section is less important at low loads, and when running light the ratio of the powers from the three sections is $\frac{5}{16}$, $\frac{5}{16}$, and $\frac{1}{6}$.

The pure impulse turbine due to de Laval is manufactured by Messrs. Greenwood & Batley, of Leeds, in standard sizes from 5 to 500 h.p. As the whole expansion from admission to exhaust pressure is carried out in a single stage the steam speed is exceedingly high, and its kinetic energy must be taken from it by the use of a very great speed of running for the turbine. Actually the small power sets run at 30,000 r.p.m., while the larger ones have a speed of 10,000 r.p.m. The speed of running of the small de Laval turbine is believed to be the highest ever attained on any practical and useful mechanical

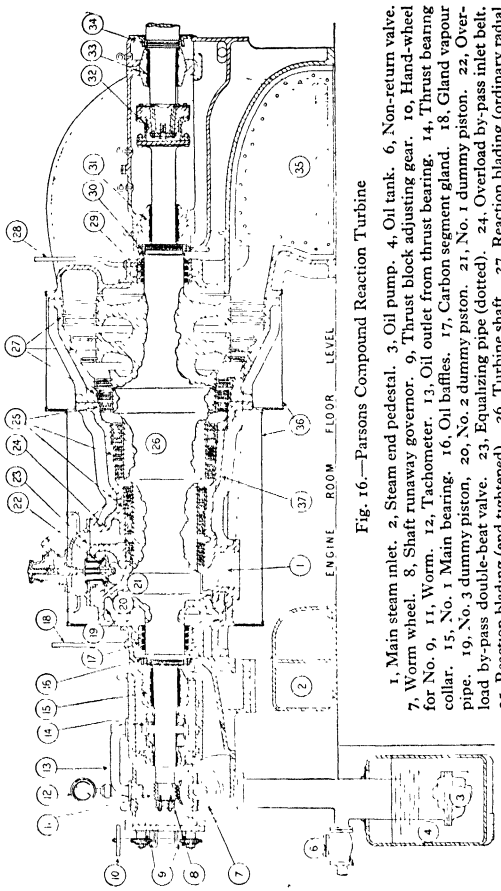


Fig. 16.—Parsons Compound Reaction Turbine

1, Main steam inlet. 2, Steam end pedestal. 3, Oil pump. 4, Oil tank. 6, Non-return valve. 7, Worm wheel. 8, Shaft runaway governor. 9, Thrust block adjusting gear. 10, Hand-wheel for No. 9. 11, Worm. 12, Tachometer. 13, Oil outlet from thrust bearing. 14, Thrust bearing collar. 15, No. 1 Main bearing. 16, Oil baffles. 17, Carbon segment gland. 18, Gland vapour pipe. 19, No. 3 dummy piston. 20, No. 2 dummy piston. 21, No. 1 dummy piston. 22, Overload by-pass double-beat valve. 23, Equalizing pipe (dotted). 24, Overload by-pass inlet belt. 25, Reaction blading (end tightened). 26, Turbine shaft. 27, Reaction blading (ordinary radial clearance). 28, Gland vapour pipe. 29, Carbon segment gland. 30, Oil baffles. 31, No. 2 main bearing. 32, Flexible coupling. 33, No. 3 main bearing. 34, Oil baffles. 35, Turbine exhaust. 36, Cylinder lagging. 37, Equalizing pipe (dotted).

device. To obtain a suitable speed for an electric generator driven by a de Laval turbine, gearing must be introduced and the double-reduction type is favoured. The driving of a rigidly held shaft at such high speeds would cause endless breakdowns and trouble, and in consequence floating bearings are used which can accommodate themselves to any condition of running. The de Laval turbine may be described as a single-pressure stage turbine with single-velocity stage. The single-pressure stage turbine is one on which all the pressure drop occurs on one set of nozzles, and the statement that it has a single-velocity stage means that the velocity is taken out of the steam on a single row of blading.

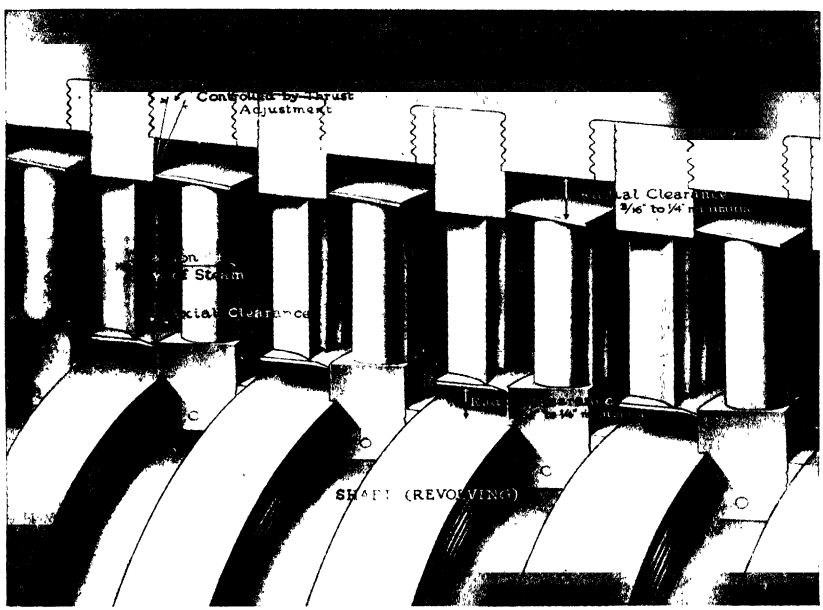


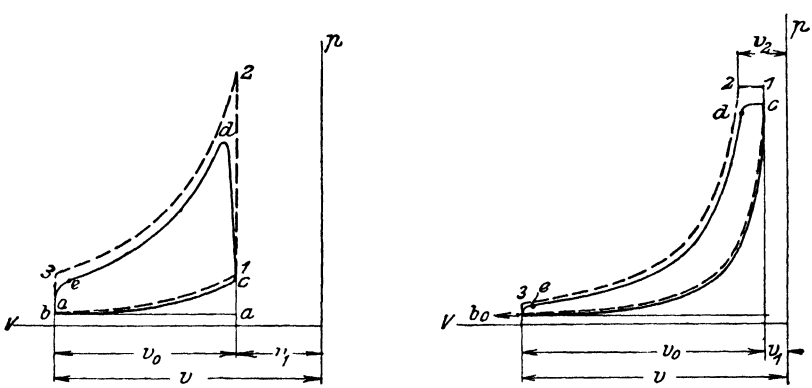
Fig. 17.—Parsons " End-tightened " Blading in Cylinder and Shaft, showing large Radial and Axial Clearance

The first practical steam-turbine was built by Sir C. A. Parsons in 1884. Fig. 16 shows a section through a Parsons-type compound reaction turbine, and shows features of the most up-to-date practice. Steam enters at port 1 and flows through the turbine to the exhaust end 35. Immediately after the first expansion is the overload by-pass inlet belt 24. The dummy pistons for balancing the motor are shown at 19, 20, and 21, and the equalizing pipes for equalizing the dummy pressure and that on the corresponding portion of the rotor are shown dotted, 23 and 37. The rotor body is forged in one piece. The top half casing fits over long upright standards in the lower half, so that it can readily and easily be removed and replaced. All high-pressure machines are fitted with an automatic atmospheric relief valve, so that should the vacuum fail at any time the turbine will exhaust to atmosphere.

The oil tank will be seen together with the oil pump, which supplies oil under pressure to all bearings. Between the steam space and the bearings are placed the oil baffles and then the carbon segment glands—these are shown at 16, 30 and 17, 29. The glands are steam packed, and 18, 28 show the vapour pipes from these glands. The steam is admitted to the turbine through a governor valve. Both this valve and the overload by-pass valve are automatic, and are controlled from the governor through steam relays. One of the most important improvements made in the detail of the Parsons machine is what is known as the end-tightened blading. This system of blading entirely eliminates fine radial clearances, appears to suffer very little deterioration with service, and gives mechanical reliability. In fig. 17 a perspective view of end-tightened reaction pairs of rows (one fixed and one moving) forming a part of the blading of a Parsons machine is shown.

3. INTERNAL-COMBUSTION RECIPROCATING ENGINES

Reciprocating internal-combustion engines can be divided into those with combustion at constant volume and those with combustion at constant pressure. The cycles for the two types are represented in fig. 18, the broken



(*a*) Constant volume combustion (*b*) Constant pressure combustion

Fig. 18.—Internal Combustion Cycle

line being the ideal working cycle and the continuous line the cycle actually realized. The engines can have either a four-stroke or a two-stroke cycle. In the former the four processes are:

1. Suction.
2. Compression.
3. Combustion and expansion (working stroke).
4. Exhaust.

In the two-stroke engine there are no separate exhaust and suction strokes,

and these processes take place during the last portion of the expansion stroke and the first portion of the compression stroke. Engines may be single or double acting, and there may be one or many cylinders acting on the same crank shaft.

The ideal diagram (fig. 18*a*) for the internal-combustion engine with combustion at constant volume consists of two adiabatic curves and two lines of constant volume. The ideal fluid would have a specific heat which is constant at all temperatures and the same before and after combustion. If w is the weight of fluid concerned in each cycle, including fresh mixture and the retained residue in the clearance volume, while C_v is the specific heat at constant volume,

$$\begin{aligned}
\text{Heat supplied, } H_1 &= wC_v(T_2 - T_1);\\
\text{Heat given off, } H_2 &= wC_v(T_3 - T_0);\\
\text{Heat transformed into work, } H &= H_1 - H_2;\\
\text{Theoretical thermal efficiency, } \eta_0 &= \frac{H_1 - H_2}{H_1}\\
&= 1 - \frac{T_0}{T_1}.
\end{aligned}$$

If the compression ratio $(v_0 + v_1)/v_1 = r$ and the ratio of the specific heats $C_p/C_v = \gamma$, then in terms of the pressures and volumes

$$\begin{aligned}
H &= \frac{p_2v_1}{\gamma - 1}\left(1 - \frac{1}{r^{\gamma-1}}\right) - \frac{p_1v_1}{\gamma - 1}\left(1 - \frac{1}{r^{\gamma-1}}\right)\\
&= \frac{v_1(p_2 - p_1)}{\gamma - 1}\left(1 - \frac{1}{r^{\gamma-1}}\right);\\
\eta_0 &= 1 - \frac{1}{r^{\gamma-1}}\\
&= 1 - \left(\frac{p_0}{p_1}\right)^{\frac{\gamma-1}{\gamma}}.
\end{aligned}$$

With the ideal cycle using combustion at constant pressure as shown in fig. 18*b*, if $(v_0 + v_1)/v_2 = r_1$ and $v_2/v_1 = r_2$,

$$\begin{aligned}
H &= p_1(v_2 - v_1) + \frac{p_1v_2}{\gamma - 1}\left(1 - \frac{1}{r_1^{\gamma-1}}\right) - \frac{p_1v_1}{\gamma - 1}\left(1 - \frac{1}{r^{\gamma-1}}\right),\\
\eta_0 &= 1 - \frac{1}{\gamma}\cdot\frac{r_2^{\gamma} - 1}{r^{\gamma-1}(r_2 - 1)}.
\end{aligned}$$

The actual cycles differ from the ideal cycles owing to (1): the specific heat during the combustion process from temperature T_1 to T_2, beginning with unburnt mixture and ending with combustion products, exhibits a large increase of specific heat as the temperature increases; (2) the specific heat of the burnt gases is different from that of the unburnt mixture; (3) heat is lost to the walls so that

the compression and expansion curves are not adiabatic; (4) time is not available to complete the combustion and exhaust processes at constant volume.

If the net mean pressure realized is p lb. per square foot, the stroke volume being v_0 c. ft., the work realized per cycle

$$H = pv_0 \text{ ft.-lb.}$$

With a four-stroke engine at n r.p.m., the indicated horse-power,

$$\text{I.H.P.} = \frac{npv_0}{2 \times 33{,}000}.$$

With a two-stroke engine

$$\text{I.H.P.} = \frac{npv_0}{33{,}000}.$$

The value of p depends on the calorific value of the working fluid. If the fuel is a gas or gasified liquid whose lower calorific value is $\mathbf{H}_1$ B.Th.U. per cubic foot, while the air supplied for combustion of 1 c. ft. of gas is V c. ft., the calorific value of the mixture

$$\mathbf{H}_2 = \mathbf{H}_1/(1 + \mathrm{V}).$$

Owing to friction and heating, the volume drawn in if measured at standard pressure and temperature (these being the conditions under which $\mathbf{H}_1$ is specified) is $\eta_v v_0$, where η_v is a fraction termed the volumetric efficiency. Hence the available heat per cycle is $\eta_v v_0 \mathbf{H}_2$. As the realized work per cycle in B.Th.U. is $pv_0/778$, the indicated thermal efficiency

$$\eta_i = \frac{pv_0}{778\eta_v v_0 \mathbf{H}_2} = \frac{p}{778\eta_v \mathbf{H}_2},$$

or

$$p = 778\eta_i\eta_v\mathbf{H}_2 \text{ lb. per square foot.}$$

If the consumption of the engine is given as W lb. per hour and the lower calorific value as h B.Th.U. per pound, the heat consumption per cycle for four-stroke and two-stroke engines respectively is

$$\frac{\mathrm{W}h}{30n} \text{ and } \frac{\mathrm{W}h}{60n},$$

and

$$\eta_i = \frac{pv_0n}{25{\cdot}9\mathrm{W}h} \text{ or } \frac{pv_0n}{51{\cdot}9\mathrm{W}h}$$

$$= \frac{2550 \text{ I.H.P}}{\mathrm{W}h}.$$

Some of the work thus performed by the working fluid is absorbed in friction of the moving parts of the engine itself and in friction of the gases in their passage to and from the engine. Hence the work

available for external useful purposes, i.e. the brake horse-power, is less than the indicated horse-power, the ratio being the mechanical efficiency η_m.

$$\eta_m = \frac{\text{B.H.P.}}{\text{I.H.P.}}.$$

$$\text{B.H.P.} = \eta_m \eta_i \frac{Wh}{2550}.$$

The overall efficiency

$$\eta = \eta_m \eta_i = \frac{2550 \text{ B.H.P.}}{Wh}.$$

The value of η_m for actual engines lies usually between 0·80 and 0·85 for constant volume combustion engines and between 0·7 and 0·8 for constant pressure combustion engines. The value of the volumetric efficiency η_v may be 0·9 for a large well-designed engine running at a medium speed, but is very much lower for small engines running at very high speeds. Under running conditions the effect of a change in atmospheric temperature is small, but the volumetric efficiency is practically proportional to atmospheric pressure. As p and I.H.P. are proportional to η_v, the power of an aeroplane engine thus falls off at an altitude. This is illustrated by the following table, which gives the barometric height at various altitudes under normal temperature conditions.

Height in feet ..	0	1000	2000	5000	10,000	20,000
Barometer, inches of mercury	30·0	28·8	27·7	24·6	20·0	17·2
Relative pressure ..	1·00	0·96	0·92	0·82	0·67	0·57

Average experimental data for the output and efficiency of various types of internal-combustion engines are given in the following tables.

(*a*) Small engines worked with petrol, benzol, methylated spirits, or paraffin. With a normal load the heat consumption per indicated horse-power hour varies between 16,000 B.Th.U. for engines of 1 or 2 h.p. and 9000 B.Th.U. for engines of 40 h.p. As a horse-power hour is equivalent to only 2550 B.Th.U. the indicated thermal efficiency is thus low. In addition to the work performed, some heat is recoverable from the cooling water and exhaust gases. The following figures were obtained with cooling water at 120° F. to 160° F. and a flow of 45 to 60 lb. per horse-power hour at normal engine output, increasing to 90 lb. per horse-power hour at half load. The temperature of the exhaust gases was between 650° F. and 1100° F. at full load and between 450° F. and 750° F. at half load. Their heat content varied from 6800 B.Th.U. at full load to 8000 B.Th.U. at half load per horse-power hour. The normal heat consumption was 14,000 B.Th.U. per indicated horse-power hour.

Heat Consumption, B.Th.U. per h.p.-hr.	Useful Work, Percentage.	Recoverable Heat, B.Th.U./h.p.-hr.		
		Cooling Water to 70° F.	Exhaust Gases.	Total Percentage.
10,000	25	2000	3200	52
12,000	21	2400	3800	52
14,000	18	2800	4400	51
16,000	16	3200	5000	51
18,000	14	3600	3600	51

The average fuel consumptions of small engines up to 50 b.h.p. are given below.

Type of Engine.	Lower Calorific Value.	Air/Fuel.	5 H.P.	10 H.P.	25 H.P.	50 H.P.
	B.Th.U./C. Ft.	C. Ft/C. Ft.	C. Ft./B.H.P. Hr.			
Coal gas	560	8·0	20	19·5	17	16·5
Anthracite gas ..	140	1·5	—	95	85	80
Coke gas	130	1·3	—	100	90	85
	B.Th.U./Lb.	Lb./Lb.	Lb./B.H.P. Hr.			
Paraffin (constant volume) ..	19,000	18	1·10	1·01	0·88	0·84
Crude oil (constant pressure)	18,000	18	0·53	0·48	0·44	0·42
Petrol	19,500	16	0·64	0·55	0·54	0·53
Raw spirit, 90 per cent ..	10,000	10	1·06	0·97	0·96	0·95
Vegetable oil ..	16,500	18	—	0·66	0·60	—

(*b*) Average data for larger engines from 50 to 5000 b.h.p., using the constant volume combustion cycle, are given in the following table.

Type of Engine.	Calorific Value, B.Th.U. per c. ft.	Gas Consumption, c. ft. per h.p.-hr.	Heat Consumption, B.Th.U. per h.p.-hr.	Heat Balance percentage.			
				Useful Work.	Friction.	Cooling Water.	Exhaust Gases.
Coal gas ..	500–650	18–24	11,000	22·6	6–10	35–43	30–40
Coke-oven gas ..	400–500	28	13,000	19·7	6–10	35–42	30–40
Generator gas ..	125–170	70–80	10,500	24·5	6–10	35–41	22–35
Blast-furnace gas ..	85–95	100	8,800	28·8	6–10	35–40	27–37

The mechanical efficiency of these engines is somewhat lower than that of the small engines, being 0·69 to 0·74 with full load and 0·55 to 0·64 with half load.

(*c*) Large engines using an approximation to constant pressure combustion

and consuming crude oil have consumptions of 0·39 to 0·40 lb. per brake horse-power hour. These also have mechanical efficiencies which are low compared to small engines using lighter fuels. The high pressures necessitate heavier construction, slower speeds, and greater dimensions, while considerable work is performed in injecting fuel or fuel and air at the high pressure of combustion. The mechanical efficiencies lie between 0·7 and 0·8. The table below illustrates the effect of variation in size of engine and of load on the efficiency as measured by the heat consumption in B.Th.U. per horse-power hour.

Normal H.P.	15 H.P.			70 H.P.			250 H.P.			300 H.P.			1000 H.P.		
No. of cylinders ..	1			1			4			3			4		
R.P.M.	230			170			350			160			125		
Load	½	¾	1	½	¾	1	½	¾	1	½	¾	1	½	¾	1
Heat consumption, B.Th.U./h.p.-hr.	11,300	10,000	9,100	8,800	7,900	7,600	9,500	9,500	9,100	9,400	—	7 600	9,600	8 700	8,500
Cooling water, 40° F. to 160° F., lb./h.p.-hr.	35	29	25	20	20	19·5	23	21·5	20	30	—	18·5	34	31	29
Exhaust gas temperature, °F.	—			480	570	700	550	660	790	550	730	930	380	460	560

The mechanical efficiencies at full load were between 0·74 and 0·78, at three-quarter load between 0·67 and 0·74, and at half load between 0·58 and 0·65. The work done by the air pump for a 15-h.p. engine was 14 per cent increasing to 28 per cent at half load, and for a 250-h.p. engine it was 8 per cent.

In order to make up a heat balance in a test it is necessary to measure the various heat losses. If w_1 is the weight of cooling water per horse-power hour and the temperatures at entrance and exit from the jacket are t_0 and t_1,

$$\text{cooling water heat} = w_1(t_1 - t_0).$$

Where no means are available for measuring the weight or volume of the exhaust gases, they could be deduced from the mixture strength, which is obtained from exhaust gas analysis as described in a previous chapter. The average specific heat of the gases is likewise furnished by the same information, the specific heats of the various constituents being known. If the volume of exhaust gas per horse-power hour is V and the proportion of N_2, CO_2, CO, and H_2O is represented by these symbols, and their specific heats per unit volume are s_1, s_2, s_3, and s_4 and their temperature t_2,

$$\text{exhaust gas heat} = V(s_1N_2 + s_2CO_2 + s_3CO + s_4H_2O)t_2.$$

In addition the latent heat of the water vapour must be added. The exhaust gas heat can also be found experimentally by utilizing the gas to heat a known quantity of water.

The energy wasted in friction, both the sliding friction of the moving

parts and the internal friction of the gases passing to and from the cylinder, can be found by running the engine without any load or by finding the work required to turn the engine at the same speed by an electric motor. Most of this friction remains in the gas or cylinder walls and is included in the measured losses from the water jacket and exhaust. The heat is conveyed from the burning gases to the cylinder wall and water jacket by conduction and radiation. The two portions, according to Schmolke, are given by the two terms of the following equation, where p represents the gas pressure, T_1 and T_2 the absolute temperatures of the gas and wall and A the area of the wall, v the mean piston velocity, and k_1, k_2, and k_3 are constants.

$$\text{Jacket loss} = k_1 p^{\frac{1}{2}} T_1^{\frac{1}{2}} (T_1 - T_2)(1 + k_2 v)A + k_3 (T_1^4 - T_2^4)A.$$

The increase in heat consumption per horse-power hour under partial load, which is shown in the last table, indicates that it is very uneconomical to run an engine except at full load, and the engine provision should be adjusted to the requirements in a manner to avoid much running at a low percentage of full load. If, for example, the daily requirements are 600-h.p. for 10 hr. and 200 h.p. for 14 hr., it is more efficient to have a separate small-power engine for the latter period rather than to use the 600-h.p. engine at one-third of its full load. Alternative arrangements would be to have either three 200-h.p. engines or one 200-h.p. and one 400-h.p. engine. The difference between the economical and the uneconomical arrangement can be found as follows, assuming that a 200-h.p. engine consumes 9000 B.Th.U. per horse-power hour at full load and that a 600-h.p. engine consumes 8000 B.Th.U. per horse-power hour at full load and 10,000 B.Th.U. at one-third load.

(1) 600-h.p. engine at full load for 10 hr.	=	48×10^6 B.Th.U.
600-h.p. engine at $\frac{1}{3}$ load for 14 hr.	=	28×10^6 ,,
Total	=	76×10^6 ,,
(2) 600-h.p. engine at full load for 10 hr.	=	48×10^6 B.Th.U.
200-h.p. engine at full load for 14 hr.	=	$25{\cdot}2 \times 10^6$,,
Total	=	$73{\cdot}2 \times 10^6$,,

Saving = $2{\cdot}8 \times 10^6$ B.Th.U.

Using fuel of 18,000 B.Th.U. per pound, saving of fuel = 155 lb.

Common types of internal-combustion engines are illustrated in figs. 19 to 26. Liquid and gaseous fuels only are used. Attempts to use solid fuel injected in a finely divided state have been unsuccessful as it is impossible to burn it with sufficient speed. Solid particles settle on cylinder walls and within the jets, leading to contraction of the bore of the jet nozzles. Solid fuel must therefore be liquefied or gasified before use in an engine. Liquid fuels must be thoroughly atomized and as much as possible vaporized to ensure a homogeneous mixture of fuel and air for complete and quick combustion. The fuel is introduced in correct amount by means of carburettors or of injection pumps. The ignition is produced either by the passage of a

spark within the cylinder or spontaneously by the heat of compression. The timing of the ignition is of great importance. If it occurs too early while the piston is still moving towards the end of the cylinder, there is a sudden abnormal rise of pressure tending to stop the engine. If ignition occurs late a smaller explosion pressure is obtained but higher pressures and temperatures towards the end of expansion. Hence the amount of combustion gases retained is greater and the amount of fresh mixture which can enter is less. As the quantity of air which can pass through the engine is a fixed quantity determined by the dimensions and design, the carburettor or pump should deliver only such an amount as can unite with this air, any excess

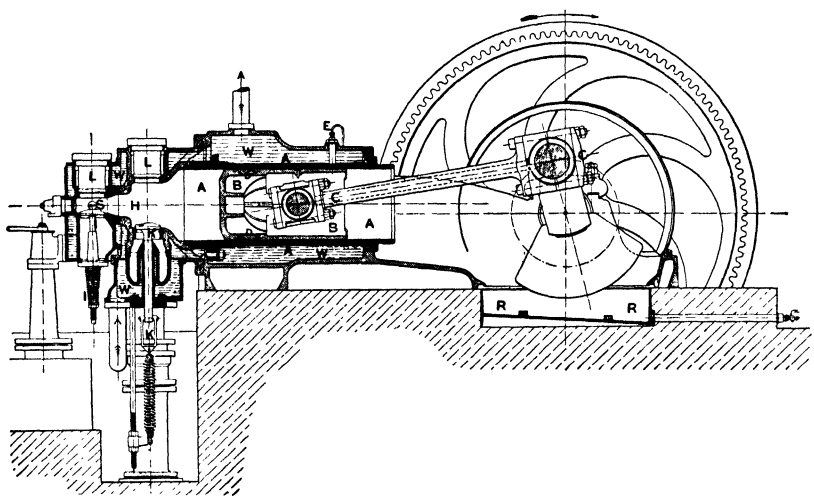


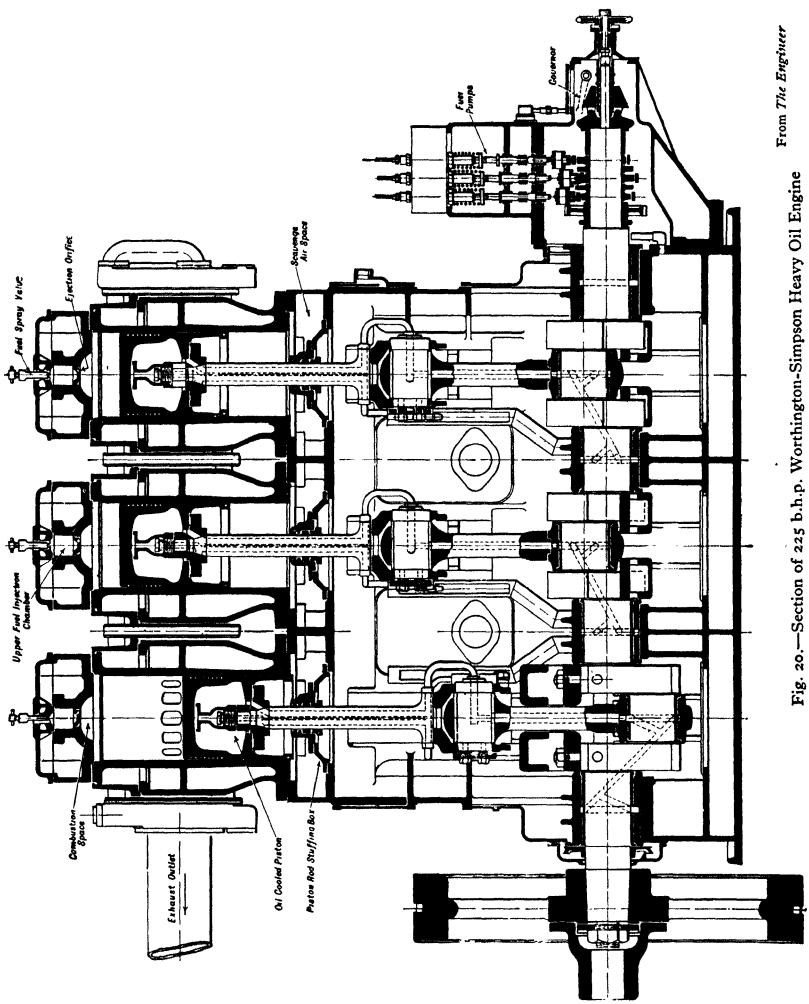
Fig. 19.—Horizontal National Gas Engine, 100 b.h.p.

amount being waste and any less amount resulting in a loss of possible power.

Methods of measuring the output of an engine, i.e. the brake horse-power and also the indicated horse-power, as well as methods of measuring the input in the form of calorific value of the fuel, are described in a later chapter.

In fig. 21 is shown a sectional elevation of the Ruston-Proctor engine. This engine has a cylinder of 21·5-in. bore and 30-in. stroke, and develops 150 b.h.p. as a maximum working load when running with producer gas. It represents about the largest size of cylinder which can be employed without resorting to water-cooling of the piston—a condition which practically marks the limit of size for this type of engine. In so far as the general construction and mechanical details are concerned, this engine differs very little from others of the same class. The crank-shaft is not a solid forging, but is built up, the crank webs and balance weights being steel castings bored out and shrunk on to the crank pin and shaft. The connecting-rod big-end bearing is of the marine type, with separate cast-steel housings for the bearings.

The connecting-rod itself has a flat foot, and the two halves of the bearing are, of course, lined with white metal. The design of the combustion chamber is generally similar to that of other makes, but it will be noticed that the water-jacket is left open at the back and afterwards closed by separate detach-



From *The Engineer*

Fig. 20.—Section of 225 b.h.p. Worthington-Simpson Heavy Oil Engine

able covers. This is done to avoid tying the inner and outer walls rigidly together, otherwise the irregular expansion will set up severe stresses which may lead to cracking. This problem of the unequal expansion of the inner and outer walls of the cylinder does not apply to any appreciable extent in small engines, but increases in importance as the size of the engine is increased, until, in very large engines, it becomes the controlling factor.

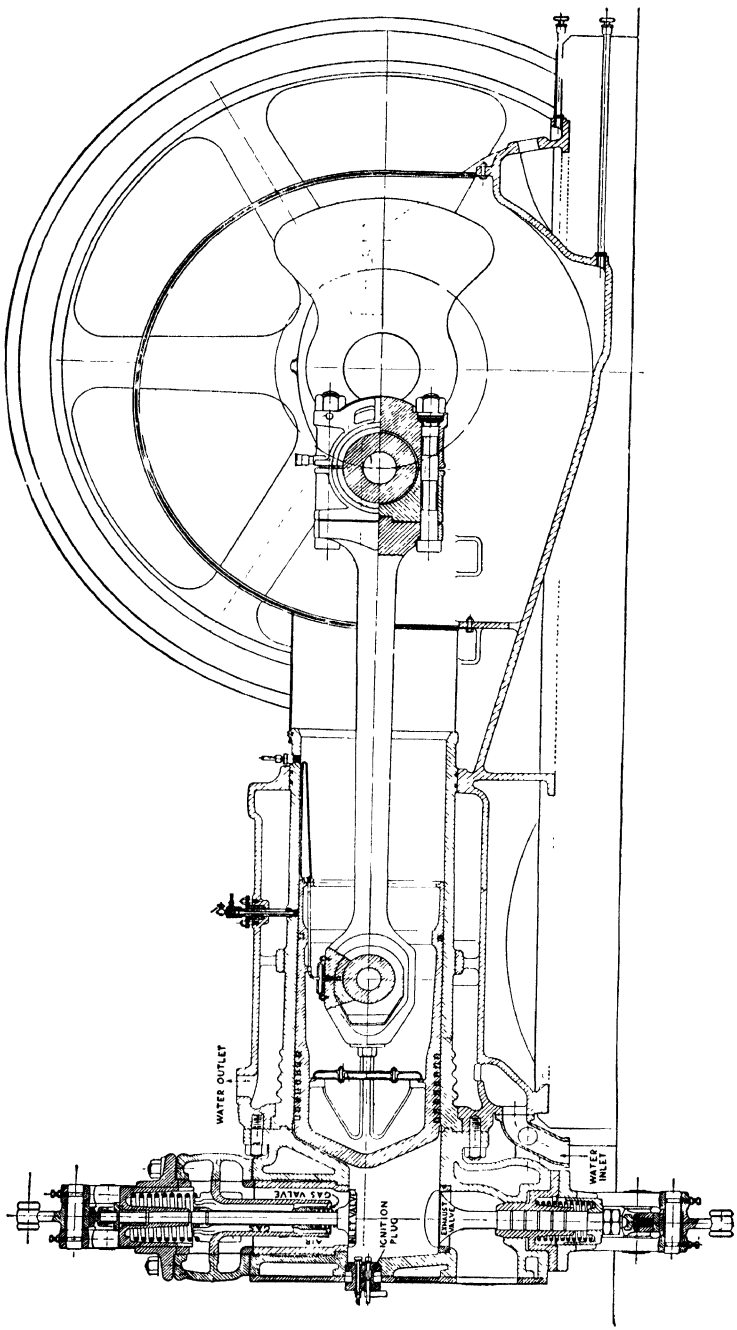
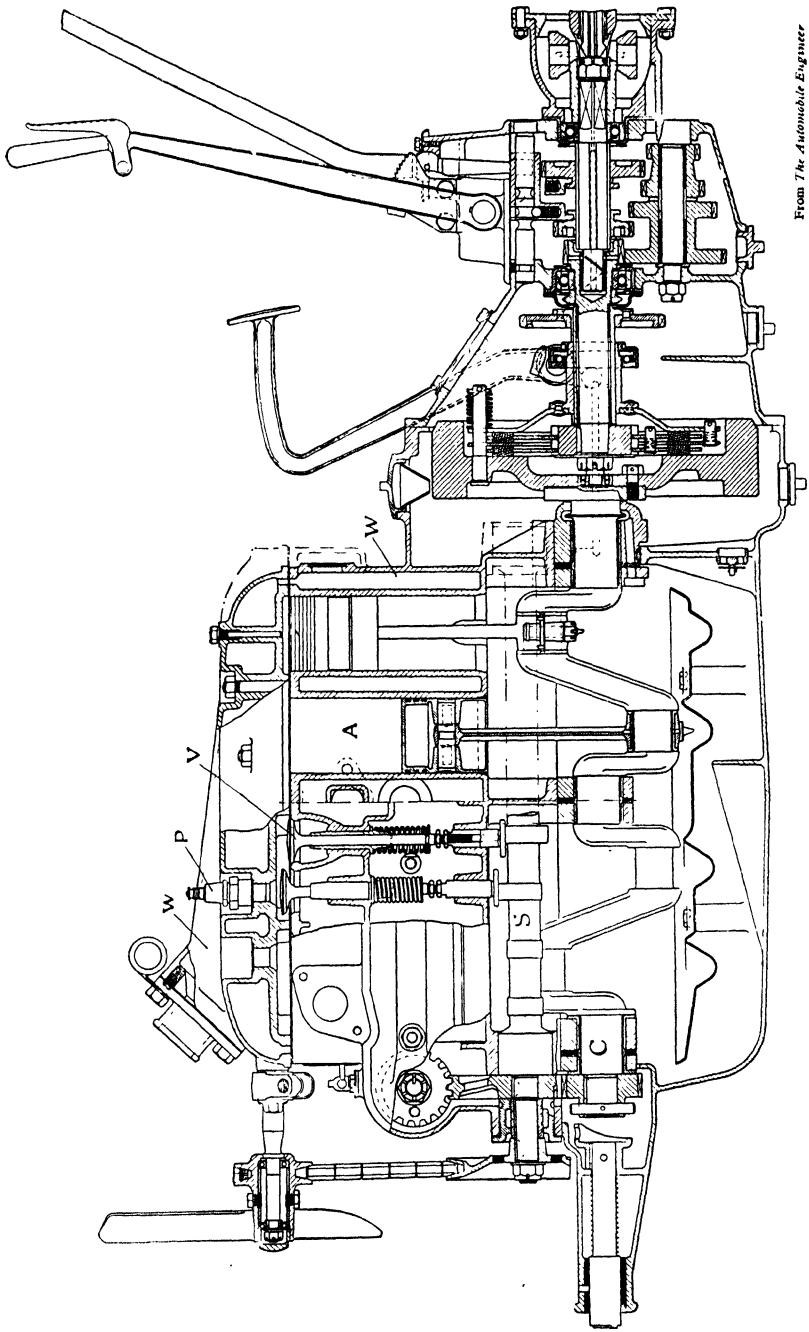


Fig 21.—Section Elevation of a 150 b.h.p Ruston-Proctor Gas Engine



From *The Automobile Engineer*

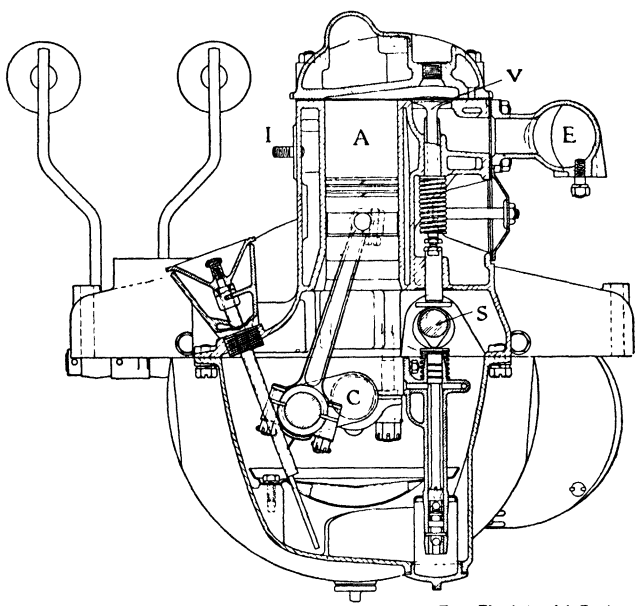
Fig 22.—General Arrangement of Morris-Cowley Power Unit. Bore, 69.5 mm; Stroke, 102 mm

The main inlet valve carries a supplementary gas valve, and the quantity of mixture is controlled by the lift of the two valves, which, again, is under the control of the governor. Both the inlet and exhaust valves are operated from a single cam on the side-shaft, an arrangement which makes it possible to place both valves in the same line and operate them direct through straight levers. The exhaust valve is very massive, both in the head and stem, with the object of maintaining an even temperature and of getting rid of as much heat as possible down the stem. It is one of the difficulties with the larger sizes of gas-engines that, in order to prevent wide variations of temperature in the working parts, with the consequent risk of both pre-ignition and failure through unequal expansion, it is necessary to make all parts which are exposed to high temperatures, and which cannot conveniently be water-cooled, very thick and heavy. The piston is lubricated by means of a forced-fed lubricator, actuated from the side-shaft, which delivers oil to the top side of the piston. The gudgeon pin is lubricated by means of a sight-fed lubricator near the open end of the liner. It will be noticed that the head of the piston is enclosed by means of a light detachable cover, the object of this being to prevent any oil from reaching the under side of the piston-head where it would carbonize. This engine runs normally at a speed of 175 r.p.m., corresponding to a piston speed of 875 ft. per minute, which is high for so large an engine of this type. The brake mean pressure is 72 lb. per square inch, and taking the mechanical efficiency as 85 per cent the indicated mean pressure will be approximately 85 lb. per square inch. Such engines cannot run with very strong mixtures owing to the high temperatures involved and the risk of pre-ignition from overheating of the uncooled parts.

A typical small petrol-engine is shown in figs. 22 and 23, which represent the arrangement of a four-cylinder water-cooled motor-car engine developing about 23 b.h.p. at 3500 r.p.m. The total stroke volume of the four cylinders is 0·055 c. ft. or 1550 c. c., and the indicated mean effective pressure about 70 lb. per square inch. The crank-shaft C is slightly offset from the axis of the cylinders A. The valves V are operated by the cam-shaft S. The intake for the mixture is shown at I, the carburettor being removed, while the exhaust manifold E is on the opposite side of the engine. The sparking plugs P are in the cylinder ends and operated by a magneto not shown. The circulation in the water-jacket W is simply by convection.

Figs. 24 and 25 illustrate a modern successful high-powered 9-cylinder air-cooled radial engine made by the Bristol Aeroplane Company. The working barrels of the cylinders are of steel machined from solid billets and are fitted with aluminium alloy heads. Of the nine steel connecting-rods, of the usual H-section, one, the " master " rod, is integral with a massive white-metal lined big-end to which the remaining eight rods are simply articulated. In the head of each cylinder are two inlet and two exhaust valves operated through push-rods and rockers. The push-rods are operated by a drum having two rings each of four lobes, driven by an epicyclic wheel in the reverse direction to that of the crank-shaft and at one-eighth of its speed. Ignition is by two nine-point British Thomson-Houston magnetos, each

serving one of the two sparking plugs of each cylinder, and the order of firing is 1, 3, 5, 7, 9, 2, 4, 6, 8, 1, . . . Three Claudel carburettors are fitted to the engine. Cylinders 4, 1, and 7 are supplied by one carburettor, 3, 9, and 6 by a second, and 2, 8, and 5 by a third. The suction impulses accordingly occur in each carburettor at equal intervals of 240° of crank-shaft revolution; the petrol vapour and air are well mixed in a circular three-start endless helix; and the engine is virtually divided into three three-cylindered groups each of which alone is capable of working independently if the others should fail. The normal output of 400 b.h.p. is obtained at a speed of 1575 r.p.m.,



From *The Automobile Engineer*

Fig. 23.—Cross-Section of Morris-Cowley Engine

but the engine develops a maximum of 436 b.h.p. at 1750 r.p.m. The fuel consumption at the normal output is 0·57 pt., and the consumption of lubricating oil is 0·03 pt., both per brake horse-power per hour. At the normal output the weight per horse-power is 1·95 lb.

Heavy-oil engines working with some approximation to constant pressure combustion are now of very many types. Air only is first admitted to the engine and compressed to a very much higher pressure than in petrol-engines. Near the end of compression the fuel is introduced by a high-pressure pump and, after starting up, ignition takes place spontaneously by the heat of compression and the heat of the cylinder walls. The rotation necessary to start up large engines is obtained by compressed air which has been compressed during a previous run and stored in steel cylinders. Some types of engine can be started by the heat of compression only, but others require

the temporary application of external heat either by an electrically heated wire or by a blow-pipe applied to a bulb addition to the cylinder. In original Diesel engines some air was injected with the fuel, thus assisting the atomization, but in many cases fuel only is now injected and other means are

Fig. 24.—400 h.p. Bristol Jupiter Aeroplane Engine

adopted for breaking up the jet and distributing it throughout the cylinder. An oil-engine giving 225 b.h.p. at 275 r.p.m. is shown in fig. 20. In this type the fuel is injected into a small chamber which is above the main cylinder space and connected to it by a small orifice. When compression occurs in cylinder and in chamber a small portion of the oil is ignited and the force

of its explosion expels the remainder through the orifice and throughout the cylinder. This engine works with a two-stroke cycle and uses 0·44 lb. of oil per brake horse-power hour. On account of their economical use of fuel and the elimination of the manual labour of stoking, heavy-oil engines are

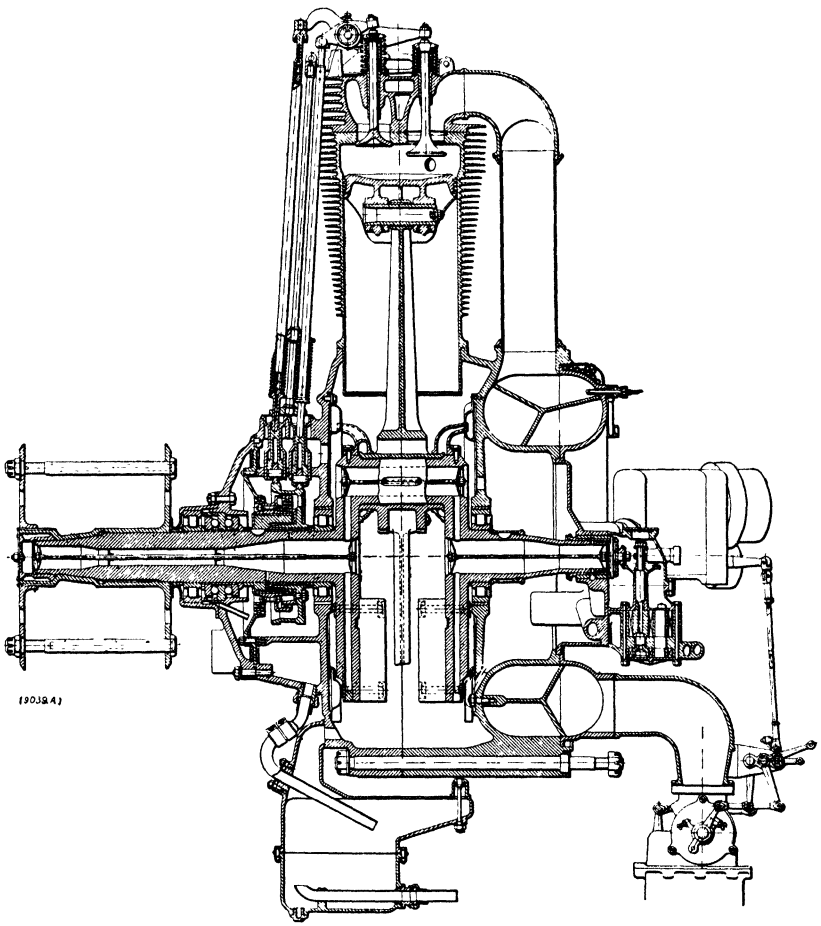


Fig. 25.—Sectional View of 400 h.p. Bristol Jupiter Aeroplane Engine

gradually superseding steam-engines on ships. A marine oil engine giving 13,500 b.h.p. at 127 r.p.m. is shown in fig. 26. Its fuel consumption is only 0·392 lb. per brake horse-power.

4. THE INTERNAL-COMBUSTION TURBINE

The advantages of a continuous impulse and rotary motion over intermittent impulses and reciprocating motion have induced many attempts to produce an internal-combustion turbine, which would have the additional advantage not possessed by the steam-turbine of the elimination of furnace and boiler losses. These attempts have so far not been sufficiently successful to compete with the reciprocating internal-combustion engine. In the case of the steam-turbine the addition of heat resulting in a change of state from liquid to vapour produces high pressures without very high temperatures. Apart from temperature effects in the combination of air and fuel by combustion, there is very little change of volume at constant pressure or very little increase of pressure at constant volume. Hence high pressures can only be obtained accompanied by high temperatures. The materials at present available for turbine blades are not capable of enduring the temperatures which would exist in reaction turbines utilizing high pressures, and it is therefore necessary to use the impulse form of turbine in which the pressure energy is transformed to kinetic energy outside the turbine and the pressures within the turbine are approximately atmospheric. To produce a high efficiency it is necessary that the air and fuel be compressed before ignition. In the reciprocating engine the compression is done efficiently in the combustion chamber. The same difficulties which necessitate the use of the impulse turbine render it necessary to have a separate compression and combustion chamber with much increased losses. Piston compressors are more efficient than any existing type of rotary compressor, but as they would introduce the intermittent feature which the turbine is intended to eliminate, the rotary form is the more desirable. The elements of an internal-combustion turbine system thus take the form shown diagrammatically in fig. 27. The air and fuel are supplied to the combustion chamber at a constant rate, and the products of combustion expanded through the nozzle and caused to impinge on the vanes of the impulse turbine. The compressors for air and fuel are driven from the turbine shaft. This system of constant pressure combustion in which the whole action is continuous has obviously many advantages, though the combustion chamber and nozzle have to continuously withstand high temperatures. Holzwarth has constructed a turbine in which the action in the combustion chamber is intermittent. A fixed

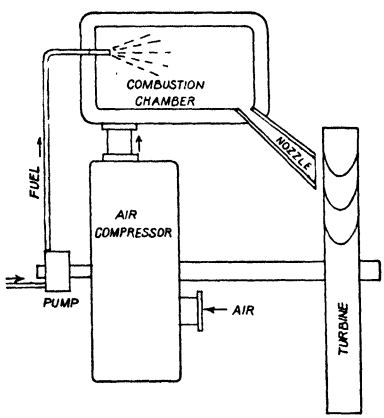


Fig. 27.—The Elements of an Internal-combustion Turbine

amount of air and fuel is introduced into the closed chamber, combustion takes place at constant volume, and the valve leading to the nozzle automatically opens for the flow of the combustion gases to the turbine. The work of compression, being against the pressure previous to combustion, is thus reduced. In spite of the use of a governor which opens the exit valve as the pressure falls, the difference between gas and vane velocity towards the beginning and end of flow must result in loss of energy due to shock.

If fuel of calorific value H B.Th.U. per pound requires w lb. of air per pound of fuel, and C_p is the mean specific heat of the products of combustion between the initial temperature t_0 and the combustion temperature t_1,

$$t_1 = t_0 + \frac{H}{(1 + w)C_p}.$$

If the expansion from the pressure p_1 within the combustion chamber to the pressure p_2 at the exit from the nozzle can be assumed adiabatic and $C_p/C_v = \gamma$, the temperature at the exit from the nozzle t_2 is given by

$$\frac{460 + t_1}{460 + t_2} = \left(\frac{p_1}{p_2}\right)^{\frac{\gamma-1}{\gamma}}.$$

The velocity of efflux from the nozzles U is given by the heat drop $C_p(t_1 - t_2)$ per pound.

$$U^2 = 50{,}000\ C_p(t_1 - t_2).$$

If the outlet velocity from the turbine wheel is U_1, the energy given up is

$$W = \tfrac{1}{2}\frac{1 + w}{g}(U^2 - U_1^2) \text{ ft.-lb.}$$

The theoretical values are modified at all stages by the heat losses and frictional resistance. Tests on steam-turbines would indicate that not more than half the above work could be realized, and of this some must be utilized in compression of the air and fuel. If p_0 is the atmospheric pressure the work of compressing w lb. of air initially occupying v_0 c. ft. is for isothermal compression

$$W_1 = p_0 v_0 \log_e p_1/p_0.$$

In practice the work required varies from $1{\cdot}5W_1$ to $1{\cdot}7W_1$. The work of compression of the fuel W_2 is of the order 0·5 per cent to 1·5 per cent of the turbine output. Thus the actual work obtained is $0{\cdot}5W - 1{\cdot}6W_1 - W_2$ and the thermal efficiency is

$$\frac{0{\cdot}5W - 1{\cdot}6W_1 - W_2}{778H}.$$

Little information is available concerning the actual performances of the few gas-turbines that have been constructed, and the figures published are of doubtful value. The Holzwarth turbine was designed to give 1000 h.p.

at 3000 r.p.m. with explosion pressures up to 14 atmospheres. It is said to have realized 450 h.p. and an efficiency of 25 per cent, but this latter is probably grossly overestimated. In the Armengand and Lemele turbine, steam was added to the products of combustion and the turbine worked by the mixed fluid. At 20,000 r.p.m. with combustion pressure of 5 atmospheres, 25 h.p. was attained, but the consumption was very high and the efficiency only 9·5 per cent. A larger design running at 4250 r.p.m. gave a still smaller efficiency. The Rateau turbine is designed to utilize the exhaust gas of the ordinary reciprocating internal-combustion engine. These gases escape at a pressure a little above atmospheric and their energy is normally wasted. They are used to drive a turbine, and on the same shaft is a centrifugal air pump which can thus feed the engine with compressed air. A greater amount of fuel can be burnt within the engine with the denser air supply and a corresponding amount of work obtained. To obtain an appreciable amount of power from such a turbine, the back pressure on the reciprocating engine is necessarily increased and this tends to decrease its performance, but the gain due to the increase of air and petrol consumption is very much greater. This system is in course of development for high-altitude aeroplane engines. The decrease of atmospheric density with altitude results in a corresponding diminution in the power of the normal aeroplane engine, but by compressing the supply of air with the turbine and pump, ground-level power can be maintained. The Rateau turbine is similar to the ordinary centrifugal air pump. It works at a high speed of 27,000 r.p.m. and has an efficiency of only 2·5 per cent. The future development of the internal-combustion turbine depends upon the attainment of high efficiencies in the compressors and on the production of blade material which will withstand higher temperatures and thus eliminate some of the losses due to having a separate combustion chamber.

5. HOT-AIR ENGINES

The hot-air engine suffers from the same disadvantage as the internal-combustion engine in that high pressures can only be obtained accompanied by high temperatures, while in addition there is the difficulty of heating air in quantity by an external source. Hence, such engines as have been constructed have been small and inefficient, and none is now used for any commercial purpose. Engines have been made with the constant-volume cycle as in the internal-combustion engine, and also with the constant-pressure cycle similar to that of the steam-engine. Fig. 28 represents the indicator diagram in the latter case, which consists of two constant-pressure and two adiabatic curves for the perfect engine. The useful work per pound of air

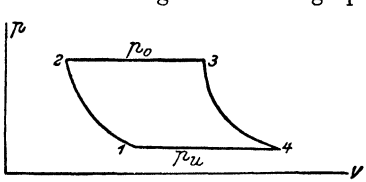


Fig. 28.—Indicator Diagram of Hot Air Engine

$$W = C_p[(t_3 - t_2) - (t_4 - t_1)],$$

where $$\frac{v_1}{v_4} = \frac{v_2}{v_3} = \frac{T_1}{T_4} = \frac{T_2}{T_3},$$

and $$\left(\frac{p_1}{p_2}\right)^{\frac{\gamma-1}{\gamma}} = \frac{T_2}{T_1} = \left(\frac{v_1}{v_2}\right)^{\gamma-1}.$$

In an actual case γ is replaced by the expansion index n. The best performance obtained for a hot-air engine is a brake thermal efficiency of 8 per cent with an engine giving 14 b.h.p. Other engines of smaller power have also a much smaller efficiency.

6. EFFICIENCIES AND USEFUL HEAT

It has already been mentioned that in commercial practice a high engine efficiency is not always of primary importance. If low-temperature heat can be utilized a high overall efficiency may be attained with a low engine efficiency. In a comparison of various types of engines it is therefore desirable to consider not only the proportion of the heat given which is converted into work but also the proportion of the exhaust heat and the heat losses to walls and water which may be utilized under favourable circumstances. The following figures represent the results attained and the range over which the performance varies for engines of different size and design. The figures given which indicate the most efficient cases are usually those attained only under special circumstances of test, and are not attainable under ordinary running conditions.

Type of Engine.	Heat Consumption in B.Th.U. per B.H.P. hour.	Brake Thermal Efficiency, per cent.	B.H.P. hours, per 100,000 B.Th.U.
Diesel engines	7,200– 9,200	35·1–27·5	13·9–10·9
High - pressure condensing steam-engines and -turbines	8,000–11,200	31·5–22·5	12·5– 8·9
Gas engines	9,200–12,400	27·5–20·4	10·9– 8·1
Internal - combustion engines (constant volume)	11,600–14,400	21·8–17·6	8·6– 6·9
Steam-turbines (medium) ..	12,400–17,200	20·4–14·7	8·1– 5·8
Compound steam - engines, superheated, condensing ..	13,600–18,400	18·6–13·8	7·4– 5·4
Compound steam - engines, superheated, exhaust ..	15,200–24,400	16·6–10·3	6·6– 4·1
Single-cylinder steam-engines, superheated, exhaust ..	24,000–29,600	10·5– 8·6	4·2– 3·4
Single-cylinder steam-engines, saturated, condensing ..	25,400–33,200	9·6– 7·6	3·9– 3·0
Single-cylinder steam-engines, saturated, exhaust	29,200–40,000	8·7– 6·3	3·4– 2·5

Of the heat which is not transformed into useful brake horse-power,

some is contained in the exhaust gases or steam, some in the cooling water of internal-combustion engines or condensers, and some is wasted in mechanical friction of the engine. A little of the last may find its way to the water or the exhaust, but it may generally be regarded as irrecoverable. Together

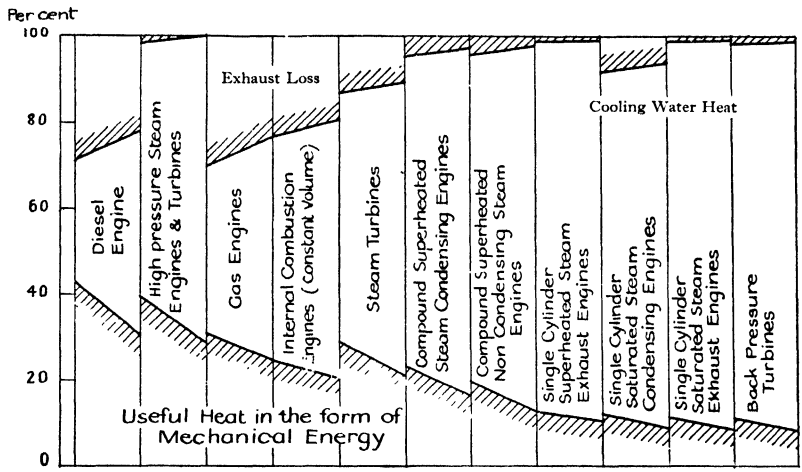


Fig. 29.—Heat Distribution

with the brake horse-power it forms the indicated horse-power. In many cases the two other quantities are also total losses, but in other cases some may be recovered. The following table indicates the possibilities in this direction. The figures are also shown diagrammatically in fig. 29.

Type of Engine.	Heat Distribution, Per Cent.		
	I.H.P.	Water.	Exhaust.
Diesel engines	44–30	28–48	28–22
High-pressure condensing steam-engines and -turbines	40–29	59–70	1·5–1
Gas-engines	30–24	38–52	32–24
Internal-combustion engines (constant volume)	24–20	52–60	24–20
Steam-turbines (medium) ..	29–21	57–68	14–11
Compound steam-engines, superheated, condensing	23–17	71–79	6–4
Compound steam-engines, superheated, exhaust	21–13	74–84	5–3
Single-cylinder steam-engines, superheated, exhaust	13–11	85–88	2–1·5
Single-cylinder steam-engines, saturated, condensing ..	12–9	78–83	10–8
Single-cylinder steam-engines, saturated, exhaust	11–8	87–91	2–1·5

The figures are independent of the furnace and boiler efficiency, which still further reduces the performance of the steam-engines relative to the various types of internal-combustion engines.

7. DIRECT-ACTION PUMPS

Any form of heat engine can, of course, be used to drive a pump for circulating or raising liquids and gases. The consideration of such pumps, which are separate entities and may equally be driven by electrical or other means, is outside the scope of this work. In certain instances, however, the action is of a more direct character. One case is that of injector pumps used for raising water and feeding it into boilers against the prevailing pressure within. The action is shown diagrammatically in fig. 30. Steam passes through the pipe S, emerges from the nozzle N with a high velocity, and mixes with the water drawn up by the suction through pipe W. The difference in the velocities of steam and water results in a large waste of mechanical energy on mixing, but sufficient momentum remains to carry the mixture through the reversed nozzle M in which the diminution of kinetic energy and increase of pressure energy is sufficient to overcome the boiler pressure and to open the valve V. By suitable arrangements the pumps can be made self-starting and automatic. One type of pump is shown in fig. 31. Other types, with a series of nozzles, are designed to use exhaust steam, a small amount of live steam being added, if necessary, against high pressures. The proportion of the total heat of the steam which is transformed into mechanical work is extremely small, but as the heat is returned to the boiler this is not of importance. If the pump is used merely to raise water and the heat of the mixture of steam and water is wasted, the efficiency is, as stated, small, but it may be noted that with an ordinary small steam-engine of 7 per cent brake thermal efficiency, combined with a pump of 75 per cent mechanical efficiency, the overall efficiency is only 5 per cent. The ejector, therefore, in the case of small plants is preferable owing to its simplicity.

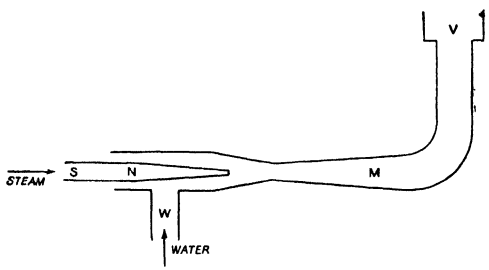


Fig. 30.—Principle of the Steam Injector

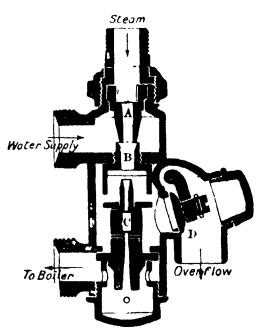


Fig. 31.—Penberthy Injector for Boiler Feed Water

A Steam jet; B, Suction jet; C, Delivery jet; D, Overflow valve.

A steam-pump of a different character is shown in fig. 32. Water is fed into the two chambers A through the valves E. In the period shown in the diagram steam is passing from K by means of the ball valve, depressing the water in the right-hand chamber and forcing it out through the exit D. When the water-level falls so low that the steam can escape to D, the turbulence and mixing of steam and water causes condensation, the ball valve closes that chamber, and the condensation is completed while the chamber refills with water. Meanwhile the operation is occurring in the left-hand chamber, the water being forced down and out of D.

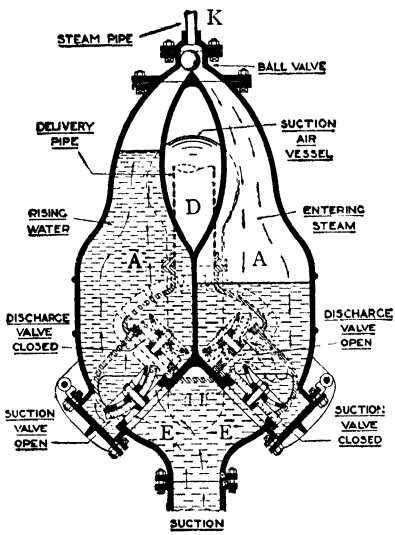


Fig. 32.—Section of Pulsometer Steam Pump

The internal-combustion engine cycle has been applied successfully for the direct raising of large quantities of water in the Humphrey pump. The cylinder head with valves, air intake, and exhaust and ignition apparatus is similar to that of any large gas-engine, but the surface of the water forms the piston and the momentum given by the explosion of the gas carries the water to the required height. The system is shown diagrammatically in fig. 33. The frequency of the system is determined by the oscillations of the water in the discharge pipe. The products of combustion flow out through the exhaust valves during the expulsion of the water and the first return movement, fresh gas and air is drawn into the cylinder during the second outward movement and compressed during the second return, when ignition occurs and the cycle is repeated. Single-cylinder pumps working at 15 cycles per minute and giving 250 to 300 h.p., working with gas from a gas producer, require about 0·9 lb. of anthracite per hour per horse-power of actual pumping work.

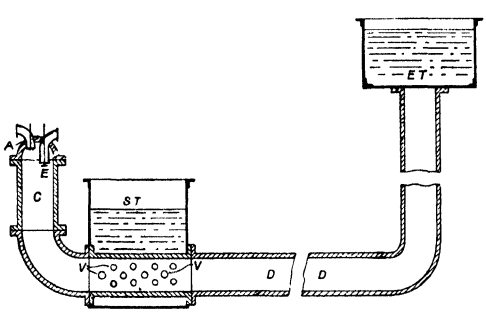


Fig. 33.—Humphrey Internal Combustion Pump

A, Inlet valve for air and gas. E, Exhaust valve. C, Combustion cylinder. ST, Supply tank. V V, Inlet valves for water. D, Delivery pipe. ET, Delivery tank.

CHAPTER VII

Heat of Steam and Water

Reference has already been made in several cases to steam condensers attached to either reciprocating engines or steam-turbines. In the latter case, the condensation of the steam is necessary to maintain the high vacuum which is required for high efficiency. In other cases, the steam is condensed also to conserve the supply of water. The usual town water supply often contains a considerable quantity of lime and other salts in solution which are deposited on the inside of the boilers. The condensed water is free from these solids and can be used again without any further addition to the deposits. In marine boilers, the deleterious effect of salt prohibits the use of sea-water, and the conservation of the fresh water is even more important. In heat accumulators the primary object is the conservation of the heat from the exhaust of reciprocating engines or from boilers intermittently employed. The accumulators may take the form of condensers, the heat being given to water, or may be simple steam-holders. The winding engines of mines and the engines of rolling mills are instances where an engine is used at frequent short intervals, and where the steam production of the boiler during the idle intervals is of sufficient importance to be worth utilizing. There is

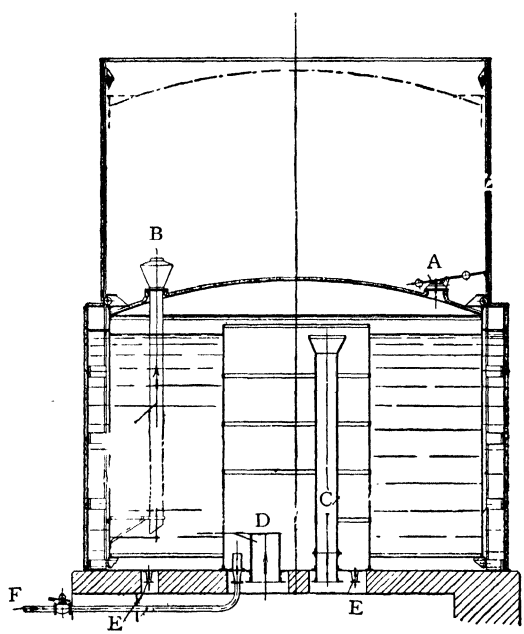


Fig. 1.—Constant Pressure Steam Accumulator (*Harlé-Balcke* System)

A, Inlet valve. B, Steam outlet with full accumulator. C, Steam outlet. D, Steam Inlet E, Condensed water discharge. F, Fresh steam inlet.

necessarily a loss of temperature and pressure in the heat accumulator, but the low-pressure steam-turbine forms a convenient and efficient way of obtaining whatever energy is still available.

1. STEAM HEAT ACCUMULATORS

Exhaust and waste steam can only be stored in the form of steam when the quantity is small, owing to its large volume. The constant-pressure

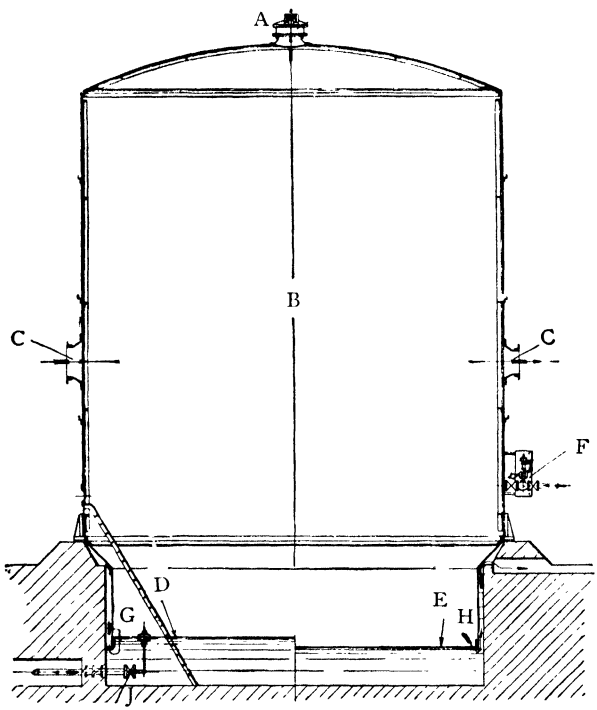


Fig. 2.—Constant Volume Accumulator

A, Safety valve. B, Steam chamber. C, Steam outlet. D, Water level with pressure below atmospheric. E, Water level with pressure above atmospheric. F, Fresh steam inlet with Servo-motor. G, Air. H, Steam. J, Float valve for discharging the surplus condensed water.

type of accumulator is shown in fig. 1. It is similar to a gas-holder, the steam being stored in a floating bell which rises or falls according to the variation in the steam supply. The constant-volume accumulator is illustrated in fig. 2. The pressure of the steam varies according to the quantity within, being limited of course to the pressure of the supply. With both forms, the chambers must be well insulated by suitable non-conductors of heat. Double-walled chambers with a quiescent layer of air between are used in some cases. Experiments on one type of constant-volume accumulator showed a loss of 0·05 lb. of steam per square foot per hour, or 0·8 per cent

per hour for the size of accumulator tested. The accumulators require, as do gas-holders, safety valves to prevent the accidental formation of too high or too low a pressure. The formation of a partial vacuum either due to excessive condensation or to a rapid withdrawal of steam may result in collapse.

2. WATER HEAT ACCUMULATORS

In water accumulators the heat of exhaust or excess steam is stored by condensing it and thus increasing the temperature of a large mass of water. The chamber is closed, and the pressure of steam and water thus increases as more steam is introduced and the temperature of the water rises. If the chamber is now connected to a space at a lower pressure, evaporation takes place and steam is re-formed, though its pressure is lower than that of the original steam. The new steam is saturated, but it is possible to produce superheated steam by the addition of a regenerator. If the original steam is superheated, it can first be passed over masses of iron or other material before being condensed in the accumulator, these masses being raised to a temperature above that of the saturated steam which issues from the accumulator. The saturated steam then becomes superheated by flowing over or through them and abstracting their heat.

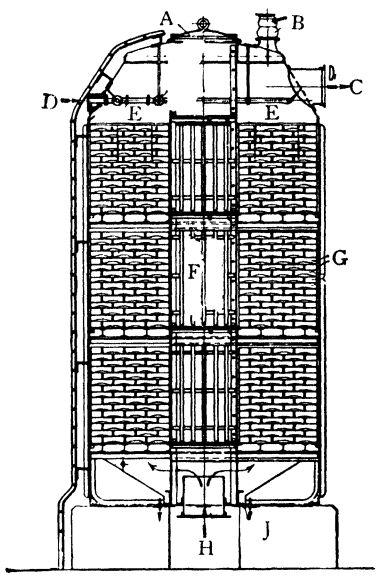


Fig. 3.—Rateau Plate Accumulator

A, Supply inlet. B, Safety valve. C, Steam outlet. D, Water for filling the plates. E, Steam. F, Supply shaft. G, Water plates. H, Steam inlet. J, Condensed water.

A simple form of accumulator is that due to Rateau, a vertical plate design being shown in fig. 3. The narrow spaces between the plates are filled with water and there is a large condensing surface relative to the volume of the accumulator and mass of water. In fig. 4 is shown a horizontal accumulator in which the steam is condensed in tubes surrounded by water in rapid circulation. The usual variation in pressure within these accumulators from the time of maximum to the time of minimum heat content may be as much as half an atmosphere with a corresponding temperature variation. If the entering steam is the exhaust of a reciprocating engine at approximately atmospheric pressure, the latent heat which is to be abstracted by the mass of water in the accumulator is about 970 B.Th.U. per pound of steam. Assuming that a temperature variation of 35° F. is allowable, a water content of 28 lb. per pound of steam is requisite. The magnitude of the allowable variation depends upon the use to which the new low-pressure steam is put.

If the steam is driving a low-pressure steam-turbine and is thus being withdrawn continuously, a large periodic initial pressure variation results in loss

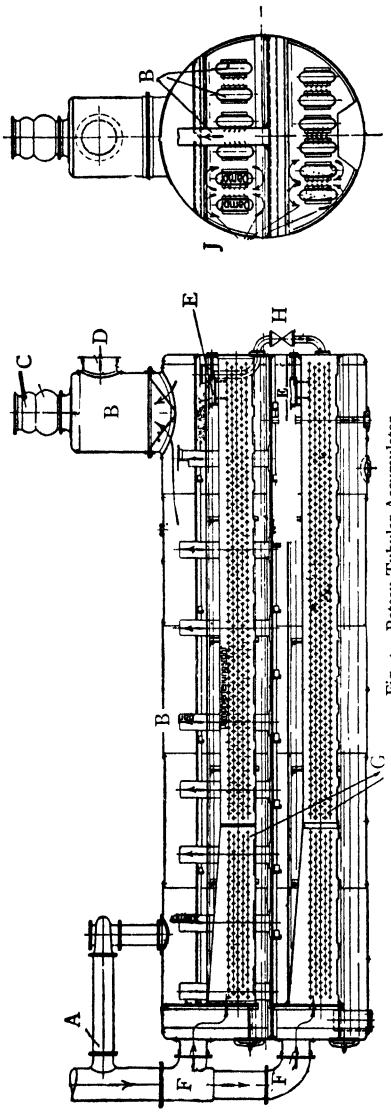


Fig. 4.—Rateau Tubular Accumulator

A, Pressure equalizing pipe. B, Steam. C, Safety valve. D, Steam inlet. E, Oilskimming Apparatus. F, Steam inlet. G, Steam distributing pipe. H, Overflow pipe. J, Water agitator

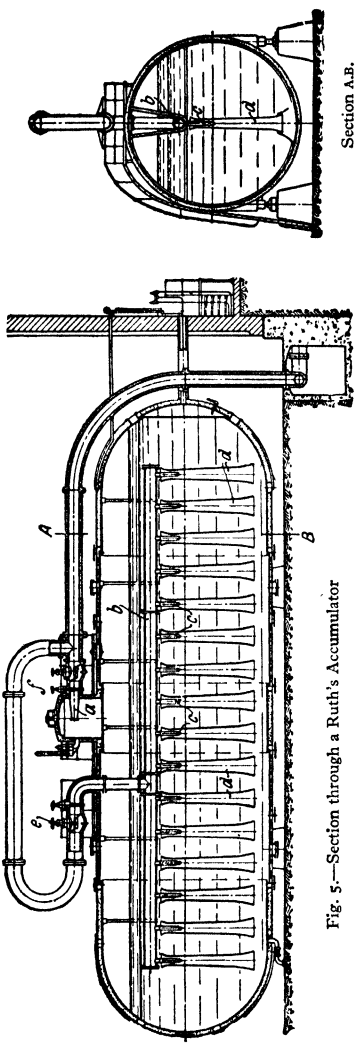


Fig. 5.—Section through a Ruth's Accumulator

of efficiency. In this case the variation should be reduced to narrow limits.

Another form of water accumulator is shown in fig. 5. The water is contained in a cylindrical boiler which is protected by insulation, and which

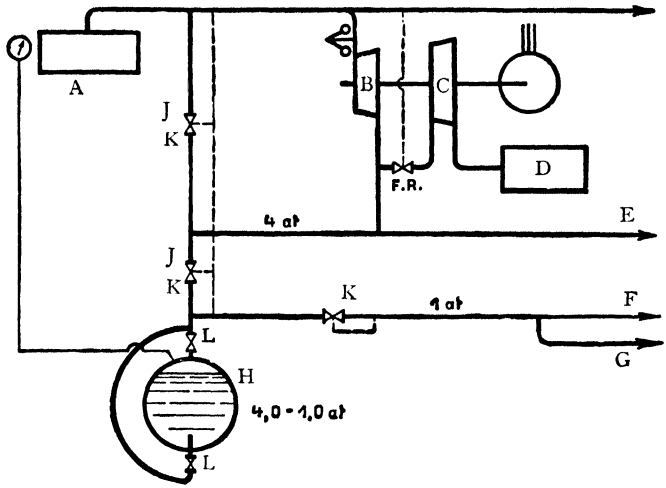


Fig. 6.—Control Diagram of a Textile Factory with Ruth's Accumulator (M.A.N., Augsburg, Nurnberg)

A, Boiler. B, High-pressure turbine. C, Low-pressure turbine. D, Condenser. E, Dressing plant. F, Dyeing plant. G Heating. H, Accumulator. J, Overflow valve. K, Pressure reducing valve. L, Check valve.

is filled to about 90 per cent of its volume. The steam is introduced to the water through the diffusor tubes *d*, and the necessary circulation of the

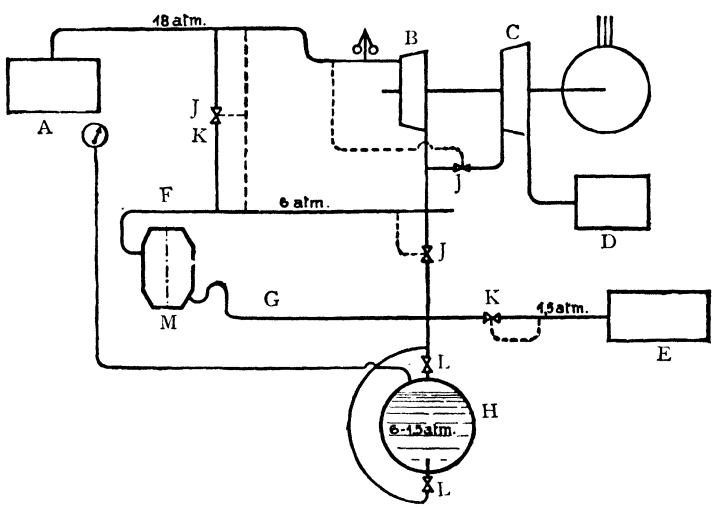


Fig. 7.—Control Diagram of a Cellulose Factory using Ruth's Accumulator (M.A.N., Augsberg, Nürnberg)

A, Boiler. B, High-pressure turbine. C, Low-pressure turbine. D, Condenser. E, Drying, bleaching and heating plant. F, Boiling pipe. G, Steam pipe. H, Accumulator. J, Overflow valve. K, Pressure reducing valve. L, Check valve. M, Boiling plant.

water is thus induced. The steam is removed from the dome through the automatic valve *f*. Accumulators of this character can be used for surplus

steam of high pressure, and for steam withdrawn from an engine before the completion of expansion as described in the last chapter.

Examples of factory systems incorporating accumulators are shown in figs. 6 and 7. The first diagram represents that of a textile factory and the

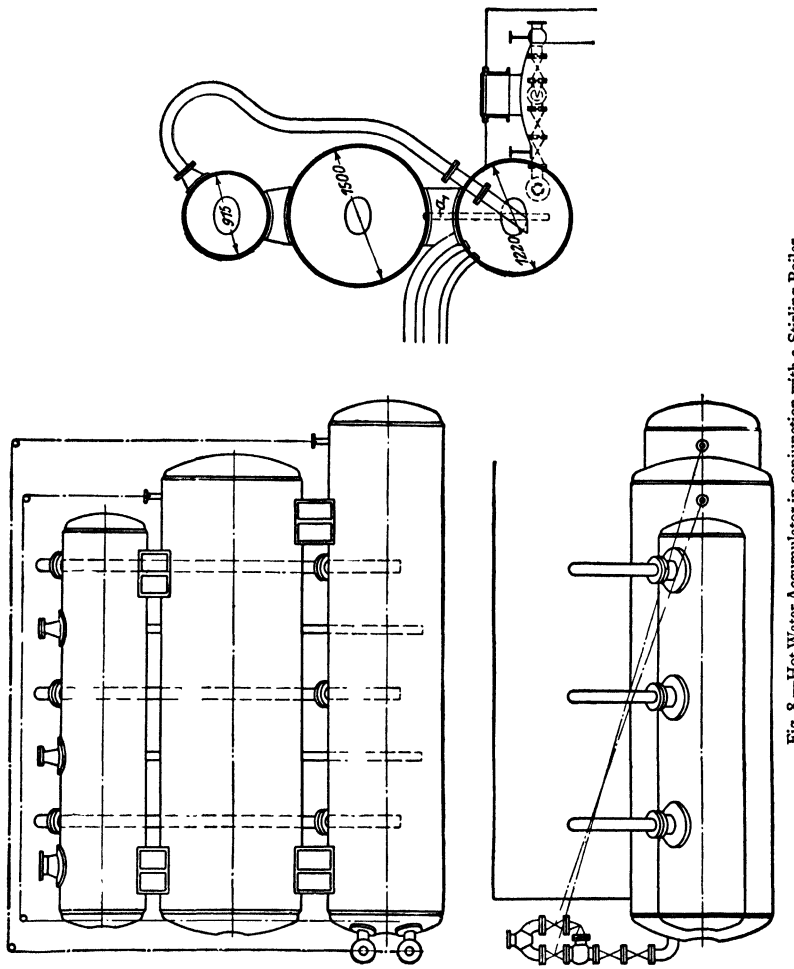


Fig. 8.—Hot Water Accumulator in conjunction with a Stirling Boiler

second the arrangement in a cellulose factory. In all cases, the accumulator is inserted at a point where input is not continuously equal to output, one or both being variable. The accumulator has an equalizing effect, taking the excess of the input during some periods and supplying the deficit of the input at other periods, so that the mean input is equal to the mean output except for small losses in the accumulator.

Many large steam-boilers are now fitted with feed-water accumulators.

This consists of a small boiler partly filled with water and maintained at the same temperature as the water in the main boiler to which it is connected. The normal amount of feed water is supplied through an economizer in which it is preheated by the furnace exhaust gases. When a peak load occurs and there is an excessive withdrawal of steam, an extra supply of feed water at boiler pressure and temperature flows from the accumulator. In this way the depth of water in the boiler can be maintained within very narrow limits even with very fluctuating demands. If the excess amount of feed water was supplied by the normal source when required, its temperature would fall, producing a corresponding fluctuation in temperature and delay in output of the boiler. The arrangement of a constant-pressure accumulator attached to a Stirling boiler is shown in fig. 8. The accumulator is placed over the upper boiler, and above it is a steam collector.

The advantages to be derived from the use of a heat accumulator on an engine are only required where energy is utilized intermittently, and they can only be realized where attention has been given to the careful planning of the whole plant. It is necessary that the power plant should consist of a high-pressure section and a low-pressure section, the accumulator being intermediate between the two. The back pressure of the high-pressure section will be a little greater than the admission pressure of the low-pressure section. The accumulator must be of such dimensions that it can absorb the excess boiler output when energy is not being taken from the high-pressure section, while at the same time it gives out heat at an approximately constant rate to the low-pressure section. By the obvious avoidance of waste, the use of accumulators enables the boilers and plant to be run at a more uniform loading and, if designed for this loading, the maximum efficiency can be thereby maintained. The varying demands can be followed more quickly than with a single large plant run partly at full and partly at reduced power. The amount of manual attention required is also less.

In estimating the size of accumulator required for the regulation of a given varying output, the following approximate values of the properties of saturated steam and of water at the boiling-point are required.

Pressures in Atmospheres (absolute).	Total Heat, B.Th.U. per C. Ft.		Weight, Pounds per Cubic Foot.	
	Steam.	Water at B.P.	Steam.	Water at B.P.
1	42	9,700	0·037	60·0
2	81	12,900	0·071	59·2
3	120	14,000	0·104	58·4
4	156	15,000	0·136	57·8
5	194	15,700	0·168	57·4
7	268	16,800	0·231	56·7
10	378	18,300	0·323	55·6
13	486	19,200	0·415	54·7
16	596	19,900	0·502	54·1
20	734	20,600	0·620	53·2
25	915	21,300	0·765	52·2

Thus, if a steam accumulator reaches a pressure of 16 atmospheres and the pressure is reduced during withdrawal to 13 atmospheres, 110 B.Th.U. per cubic foot of steam are released, which is sufficient to evaporate 0·13 lb. of water forming steam at 13 atmospheres. For the same pressure variation in a water accumulator, 700 B.Th.U. are released, which would cause the evaporation of 0·85 lb. of water. These figures require a small reduction owing to the work done in expansion when the pressures are reduced.

3. CONDENSERS

Various methods are employed for the condensation of steam from power plants. In surface condensers, the steam is passed through tubes and the heat abstracted by cold water flowing on the outside, or the position of steam and water may be reversed. In evaporative condensers, the flow of water is very small so that evaporation takes place. In jet condensers of various types, the water and steam intermingle. In order to maintain the required vacuum, provision for the removal of the condensed steam must be made, and in the case of jet condensers the added water must also be removed. This may be effected by piston or rotary pumps. With high-level or barometric condensers, the water flows away by gravity. The removal of air is usually attained by ejector pumps, although other types of air pumps are also used.

In a perfect condenser of any type, the cooling water finally attains the temperature of the condensed steam. If w is the weight of steam to be condensed in unit time, t its temperature, and L its latent heat at that temperature, while w_1 is the weight of water in unit time, t_1 its initial temperature, and t_2 the final temperature of water and condensed steam,

$$w_1(t_2 - t_1) = w(\mathrm{L} + t - t_2).$$

The weight of water per pound of steam being inversely as its temperature rise, the effect of increasing the flow of water when it is already large is not great. The weight of steam to be condensed is, of course, the difference between that supplied to the engine and that condensed within the engine during the working cycle.

4. SURFACE CONDENSERS

As the condensed steam is kept separate from the cooling water and is free from lime and other salts, it can be returned to the boiler and used repeatedly with advantage. Except for leakage, little air is introduced into the system, the air pumping required to maintain the vacuum being thus small. An example of a surface condenser is shown in fig. 9. The cooling water flows from A through the lower tubes to B, and returns through the upper tubes to the outlet C. The steam, entering at D, flows across the cold tubes and when condensed is withdrawn by the pump attached to E. The efficiency of all surface condensers depends greatly on the cleanliness of the

surfaces, and any deposits of oil, grease, or mud greatly reduce its effectiveness. If clean cooling water is not available, it is necessary to clean the tubes periodically either mechanically or by the circulation of hot cleansing solutions. An alternative method is to allow the warmed water to cool and to use it again and again, in the same manner as the steam water is used repeatedly. The outer surface of the tubes also receives oil carried over by the steam from the engine. With a reciprocating engine, the weight of oil deposited per 1000 lb. of steam is of the order of 0·01 to 0·015 lb., but the thin film thus formed has a large effect on the heat conductivity. A layer of oil 2×10^{-5} in. thick will exert the same resistance to heat flow as a layer of scale 2×10^{-4} in. thick, or as a sheet of iron 10^{-1} in. thick, i.e. the resistance is 5000 times that of iron of the same thickness.

The general temperature of the condenser and hence that of the condensed steam may vary with the atmospheric temperature unless precautions are taken to avoid this effect. As the condensed steam is used for feed water, usually being preheated in the economizer, there would be a corresponding variation in the demands made on the pre-

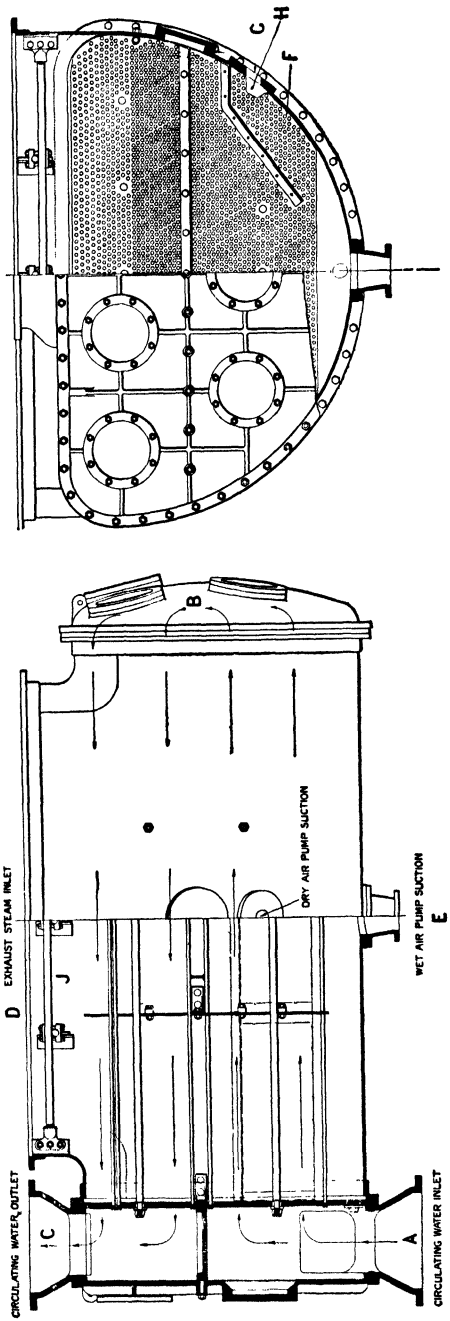


Fig. 9.—Weir Uniflux Condenser

heater for a constant engine performance. The temperature of the condensate can, however, be regulated by means of the supplementary feed water required to make up leakage losses in the system. This additional feed varies between 5 per cent and 10 per cent of the water consumption. In the surface condenser shown in fig. 10, a number of water pipes connected to the chambers *t*, *t'* are separate from those containing the cooling water. When the condensed steam is cooler than normal, it is passed through these tubes and heated a few degrees before passing to the economizer. If, on the other hand, the condensed steam is warm and the supplementary feed water is cold, the latter can be passed through this nest of tubes.

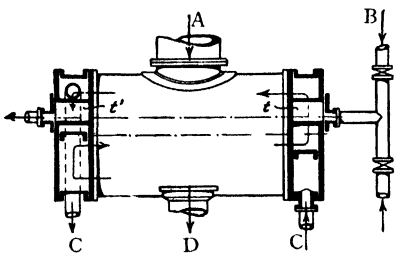


Fig. 10.—Pre-heating of Feed Water in a Surface Condenser

A, Exhaust steam. B, From fresh supply. C, Cooling water. D, Condensate.

5. EVAPORATIVE CONDENSERS

An example of an evaporative condenser is shown in fig. 11. The steam circulates within the pipes, while a small flow of water is maintained over the outside surfaces which are corrugated to increase the area. In condensing the steam, the thin film of water evaporates, and, owing to the high latent heat, the amount of water required is very much smaller than with other types. The amount of surface required is large as the heat conductivity across a surface when the fluid is nearly stationary is low. The condensers are therefore bulky, which, however, is an advantage in cases where there are occasionally heavy overloads.

6. JET CONDENSERS

Jet condensers are of two types. In the first type, the steam is passed into a large chamber into which water is sprayed downwards from several jets. To facilitate the mixing of the steam and water the nozzles are designed to eject the water in a finely atomized condition. At various heights, perforated trays are placed so that the water temporarily collected in them again issues as spray from the perforations. Examples of a vertical and a horizontal design are shown in figs. 12 and 13. The second type of jet condenser is in the form of an injector, and is shown in fig. 14. The stream of cooling water entering at B and issuing from the nozzles E, creates there the vacuum which draws the steam from A into the jet and subsequently condenses it.

In these condensers the pumps have not only to remove the condensed steam and the cooling water, but also large quantities of air which are introduced with the water. The condensate, free from solids in solution, becomes

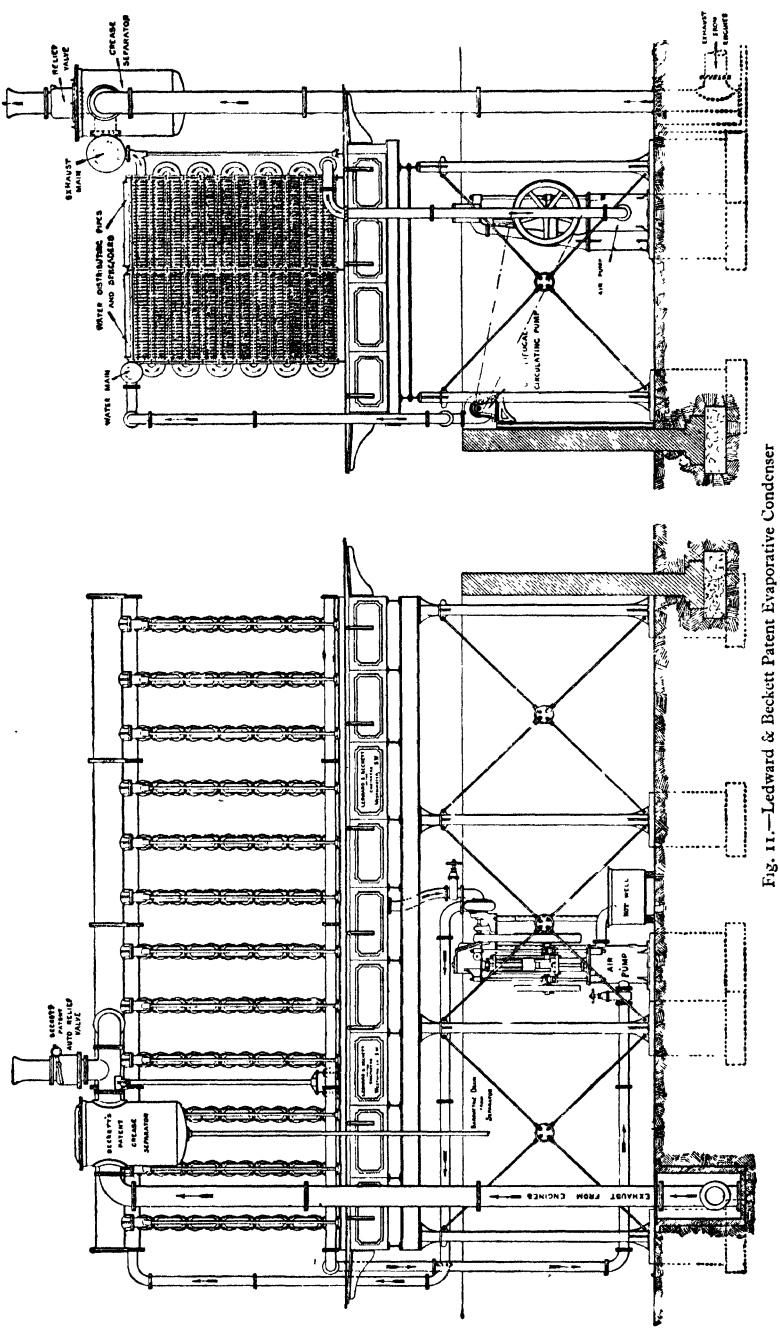


Fig. 11.—Ledward & Beckett Patent Evaporative Condenser

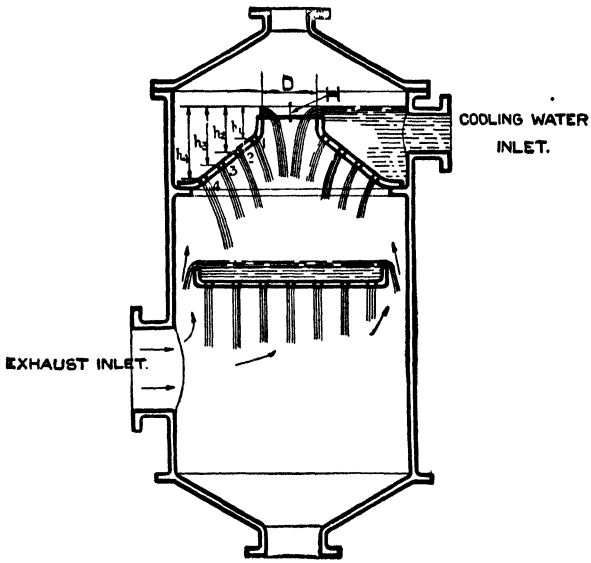


Fig. 12.—Contra-flow Jet Condenser

mixed with the water, and the advantages of again using it in the boiler and engine are lost. The abstraction of the heat from the steam is, however, much more efficient with direct contact and without the intervention of a solid and usually greasy separating wall. The size of jet condensers can thus be very much less than that of surface condensers dealing with the same quantity of steam.

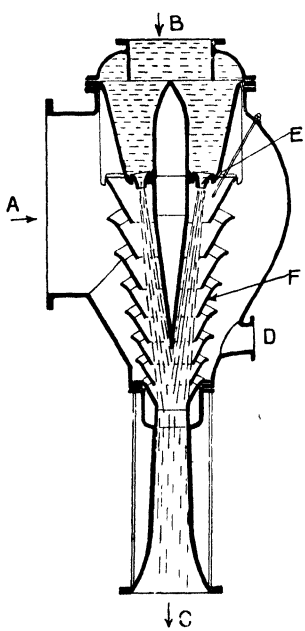


Fig. 14.—Körting Multi-Jet Ejector Condenser

The arrangement of a plant with a jet condenser and a gravimetric removal of the water, forming a high-level or barometric condenser, is illustrated in fig. 15. The condenser is similar in character to that shown in fig. 12, but is situated above a column of water equal to the height of a water barometer. The cooling water is pumped to the top and enters by the pipe A, while the steam from the turbine C enters by the pipe B. If the steam is efficiently condensed, it will flow together with the cooling water down the pipe D, in which the water level at barometric height remains constant, while the chamber of the condenser is similar to the Torricellian vacuum.

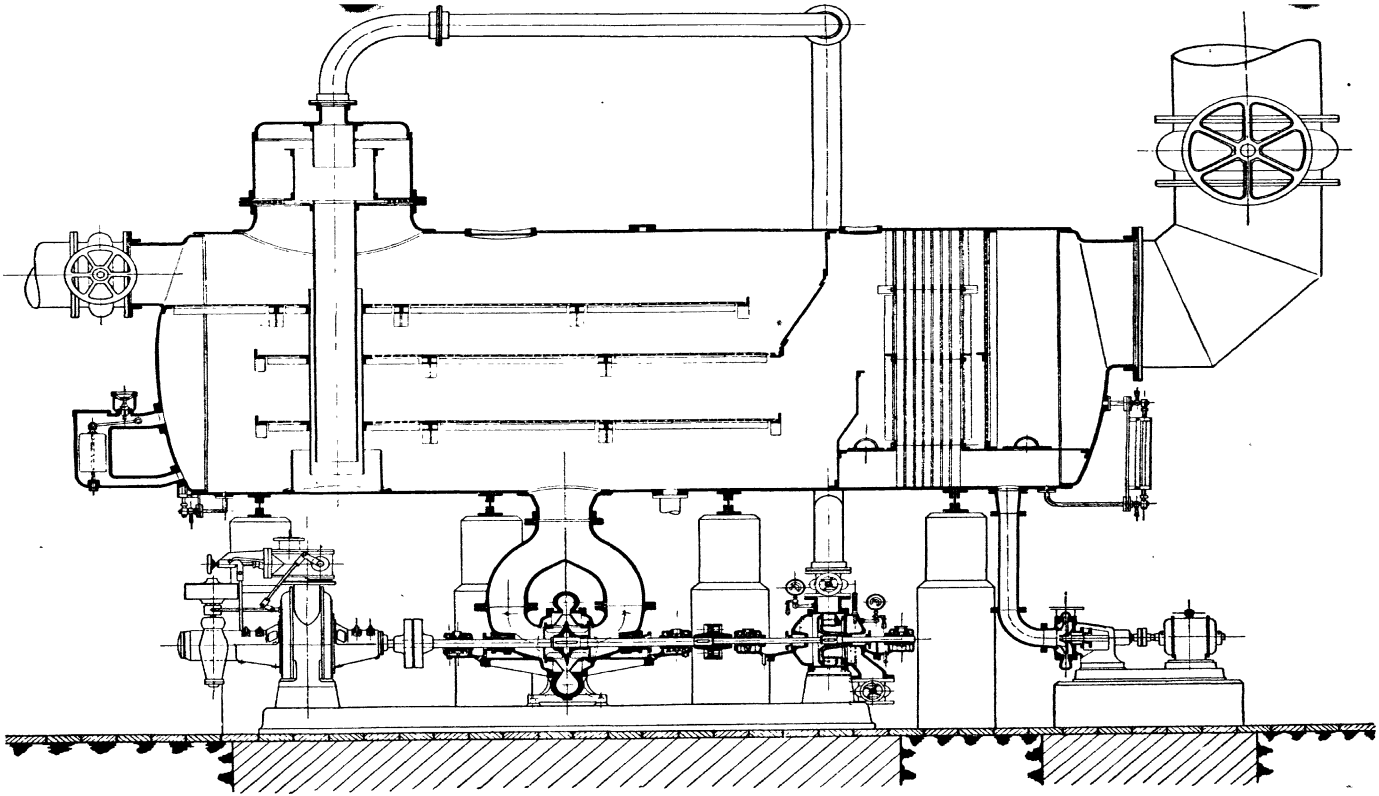


Fig. 13.—Balcke Horizontal Mixed Condenser Plant

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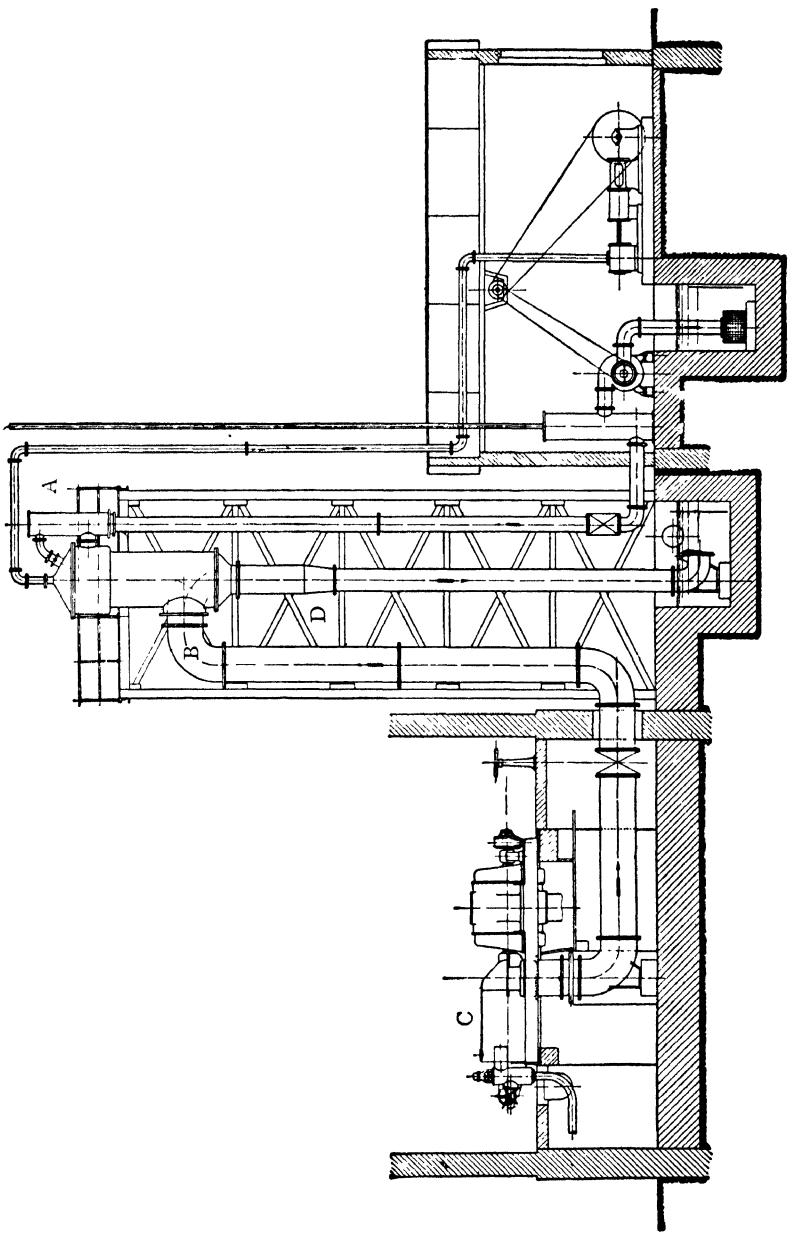
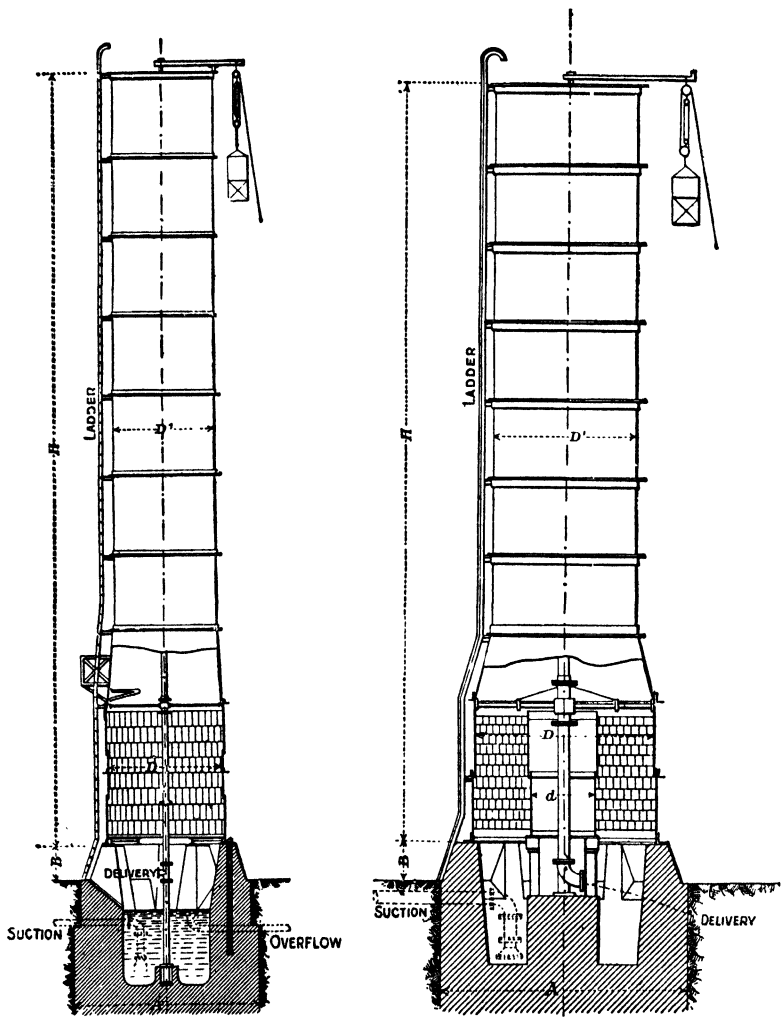


Fig. 15.—Steam Turbine Plant with Barometric Condenser

7. WATER COOLING

The cooling of water from condensers and from the jackets of internal-combustion engines is attained by imparting its heat to air. A simple method



Section of Natural Draught Cooling Tower. Less than 18 ft. Diameter.

Section of Natural Draught Cooling Tower. More than 18 ft. Diameter.

Fig. 16

is to run the water into a pond or large tank where the cooling takes place at the surface partly by transmission to the air and partly by evaporation of some of the water. This arrangement, however, is slow, and requires a

large reserve of water. With the water-cooled engines of motor-cars and aeroplanes, the amount of water must be reduced to a minimum and there must be no loss by evaporation. Hence a closed circuit is used and the water is cooled by passing it through very narrow tubes or passages round which a stream of air is induced by the speed of the machine, assisted sometimes by a fan. With small engines the water circulation is due to convection, but in larger engines a water pump is always used. The scrubbing action due to the velocity of both air and water enables a rapid cooling to be obtained. The amount of heat to be dissipated may be assumed to be approximately equal to the horse-power of the engine. If 1 h.p. is taken as 2550 B.Th.U.

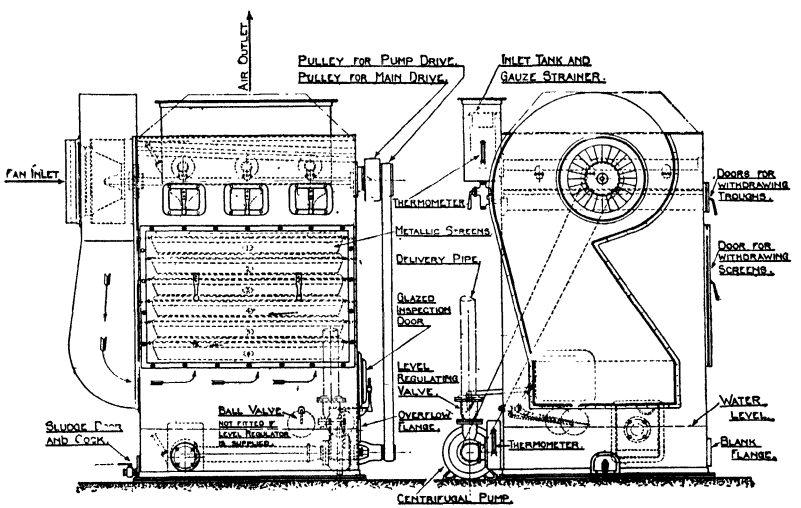


Fig. 17.—Heenan Water Cooler

per hour and there is a rise of water temperature in the engine jacket of 30° F. and an equal fall of 30° F. in the radiator, the flow of water required in pounds per hour per horse-power is 85. As a gallon of hot water at about 170° F. weighs a little less than 10 lb., the flow required is 8·5 gall. per hour or 0·14 gall. per minute for each horse-power of the engine. Radiators of good design will dissipate from 20 to 30 h.p. per square foot of frontal area at an air speed of 30 miles per hour. When the speed is quadrupled the horse-power dissipated is increased three times.

Water-cooling towers are used for the cooling of water from power plants where the pond method is too slow or space is not available. The towers are either of the open or the chimney type. The former depend upon the wind for the flow of air, while in the latter there is a forced or natural draught upwards through the cooler. The water is directed upwards in a fine spray, falling back into a tank which covers the floor area. The flow of both air and water is broken up by bars or other obstructions within the cooler. The cooling is assisted by evaporation of some of the water which, being

carried away by the air, is lost. This loss is greater with the open type than with the chimney type. Under average conditions the cooling due to evaporation in this country may represent 80 per cent of the total cooling. If the air is initially at 60° F. and is half saturated, each cubic foot of air will contain 0·0004 lb. of water vapour and will weigh 0·0765 lb. If the air leaves at 80° F. and is fully saturated, it will occupy a slightly greater volume and will contain much more water, viz. 0·0016 lb. To evaporate the additional water requires 0·0012 × 1050 = 1·26 B.Th.U. The air by its increment of temperature of 20° F. has received 0·0765 × 0·242 × 20 = 0·37 B.Th.U. The total amount of heat taken from the bulk of the water is 1·26 + 0·37 = 1·63 B.Th.U., and the proportion of this due to the evaporative effect is 1·26/1·63 or 0·77. Assuming that the bulk of the water has been cooled 30° F., the weight so cooled by this amount of air is 1·63/30 = 0·054 lb. The proportion of water lost by evaporation is given by 0·0012/0·055 = 0·022 or 2·2 per cent.

A modern design of water cooler is shown in fig. 16.

A water-cooler of medium size used in connexion with stationary internal-combustion engines, hydraulic dynamometers, condensers, and refrigerating plant is shown in fig. 17.

8. WATER DISTILLATION

Where clean water is not available it is necessary to purify the feed water of power plants in order to avoid objectionable deposits of lime, grease, and mud within the boilers and engines. The removal of salts in solution by chemical means has been referred to in Chapter V. Solids in suspension can be removed by the use of settling tanks. Distillation at ordinary atmospheric or low pressures will remove practically all foreign matter. Where the exhaust steam of a plant is condensed and recovered, there is yet some loss by leakage and the additional feed water required should be purified.

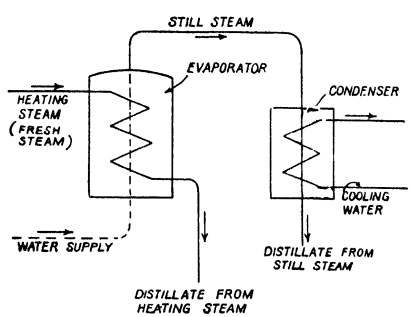


Fig. 18.—Distillation Process

The evaporation of the water is most conveniently performed by the use of steam at a pressure a little higher than atmospheric. The elements of the system are shown in fig. 18. A more efficient arrangement is shown in fig. 19, in which the feed water is first used as cooling water, and after being thus heated nearly to boiling-point, it passes into one or more evaporators. A further improvement is that given in fig. 20, in which the feed water, after being used as a condenser, is added to the live steam and the condensate from both is collected together. The heat distribution of such a system is shown in fig. 21, and the actual arrangement of a plant in fig. 22.

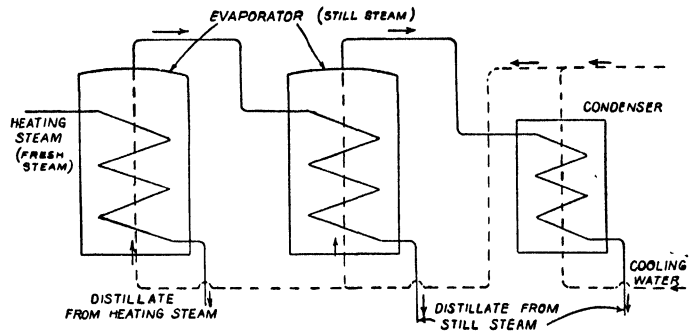


Fig. 19.—Multiple Stage Evaporator

A method of a different character is operated by the plant illustrated in fig. 23, which has a thermal efficiency of 50 to 60 per cent. Part of the exhaust steam of the engine or turbine flows directly into the condenser and part into the evaporator. The pressure in the evaporator is less than in the main condenser, this effect being maintained by the use of an auxiliary condenser in which the water is colder than in the main condenser. A considerable flow can thus be obtained through the evaporator to the auxiliary condenser, and in its passage heat is abstracted for the vaporization of the supplementary feed water. In the auxiliary condenser both the exhaust steam and the additional water vapour are condensed. A plant of this character attached to an 8000 kw.

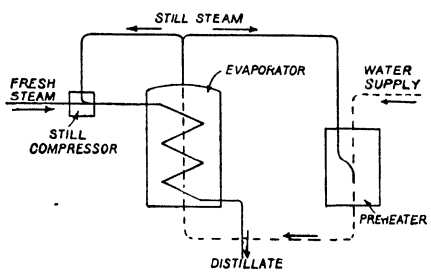


Fig. 20.—Thermo Compression Process

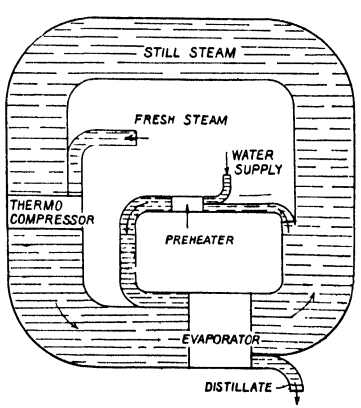


Fig. 21.—Heat Diagram

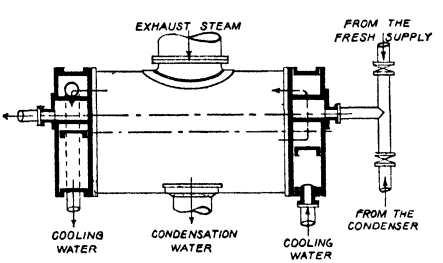


Fig. 22.—Pre-heating of Feed-water in a Surface Condenser of a Steam Turbine

turbine has a main condenser with a cooling area of 14,000 sq. ft., an auxiliary condenser of 2250 sq. ft., and an evaporator with 1150 sq. ft.

With an inlet water temperature of 82° F., an outlet temperature of 97° F., and a vacuum of 93 per cent in the main condenser, a supplementary water supply of 8 per cent can be distilled.

The heat transmission from the steam to the water depends principally on the surface conditions. Many tests on the effect of the steam conditions are contradictory. Air forms a cushion between walls and steam, and a very

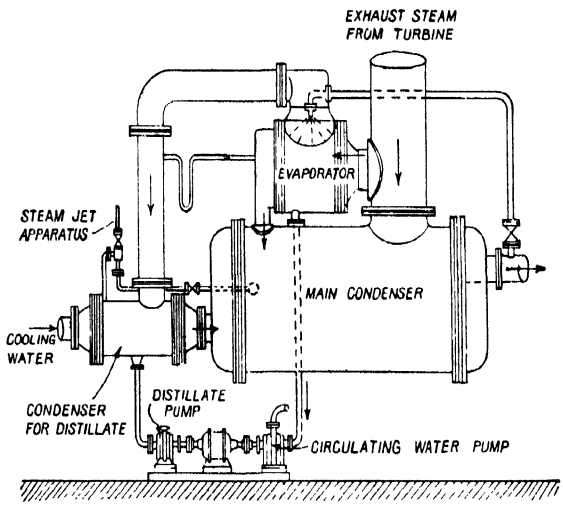


Fig. 23.—Distillation Plant

thin film may reduce the transmission to one-half or one-quarter. According to Josse the velocity of the steam exerts little effect but the velocity of the water was very important. The heat transfer with air-free steam and water velocities of different amounts is given by him as follows:

Water Velocity.			Heat Transfer.			
1 ft. per second	....	290	B.Th.U.	per	square	foot per hour
2 "	....	480	"	"	"	
3 "	....	640	"	"	"	
5 "	....	850	"	"	"	
7 "	....	1020	"	"	"	

CHAPTER VIII

Heat Balance and Energy Measurements

1. HEAT BALANCE

In order to bring a plant to the highest possible degree of efficiency, it is necessary to follow the heat energy put into the plant throughout its course and to know what proportion of it is used or lost at each stage. By a com-

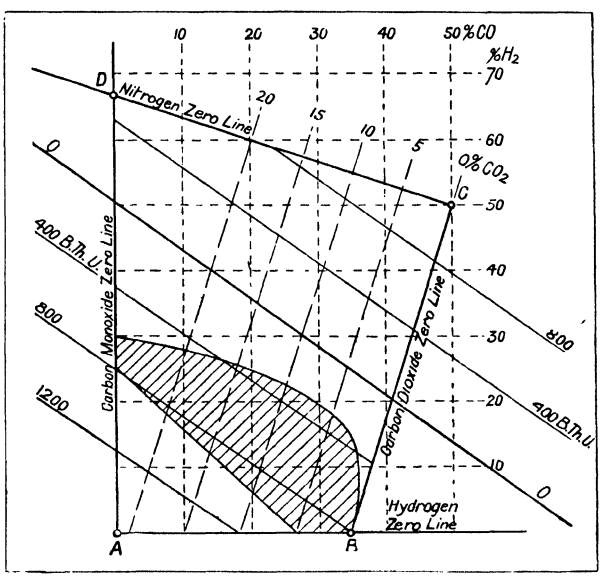


Fig. 1.—Heat of Formation Diagram

parison with the results which may theoretically be obtained or with the best experimental results in similar plants, avoidable and unavoidable losses may be distinguished and attempts to reduce the former initiated. The sum total of the different proportions of the energy as estimated at any stage must, of course, balance the initial energy put in.

The heat content of any solid, liquid, or gas can be found from a knowledge of its mass, temperature, and specific heat. The work done in any process can be obtained from the area of a pressure-volume diagram or a tem-

perature-entropy diagram as described in Chapter I, or by the variation in total heat. In the consideration of a whole plant, the energy is initially in the form of calorific value of the fuel, whose value is known or can be found experimentally. The quantity of fuel, water, or other substances introduced can also be obtained without difficulty, an exception being the estimation of the combustion air. Methods of estimating this, however, by means of exhaust gas analysis have been indicated in Chapter I. In some cases it is difficult or the requisite means are not available to measure directly the apportionment of heat at an intermediate point in a process and similar indirect means are resorted to. In the case of a plant heated by a gas producer it is necessary to separate the efficiency of the producer from the efficiency of the plant. The heat content of the producer gas must therefore be known, including the calorific value and sensible heat. These may be measured directly or they may be deduced from a gas analysis, together with charts of previous test experiments. Such a set is shown in figs. 1, 2, and 3, which represent the results obtained from the combustion of carbon under various conditions of temperature and with or without water and steam. Within the area ABCD, the ordinate of any point represents the percentage of H_2, the abscissa that of CO, and the distance from CB gives the percentage of CO_2. Gas unaccounted for is N_2. Thus the point A with no CO or H_2, 21 per cent of CO_2, and 79 per cent of N_2 gives the analysis of pure flue gas. The point C with 50 per cent CO, 50 per cent H_2, and no CO_2 or N_2 represents a perfect water gas while point B represents perfect producer gas. The inclined lines in fig. 1 give the values of the heat of formation of the gas. It will be seen that for gas analyses represented by points in the upper part of the area, the heat of formation is negative, i.e. such gas mixtures could only be obtained from carbon, air, and water by the employment of external heat or energy. The gases from actual generators are usually given by points within the shaded area, the positive heat of formation per pound of carbon being the heat released in the generator. In fig. 2 the thermal

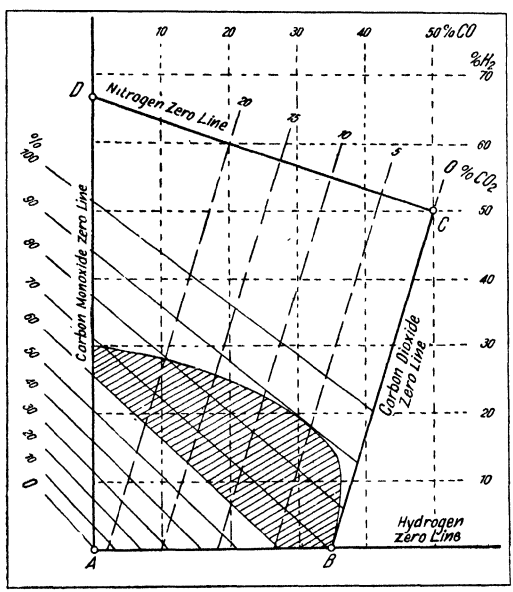


Fig. 2.—Thermal Efficiency Diagram

efficiency of the generator and in fig. 3 the calorific value of the gas is given. With such diagrams, prepared for the fuel used in the gas generator, the work entailed in the continual measurement of its performance may be reduced in cases where the performance of the remainder of the plant is primarily required. The data can be made to include the sensible heat as well as the calorific value, if the arrangement of the plant is such that both are utilized.

As a very simple example of the kind of balance sheet which it is desirable to construct, the case of combustion in a furnace may be considered. Let it be assumed that coal of calorific value 11,500 B.Th.U. per pound is burnt with air in excess of that required for perfect combustion amounting to 50 per cent. The maximum temperature in the furnace, which is in the neighbourhood of the fire bridge, is 2900° F. Heat is given up to the body being heated, and some is lost by radiation, so that at the end of the hearth the temperature is 2200° F. This temperature should be neither greater nor less than that required by the body being heated. In the passage from hearth to chimney there is further loss and also in the chimney, the final temperature of the flue gases being (say) 1500° F. If the combustion is complete at the attainment of the highest temperature, the heat content is proportional to the temperature and mean specific heat at any point. A table of the following character then gives the principal data of interest.

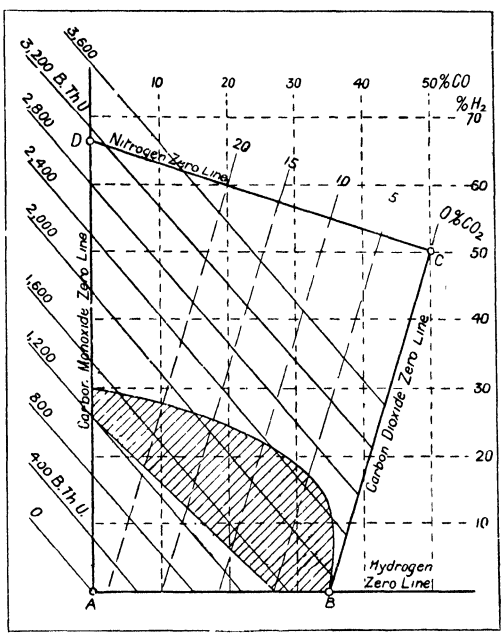


Fig. 3.—Calorific Value Diagram

Calorific value per pound of coal ..	11,500	B.Th.U.
Heat given to body	2,700	B.Th.U.
„ lost by radiation, &c., in hearth	700	„
„ lost in passages and flue ..	2,900	„
„ remaining in flue gases ..	5,200	„
Total	11,500	B.Th.U.

The temperature curve is shown in fig. 4. The curve giving the heat content,

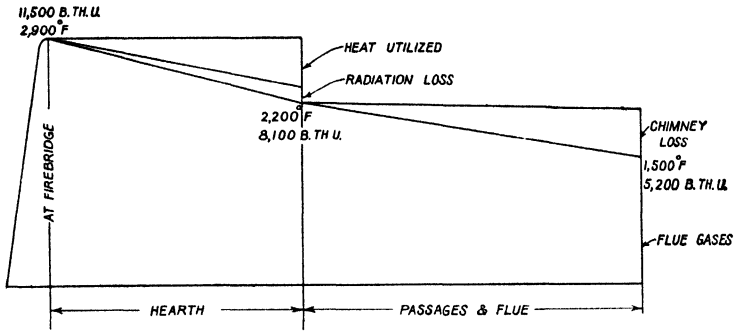


Fig. 4.—Combustion with 50 per cent Excess Air

although similar, is not exactly of the same form, owing to the increase of specific heat with temperature. The relative figures for the heat content are

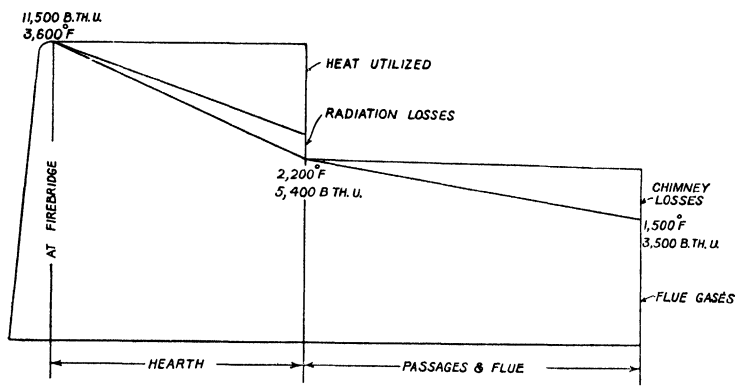


Fig. 5.—Combustion with no Excess Air

inserted on the temperature curve. This curve may now be compared with that which might have been obtained if the minimum quantity of air for complete combustion had been used, as given in fig. 5. The effect of increasing or decreasing the excess air can be found quantitatively, and the advantages of a small or no excess with its high resultant temperatures compared with those of the large excess with its smaller heat losses.

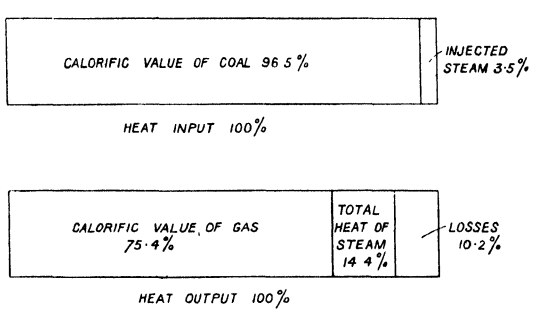


Fig. 6.—Heat Balance of Gas Generator

Another simple method of illustrating the energy distribution before and after a process is shown by fig. 6, in which an area representing the total energy introduced is divided into parts according to the proportions in which it is distributed. The example is that of a gas generator described in a former chapter which is surrounded by a water jacket for the production of steam from the sensible heat of the actions. The quantities concerned are given by the following test results.

HEAT INTRODUCED PER POUND OF COAL

Calorific value of coal ..	14,000 B.Th.U.	= 96·5 per cent
Injected steam	500 ,,	= 3·5 ,,
Total	14,500 B.Th.U.	= 100 per cent

HEAT OUTPUT PER POUND OF COAL

Calorific value of gas ..	10,930 B.Th.U.	= 75·4 per cent
Total heat of steam ..	2,090 ,,	= 14·4 ,,
Losses	1,480 ,,	= 10·2 ,,
	14,500 B.Th.U.	= 100 per cent

The injected steam being part of the steam production, the efficiency of the generator is really represented by the ratio of the calorific value of the gas, plus the heat of the unused steam, to the net heat introduced, which is the calorific value of the coal. Thus

$$\text{Efficiency} = \frac{10{,}930 + (2090 - 500)}{14{,}000} 100$$

$$= 89{\cdot}5 \text{ per cent.}$$

A more picturesque method of showing the course of the energy is illustrated in fig. 7, which gives the distribution in a gas generator supplied with steam from a separate boiler. Of the total energy supply of 100 per cent, 7·5 per cent is supplied to the boiler, in which 3 per cent is lost and the 4·5 per cent is contained in the steam produced. In the passage to the grate a further 0·5 per cent is lost owing to condensation, so that the injected steam contains 4 per cent of the energy supply. Radiation and cooling water are responsible for losses amounting to 6 per cent, and the residue of ashes contains 3·5 per cent, the useful heat in the generator

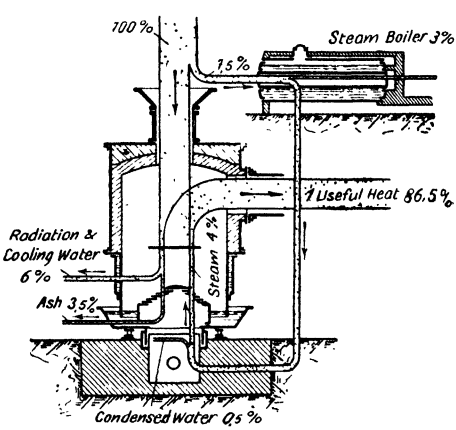


Fig. 7.—Heat Balance of Generator

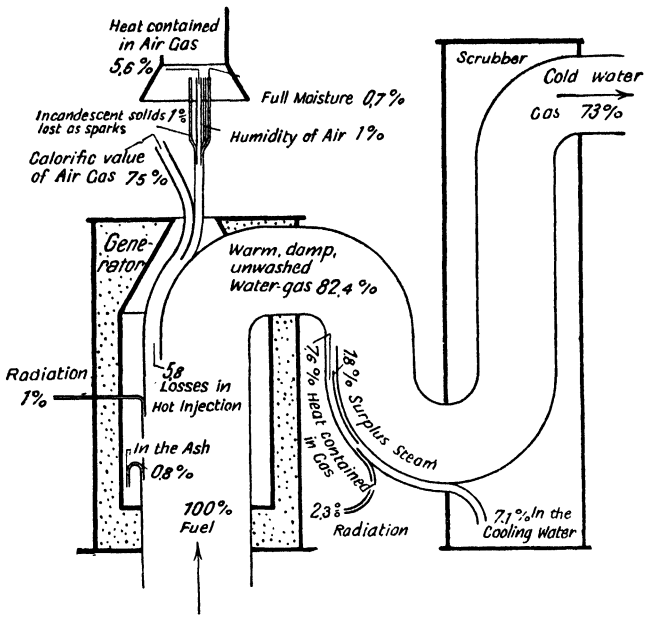


Fig. 8.—Heat Balance of the Dellwick-Fleischer Process

gas, which includes both calorific value and sensible heat, being 86·5 per cent. The operation of the Dellwick-Fleischer process for the production

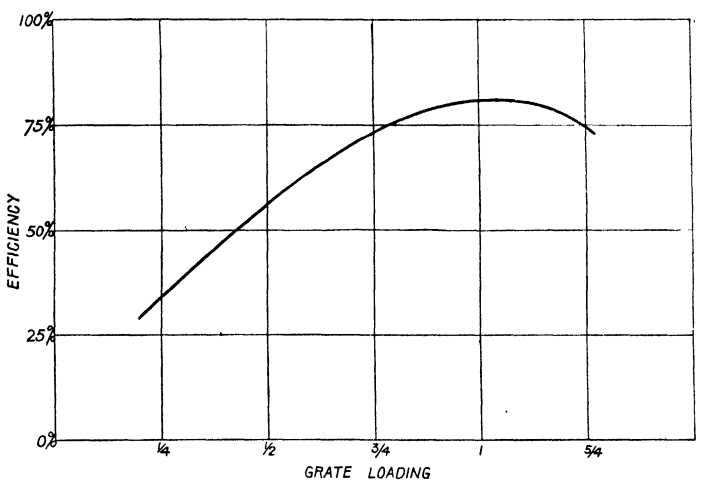


Fig. 9.—Furnace Efficiency and Grate Loading

of water gas is given in fig. 8. The coke is first heated by high-pressure air in quantity approximating to that for perfect combustion and air-gas

of small calorific value produced. Steam and air are afterwards injected and water-gas obtained as described in Chapter IV. The heat distribution and losses for both air-gas and water-gas periods are given in the diagram.

Diagrams of the character described above can only represent the input and output of a plant under some particular conditions or under some average conditions of operation. Where a plant is operated under conditions which vary, it is necessary to represent each component of the output

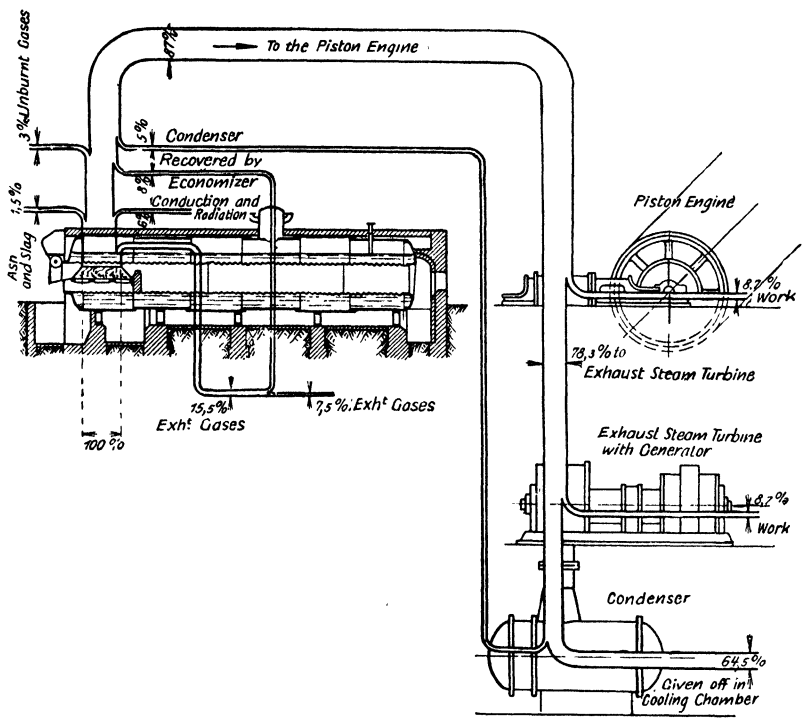


Fig. 10.—Heat Diagram of a Power Plant using Steam

by a curve which shows the variation of its value with the variation of the conditions. The effect of varying the depth of fuel on the grate of a boiler furnace is shown in fig. 9, in which the efficiency is plotted against the loading. The loading which gives the maximum efficiency is taken as the normal loading. With too heavy a loading, the rate of combustion is reduced and the losses increase owing to the increase in time, while with too light a loading, the excess of air which is permitted to enter also increases the losses by the increase in exhaust heat. The importance of accurate stoking is thus shown graphically in the figure. If the variation in conditions consists of one or more single changes, such as the substitution of one kind of coal for another, the results obtained can be represented graphically by a number of isolated points instead of a continuous curve.

The quantities concerning which information is desired may be classified generally in all plants under the headings *input*, *useful output*, *and losses*, but their individual character naturally depends upon that of the plant. Some of the measurements required in the testing of plants for combustion and gasification have already been indicated in the preceding examples.

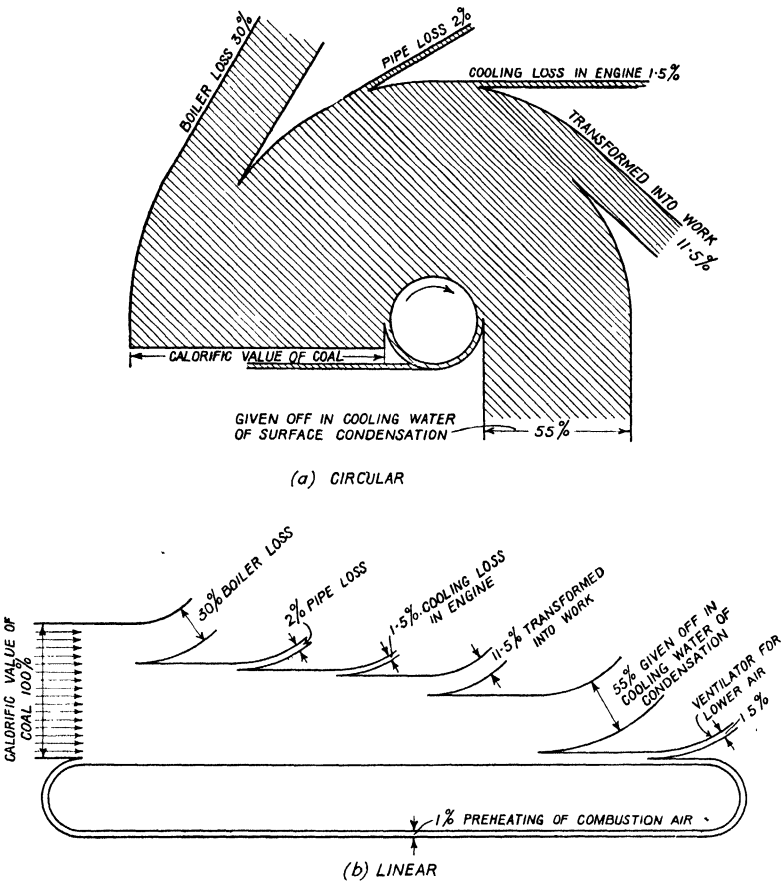


Fig. 11.—Heat Diagram

They consist essentially of the measurement of calorific values and sensible or total heat. In a complete power plant there is in addition to similar measurements that of work done. An example of the flow of energy in such a plant is given in fig. 10. The energy is supplied in the form of the calorific value of coal and is used for the production of high-pressure steam whose total heat represents 87 per cent of that calorific value. The sensible heat of the exhaust gases is 15·5 per cent, but as 8 per cent is recovered in the economizer, the loss due to this cause is only 7·5 per cent. The calorific

value of gases incompletely burnt represents a loss of 3 per cent. The losses due to conduction and radiation are 6 per cent, and those due to ash and slag 1·5 per cent. Condensed steam returned from the condensers adds 5 per cent to the heat energy. The high-pressure steam used in a back-pressure reciprocating steam-engine with an efficiency of 10 per cent gives an output in mechanical work of 8·7 per cent of the original heat content of the coal, the remainder of its energy, except for small heat losses, being still contained in the exhaust steam. A further 8·7 per cent of mechanical work can be obtained by using the exhaust steam to drive a low-pressure condensing steam-turbine, the heat in the exhaust steam being reduced to 69·5 per

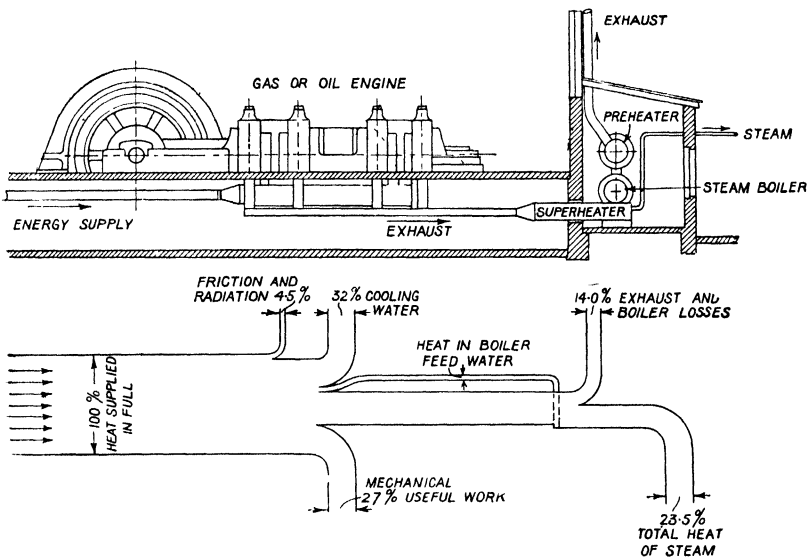


Fig. 12.—Heat Diagram of an Oil or Gas Engine

cent of that of the coal. In the cooling water of the condenser 64·5 per cent is abstracted, leaving 5 per cent in the condensed steam which is returned to the boiler. The energy transformation might be still further followed to the water cooler, where the heat content of the water is transferred to the air. Information concerning a plant of a similar character with a single condensing engine is given in a less elaborate manner in diagrams of the form shown in fig. 11. The energy distribution in a gas-engine plant is given in fig. 12. The exhaust gas of the engine is used for the production of superheated steam. In these diagrams the importance of utilizing the exhaust heat is revealed in a forcible manner by a comparison of the amount of energy obtained for the primary purpose of the plant with the amount obtained from the exhaust. In the gas-engine the useful work forms 27 per cent of the calorific value of the gas, while a further 23·5 per cent is abstracted from the exhaust as total heat of the steam produced.

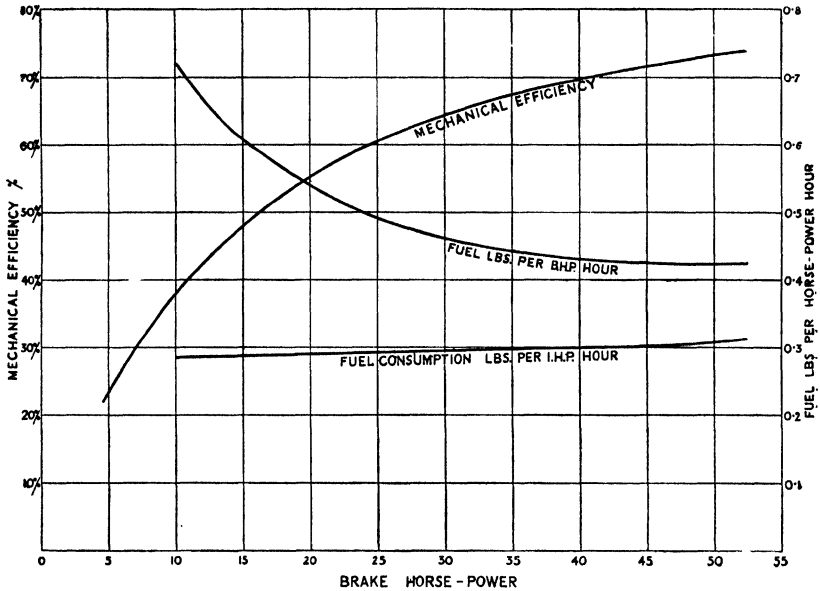


Fig. 13.—Curve showing Mechanical Efficiency and Fuel Consumption on a Diesel Engine

Smelting furnaces, gasworks, and many other industrial plants are usually run under constant conditions. When the energy transformations have been

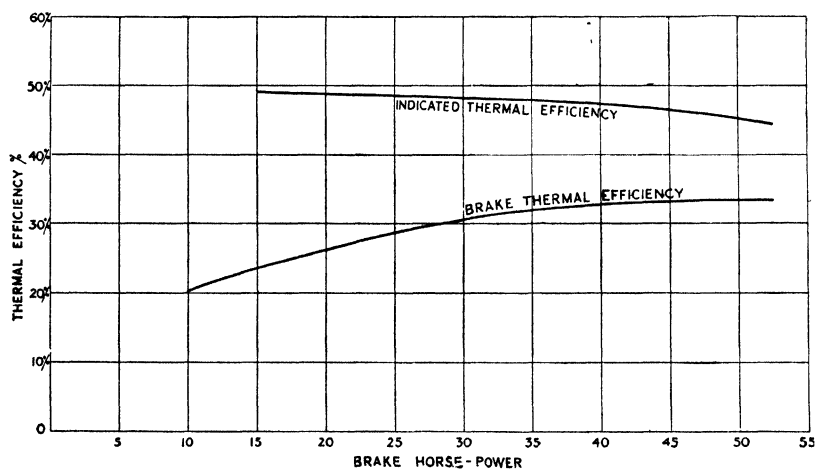


Fig. 14.—Curve showing Indicated and Brake Thermal Efficiencies at Various Loads

investigated, the optimum conditions found, and avoidable losses eliminated, it is afterwards necessary only to maintain those best conditions. The determination of the most efficient grate loading in a furnace has already been

mentioned, while the determination of the most favourable air blast is another point of which knowledge is required. In the case of power plants, however, there may be a continual changing in the conditions of running, and a much more comprehensive knowledge is required of the resultant energy changes. Engines used for driving electric generators, machinery, and similar work are usually run at a constant speed, and variation in output according to the demands made upon them is attained by throttling of the steam, fuel, or air supply. Engines used on road vehicles or aeroplanes experience an automatic change of speed at full throttle when the work required of them varies,

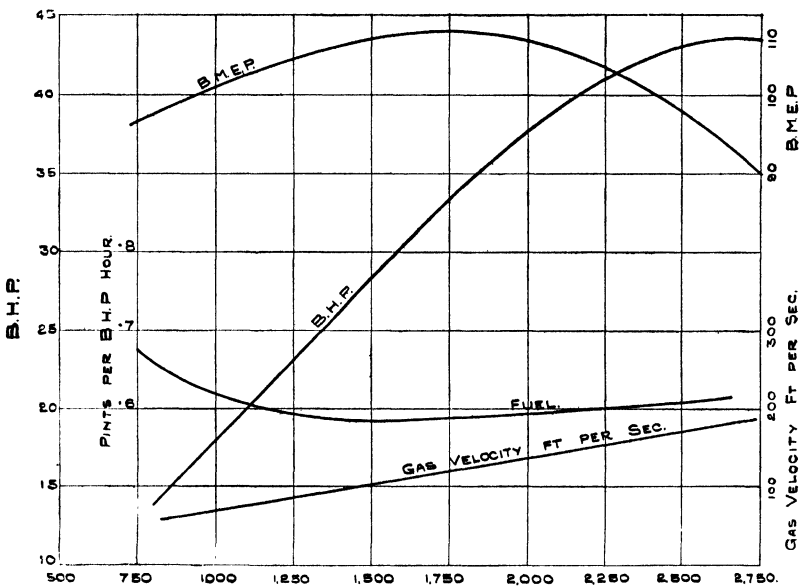


Fig. 15.—14 h p. Vauxhall Engine. Gas Velocity, Brake Horse-power, Brake Mean Effective Pressure, and Fuel Consumption Curves

while the output may also be regulated by the throttle. Fig. 13 shows the information required and the results obtained on a Diesel engine running at a constant speed, but with an output varying from 10 to 50 b.h.p. The curve reveals that while the indicated work obtained per pound of fuel is nearly the same under all conditions, the proportionate loss due to friction is so much greater under small load, that the fuel consumption per brake horse-power hour at 10 b.h.p. is nearly twice that at 50 b.h.p. The corresponding brake and indicated thermal efficiencies are given in fig. 14. The performance of a motor-car engine at full throttle but at varying speed is given by the curves shown in fig. 15. Under these conditions there is little change in the fuel consumption per brake horse-power hour, for the reduced power occurs at lower speed where the losses are also reduced. For the full comprehension of the heat balance under all conditions of running, many similar curves, such as those of the exhaust heat, cooling water heat, friction and pumping

losses, are required for internal-combustion engines and corresponding results for steam-engines.

2. ENERGY MEASUREMENTS

In the construction of the heat balance sheet for a commercial plant and in the measurement of the various physical quantities involved, the accuracy of the determinations is necessarily much less than can be attained in experiments on a small scale in a laboratory. In the majority of cases the larger quantities can only be determined with an accuracy of 1 or 2 per cent, and it is therefore useless to employ refined methods for other quantities even if they are available.

3. SOLID FUEL SUPPLY

The weight of solid required in the test of a power plant is usually that of the fuel. This can be obtained accurately by simple weighing of each truck-load and this method is adopted in small-scale plant or in tests of short duration. In prolonged tests, machines which measure and automatically record the weight or volume used are more convenient. One such type for measuring the volume of coal fed to a boiler is shown diagrammatically in fig. 16. The coal is fed from a hopper on to a short travelling-plate conveyor, the quantity being regulated by an adjustable plate. Resting on the upper surface of the layer of coal is the lower edge of an inclined plate whose inclination and position are hence determined by the depth of the coal. The record of the meter is controlled by the speed of the conveyor and by the position of the inclined plate, and gives the volume of coal being used in the time of experiment. It is necessary to know the average density of the coal in order to find the weight. Machines are also constructed for recording the weight directly instead of the volume. The coal supply is fed into a box which forms part of a weighing machine. When the weight reaches a certain fixed amount, which is determined by the balancing weights, the movement of the box automatically cuts off the supply and causes the box to tip up and to be emptied to the boiler feed. When empty, the box returns to the upright

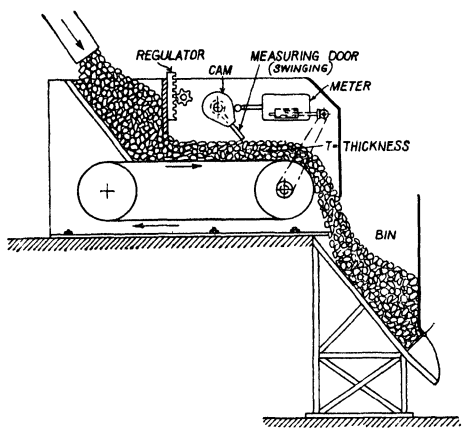


Fig. 16.—The Lea " Cubimeter "

position and the supply is renewed. The number of times the box is inverted is recorded and gives the total weight used.

4. WATER AND LIQUID FUEL SUPPLY

There is a very large number of ways of measuring the supply of a liquid, the methods varying according to the amount flowing and according to the velocity. The meters illustrated in figs. 17 and 18 shows two types of water-meter based upon the same method, and they are suitable for flows of large amount. In the Deacon water-meter a valve rests upon a conical seating when there is no flow, and when there is a current of water the valve is raised and the water flows through the annular space between valve and hollow cone. The controlling force is the weight of the valve, which is floated

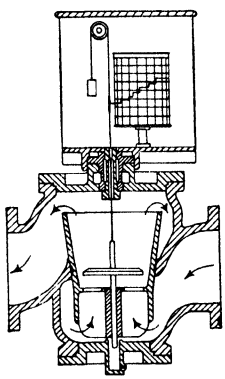


Fig. 17.—Deacon Water Meter

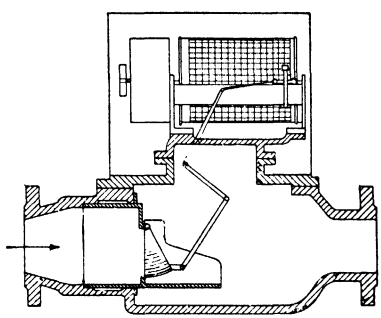


Fig. 18.—Kent Water Meter

by the pressure required to force the water through the annulus. It follows that this pressure must be the same at all flows, and that when the flow is increased the valve rises until the increase of the annular area counteracts the effect of the increase of flow upon the frictional resistance. The rise and fall of the valve is recorded on a drum revolving at a constant rate. The ordinates of the curve thus traced give the flow at any instant and the area gives the total flow during the time of test. In the Kent water-meter the valve takes the form of a door placed vertically and hinged along its upper edge. The flow of the water opens the door until the turning moment of the frictional resistance of the flow between the edges of the door and the containing piping is equal to the restoring moment due to the weight of the door. Continuous records of the position of the door are obtained by suitable links and a drum recorder.

Instruments based upon the same principle are used for measuring supplies of liquid fuel, but as the quantities involved are small, the instruments are necessarily of finer construction. The most common form of petrol flow-meter used in connexion with aero-engines is of the type illustrated by fig. 19. The vane AB takes the place of the valve and door of the previous

examples, and the controlling force is the delicate spring S. The force on the vane due to the flow causes it to deflect until the tension of the spring is equal and opposite to the force. As the deflection becomes greater, the passage C through which the petrol flows changes in size, and by a suitable shape of tube it is possible to obtain an even scale connecting the amount of flow and the deflection of BC. Another design of flow-meter for fuel consumption is the Cowey flow-meter, which is very similar in action to the Deacon water-meter. It is illustrated in fig. 20. The fuel passes up the vertical glass tube B, along the axis of which is a conical pin A. A small cylindrical float C is threaded on the pin, and it takes up such a position in the tube that the pressure required to force the liquid through the annular space between float and pin is equal to the weight of the float. Hence the greater the flow, the higher the position of C. Instruments of this type having gravity control must be upright, but they are free from possible inconstancy due to variation in delicate spring control. The float should be constructed of a thin piece of metal tubing with a thin perforated disc at the top as shown at *b*. The frictional resistance experienced by a liquid or gas in passing through a sharp-edged orifice in a thin plate is found to be practically independent of viscosity, and with a float of this form the readings are very nearly independent of temperature. If the float is constructed as shown at *a*, the annular space D takes the form of a short tube and there is a considerable difference in the frictional resistance at different temperatures.

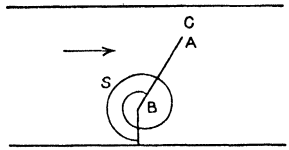


Fig. 19.—Principle of Fuel Flowmeter for Internal-combustion Engines

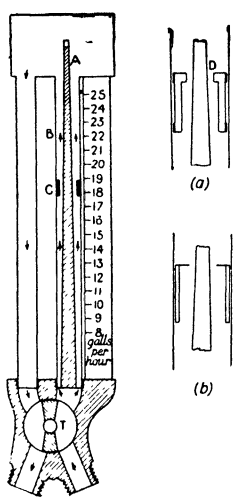


Fig. 20.—Cowey Petrol Flow-meter

Another method used for the measurement of large flows of water is that of the weir. The water is allowed to flow into an open vessel and out again by an orifice or notch in one side. The water takes up such a level that the head at the orifice or notch is sufficient to carry off the supply, and this level is recorded by a float which operates the recording mechanism as it rises and falls. For moderate flows, instruments of the drum type in which the flow of water causes the rotation of a number of vanes forming a paddle are often used. The number of moving parts in instruments for this purpose, and hence the possibilities of erratic behaviour, can be reduced by basing them on the simple measurement of the frictional resistance of a fixed passage, or by using the Venturi or Pitot methods. In all these cases, however, the primary measurement is proportional to the square of the velocity, and

unless there is a rectifying arrangement in the recorder the sensibility at low velocities is much less than at high velocities.

5. STEAM, AIR, AND GAS FLOW

The most accurate method of measuring a flow of steam is to condense it and weigh it as water, but it is seldom that this method is practicable. For large steam and gas flows, instruments similar in principle to the Kent and Deacon water-meters are used. For small flows of gas, instruments of the drum type are used. In other cases and for general experimental purposes, the most common method of measurement is by the Venturi or Pitot tubes or the determination of a frictional resistance.

The Pitot tube measures the gas velocity at the point of measurement, and if it is inserted

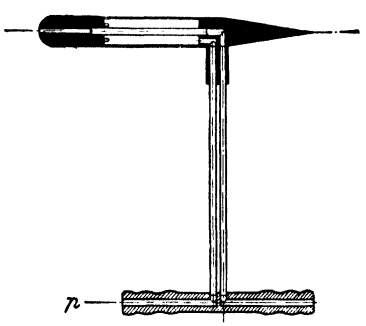


Fig. 21.—Pitot Tube

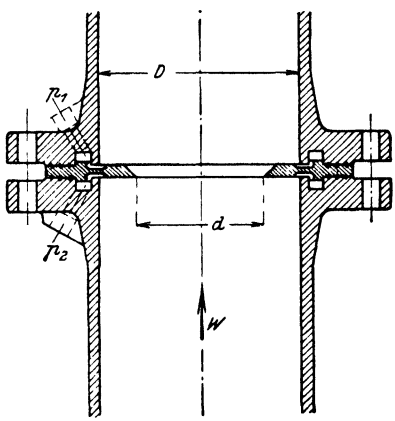


Fig. 22.—Throttle Plate

along the axis of a pipe carrying a flow of air or gas, it is necessary either to calibrate the instrument *in situ* or to know the relationship between the velocity at the centre and the mean velocity across the section for the character of pipe and the range of velocities used. The insertion of the Pitot tube necessarily disturbs the flow, and to minimize this it is constructed in the form shown in fig. 21. One orifice is directed towards the stream and experiences both static and dynamic pressure of the stream, while the other with openings along the stream lines experiences only the static pressure. Hence the difference measured by a delicate pressure gauge is that of the dynamic pressure $\frac{1}{2}\rho v^2$. The principle of the Venturi tube has been given in Chapter I. If the velocity of the gas at two points of different cross section are given by v_1 and v_2, neglecting any small difference in gas density at the two points, the difference of pressure $p_1 - p_2 = \frac{1}{2}\rho(v_2^2 - v_1^2)$. This equation holds only if there is no frictional resistance between the two points. In an actual construction, the frictional resistance of a length of tube, even if of smooth stream-lined surface, may be quite appreciable and will add to or subtract from the pressure difference according as the tube

is decreasing or increasing in section in the direction of flow. For simplicity and ease of reproduction, the measurement of gas flow by the frictional resistance of a fixed passage has proved most convenient.

Standard forms of orifice which can be inserted in a cylindrical pipe and used to measure gas flows by frictional resistance are shown in figs. 22 and 23. The former consists of a sharp-edged circular orifice cut in a flat thin metal plate, while the latter is a short tube of diminishing cross section, with a longitudinal section in the form of two quadrants of circles. For small differences of pressure,

$$V = kA\sqrt{\frac{2g(p_2 - p_1)}{\rho}}$$

where V = volume of gas flowing in unit time,
ρ = density of gas,
$p_2 - p_1$ = difference of pressure on the two sides of the orifice,
A = area of orifice $= \frac{\pi}{4}d^2$,
k = coefficient of contraction.

With the sharp-edged orifice the value of k varies from 0·6 when the diameter d is small compared to the pipe diameter D, to nearly unity when the orifice diameter is almost equal to that of the pipe. The coefficient for a rounded orifice of the proportions shown in fig. 23 is between 0·95 and 1·0.

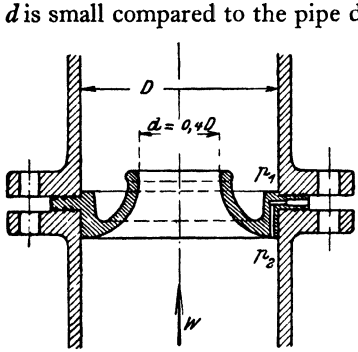


Fig. 23.—Standard Nozzle

It has already been mentioned that instruments of the character described, whose readings are proportional to the square of the quantity to be measured, have the disadvantage of low sensitiveness at low readings. A more serious objection to them arises where the mean value of a rapidly changing quantity is required. The air flow to an internal-combustion engine is intermittent, and any instrument whose readings are not proportional to the air flow would give an erroneous value to the mean flow. The average difference of pressure measured by Venturi tubes, Pitot tubes, and frictional resistance of orifices is proportional to the mean square of the varying velocity, which is greater than the square of the mean velocity. If the instruments are calibrated with steady flow, they read too high with varying flow. This may be corrected by using a pressure indicator whose readings are not proportional to the pressure but to the square root of the pressure, and which will then give indications proportional to the velocity. The advantages of using an ordinary and simple pressure gauge are so great, however, that an alternative plan is adopted. Between the measuring instrument and the engine, a larger box is inserted which acts as an air reservoir. The flow from box to engine is intermittent, but owing to the inertia of the large air

volume in the box, the flow into the box lags behind the flow out and is moreover uniform. If the air is being drawn from the atmosphere, there is no necessity to place the orifices in a tube, and they can be cut in thin metal plates forming part of the walls of the box. Such an arrangement is shown in fig. 24, the difference of pressure required to draw the air through the orifices being the depression below atmospheric of the air in the box.

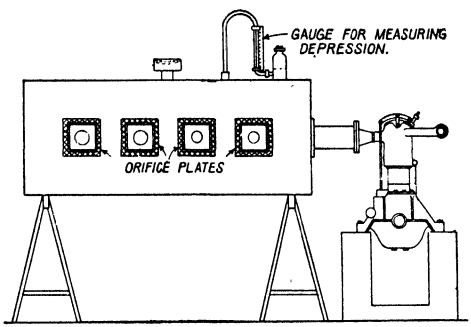


Fig. 24.—Measurement of Air Consumption of Internal-combustion Engine

There are several electrical methods of measuring the flow of gases. They have the advantage that the reading instruments may be at considerable distances from the point of measurement. In the hot-wire anemometer, an electric current is passed through a wire stretched across the flow and is adjusted to such a strength that the temperature of the wire, which is measured by its resistance, is always the same. The greater the flow of air past the wire the more current is required, and its value furnishes a measure of the gas velocity. If there are surges of gas, as in the case of the intake air of an internal-combustion engine, this instrument also gives high readings. The electric flow-meter which is illustrated in fig. 25 does not suffer from

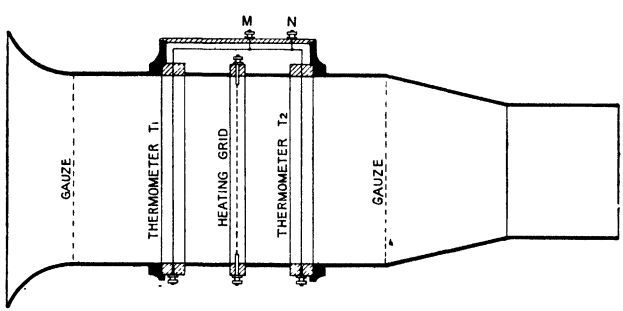


Fig. 25.—Electric Air Flowmeter

this defect unless the surges are very violent. A heavy electric current is passed through the heating grid, which is formed of wire or strip bent into a zigzag shape, and the whole gas is raised in temperature by a fixed amount of 2° F. or 3° F. This increase in temperature is measured by the difference in the electric resistances of the two thermometers T_1 and T_2. When there is no current passing in the grid, the temperatures at the two thermometers will be the same and a point M in the connecting wire will have a potential mid-way between that of the two outer ends. If heat is applied until the

mid-point is at N, the resistance of T_2 is greater than that of T_1 by an amount representing the fixed temperature rise. The current in the grid necessary to bring the mid-point to N increases with the flow, which it accordingly measures. This instrument has been used to measure the air consumption of an aeroplane engine at high altitudes. It is also used to measure the flow of coal-gas and of natural gas in large quantities.

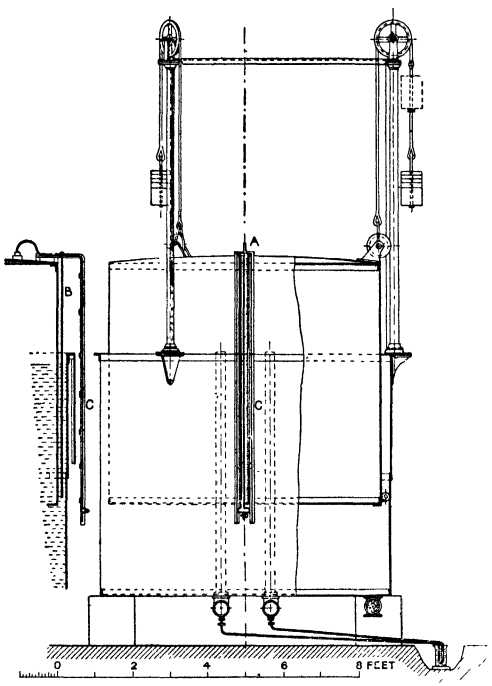


Fig. 26.—Gas or Air Holder for Internal-combustion Engine

Although some of the instruments described can be used where there are quick periodic variations of flow, the only method where the variations are slow is that of the gas-holder. The engine or plant is fed with air or gas from the usual system until conditions are steady, and is then supplied from a gas-holder during the test period. The most convenient form of holder is the floating-bell type with constant pressure. The volume of gas used is measured directly from the fall of the bell, and the weight is obtained by means of the density at the pressure and temperature within the bell. A design suitable for tests on a small engine is shown in fig. 26. The chief source of inaccuracy is the variation of temperature throughout the holder. Owing to their size it is necessary to erect them in the open and the temperature at one point may be considerably different from that at another, particularly in sunny weather. In some cases it is possible to use a supply of gas from small containers at high pressures, the amount used being measured by the fall of pressure in the gas bottle. This method, however, necessitates continuous adjustment of the outlet valve to give a constant flow.

6. PRESSURE

The measurement of a pressure is required in several of the flow-meters for liquids and gases. The regulation of the draught in furnaces also involves the determination of a small pressure, while in steam boilers high pressures are to be measured. For many purposes a simple U-tube pressure gauge

is sufficiently accurate. For example, if a flow of air or gas is measured by the frictional resistance in passing through an orifice and the pressure difference is 2 in. of water head, this height can be measured with an accuracy of 1 per cent corresponding to an accuracy of half of 1 per cent on the flow.

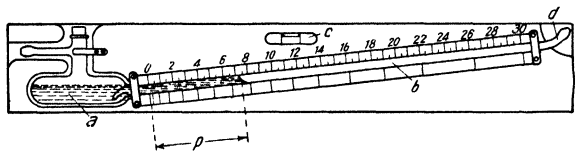


Fig. 27.—Inclined Pressure Gauge

For very small heads an inclined form of gauge can be used as shown in fig. 27, the vertical height being measured by the horizontal distance to the surface and the known inclination of the gauge. Another method of obtaining a magnified reading is shown in fig. 28. The U-tube is formed by two concentric cylinders, the inner one being pierced near the bottom for intercommunication. The pressure is applied to the concentric space between the cylinders, which is closed, while the central space is open to the atmosphere. To avoid rust and reduce surface tension effects paraffin oil is used in the gauge. A positive pressure causes the surface in the outer space to be depressed and that in the central space to rise, and vice versa with a negative pressure. The level in the central space is measured by means of a float A, partly counterpoised by the weight C with a connecting silk fibre passing over the small pulley B. To reduce friction the spindle of the pulley is mounted on jewel bearings. Its rotation is measured by the pointer P, and differences of level of 0·001 in. produce a deflection of 1 division of the scale D.

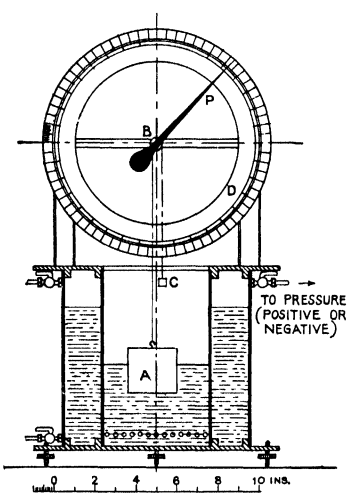


Fig. 28.—Gauge for measuring Small Pressure Difference

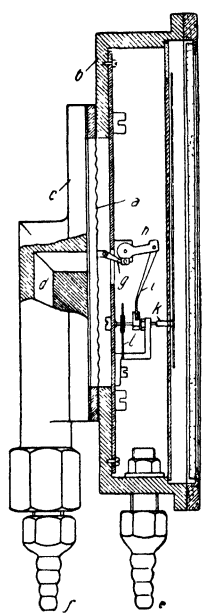


Fig. 29.—Diaphragm Pressure Gauge

It is unnecessary here to consider the use of the barometric tube for the measurement of considerable pressures or of the common Bourdon gauges for pressure and vacuum measurements. One example of a diaphragm gauge

is shown in fig. 29, the mechanism by which the small deflection is multiplied to give a large rotation of the pointer being self-explanatory. Recording mechanism for the continuous registering of the pressure as in the Deacon water-meter can be attached to most of these gauges or to the float type of gauge.

7. TEMPERATURE

The measurement of the temperature of water or other fluids presents little difficulty, and mercury thermometers of ordinary type may be utilized. The determination of the temperature within a furnace or of a hot gas is less simple. For temperatures up to 1000° F., the platinum resistance thermometer will furnish accurate values if the necessary precautions are taken, while thermo-couples may be used up to 2000° F. When used for the measurement of a gas temperature both these instruments are liable to error unless protected from the effects of radiation. In general, the gases are in a flue or furnace with walls whose surfaces are at a much lower temperature. Any body placed in the gases takes up such a temperature that the heat it is receiving from the gas is equal to that which it radiates to the walls, and unless the latter is zero the body will never reach the gas temperature. It is therefore necessary, particularly when the gas temperatures are very high, to protect the thermometers by placing the bulb within a concentric shield, leaving a small space between bulb and shield for the circulation of the gas.

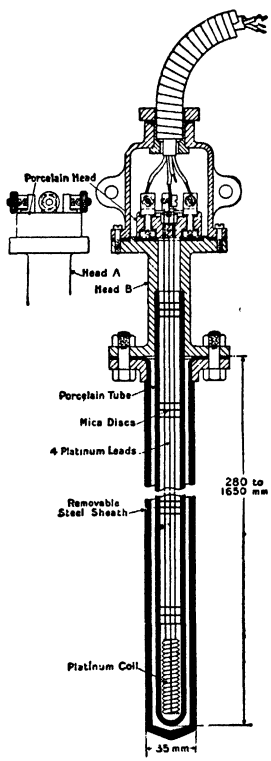


Fig. 30.—Platinum Resistance Thermometer

The construction of a platinum resistance thermometer for furnace work is shown in fig. 30. A coil of fine platinum wire is wound on mica and contained at the bottom of a protecting tube of porcelain. Leads of heavier wire connect it to a Wheatstone bridge with equal ratio arms as illustrated in fig. 31. A second pair of exactly similar leads are also contained in the tube and are connected to the adjacent arm of the bridge. If also a resistance is added to this arm by means of coils until the bridge is balanced, this added resistance must be equal to that of the fine platinum coil forming the thermometer bulb. If R is the resistance of platinum at temperature t and R_0 that at zero temperature,

$$R = R_0(1 + at + bt^2).$$

The values of the constants a and b can be found by measuring the resistance

at zero temperature and at two other known temperatures. The relation can then be used for the determination of the unknown temperature at which a resistance has been measured. Where great accuracy is not required, the resistances in the bridge arms may remain fixed and the deflection of the galvanometer produced by the change in resistance when the thermometer is heated can then be used to indicate the temperature or to operate a recorder of the temperature.

In thermo-electric pyrometers the temperature is measured by the electromotive force produced at the junction of two dissimilar metals fused together and placed at the point where the temperature is required. They can be used for higher temperatures than the resistance thermometers and will measure the temperature at a point, while the resistance thermometers are more bulky

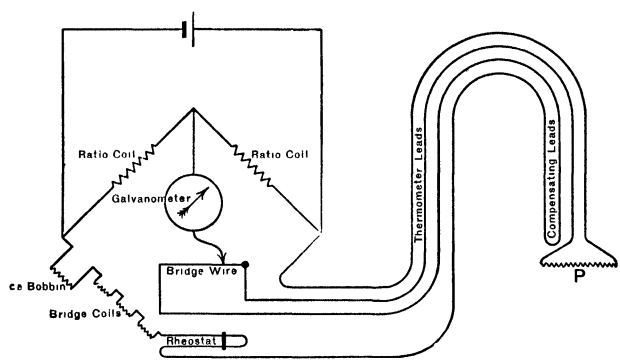


Fig. 31.—Electric Circuit for Platinum Resistance Thermometer Measurements

and give the average temperature over a small volume. They respond quickly to changes of temperature, but they are less accurate over the range in which the resistance thermometers are also used. If one metal is an alloy the relationship between the electromotive force and the temperature is given by

$$E = a + b \log t.$$

The electromotive force can be measured by a potentiometer, but it is usually sufficient to note the current when the couple is placed in circuit with a high resistance and a galvanometer. For ordinary work thermo-electric couples of which one metal is the alloy constantan and the other iron or copper are used, but the oxidation of the iron or copper causes gradual deterioration. For more accurate work, one metal is platinum and the other platinum-iridium or platinum-rhodium. The constantan couples are only suitable for medium temperatures, the platinum couples being used for high temperatures.

For temperatures above 2000° F. it is necessary to use some form of radiation pyrometer, though they can, of course, also be used at somewhat lower temperatures. The Féry radiation pyrometer is one of the most accurate instruments of this class, and is shown in fig. 32. The instrument is directed

to the point whose temperature is desired by means of the eyepiece O, and the radiation falling upon the concave mirror M is concentrated upon a thermo-couple of very small heat capacity. The temperature of the thermo-couple is thus raised and the resulting electromotive force is recorded by a sensitive galvanometer. The deflections are proportional to the fourth power of the *absolute* temperature. The amount of heat falling upon the mirror is inversely proportional to the square of the distance from the source of heat if the small absorption of the intervening air is neglected. The area of the image formed at the focus is also inversely proportional to the square of the distance, except when the distance is very small. Hence the intensity of the heat per unit area at the focus is independent of the distance, and the heat received by the thermo-couple, which is very small and less than the image, is independent of the distance. The pyrometer readings are therefore very nearly unaffected by the distance at which it is placed.

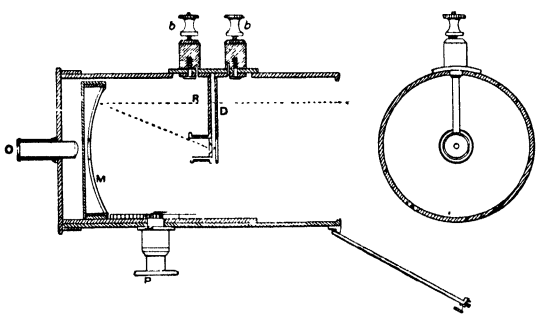


Fig. 32.—Féry Radiation Pyrometer

It is sometimes necessary to estimate the temperature at a point in a furnace where none of the methods described is available. An approximation to the temperature can be obtained by inserting a series of pellets formed of metals or alloys of known melting-point. These are left for a sufficient length of time to acquire the temperature of the region and are then withdrawn, when the temperature attained must have been above the highest melting-point of those that have melted and below the lowest melting-point of those that have not. The Seger cones, to which reference has previously been made, are a series of clay mixtures of different composition which are used instead of metals. They are in the shape of pyramids about 3 in. in height and with a base of $\frac{5}{8}$ in., and on fusing the point bends over towards the base. The difference in the melting-points of consecutive cones in the series is 36° F. for low-temperature cones and 72° F. for high-temperature cones. The range covered is from 1350° F. to 3600° F.

8. CALORIFIC VALUE

The calorific value of a solid fuel is found by means of the bomb calorimeter, one type of which is shown in fig. 33. A small quantity of the fuel is weighed and placed in the crucible at the centre of the mild steel bomb, which is immersed to the neck in a bath of water. Oxygen is introduced at a high pressure and the bomb closed.

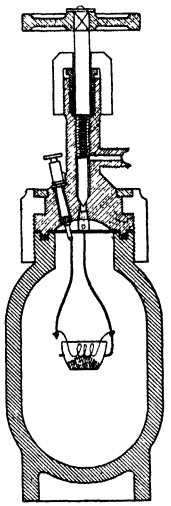


Fig. 33.—Bomb Calorimeter

The fuel is ignited by the passage of an electric spark at its surface. The water is continually stirred and its temperature is noted before the ignition, during the burning which may take a minute or thereabouts, and afterwards while the bomb and water are cooling. From the last information the loss of temperature during the combustion period can be deduced and added to the registered rise in temperature. The heat produced is then equal to the product of the water equivalent of the apparatus into the corrected temperature rise. It is not possible to obtain the water equivalent of the bomb from its weight and specific heat

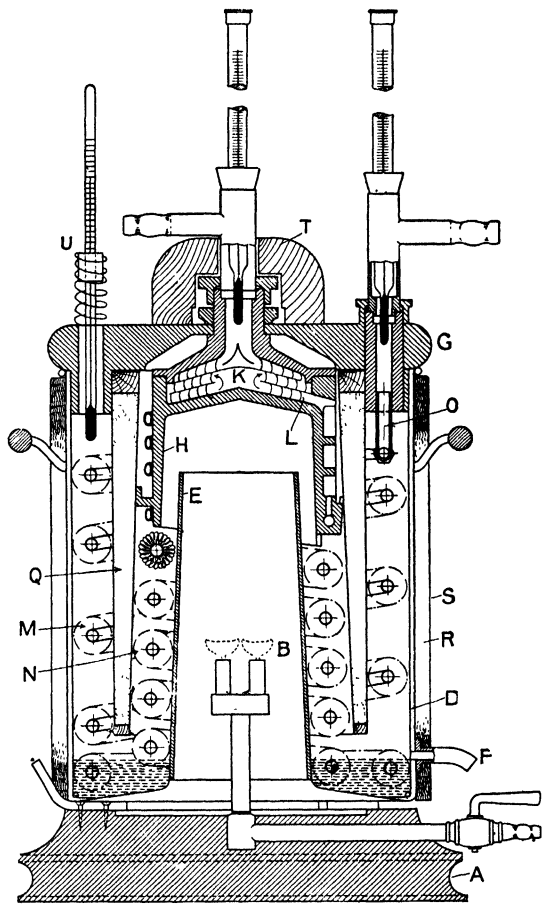


Fig. 34.—Boys' Calorimeter

as the upper portion does not attain the temperature of the remainder. It is therefore necessary to calibrate the apparatus by the combustion of a fuel of known calorific value such as cane sugar.

The calorific value of a gas is measured by one of the forms of continuous flow calorimeter. In London the standard type is that of Boys given in fig. 34. The gas is consumed at a constant rate, measured by a meter through

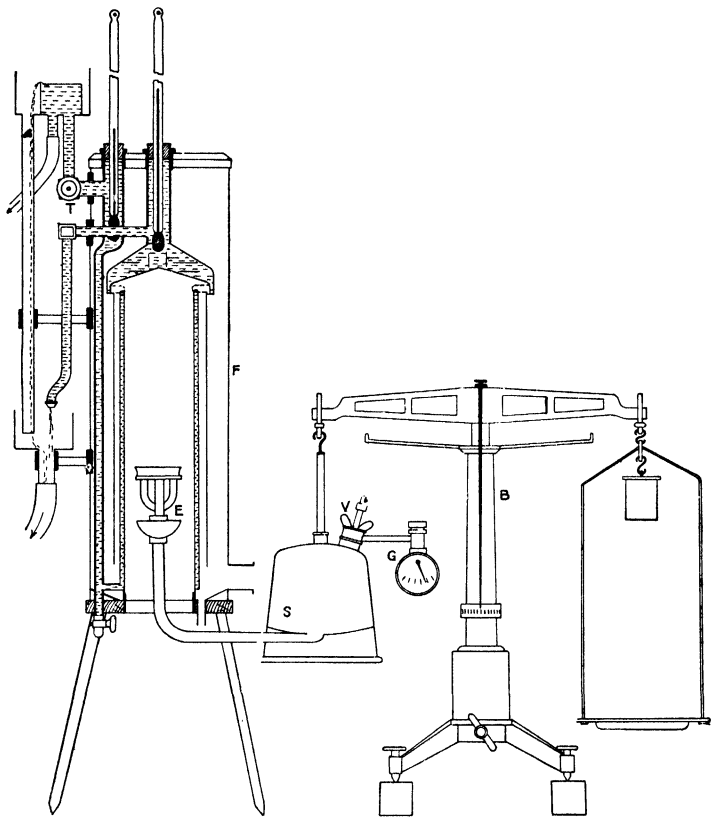


Fig. 35.—Calorific Value of Liquid Fuel by Continuous Flow Method

which it is passed, in the burner B. The hot gases pass up the chimney E, down the surrounding space, and up again and out by holes in the cover G. A circulation of water is maintained through the coils M and N and through the chamber LK, the initial and final temperatures being measured by two delicate thermometers. The gases give up their heat to the water, and their temperature as they escape to the atmosphere is usually the same as that of the atmosphere, the thermometer U serving as an indication thereof. After an initial heating period the apparatus attains a steady temperature and all the heat is transmitted to the water, none being used to raise the

temperature of the apparatus. The water in the outer coil M is cold and there is no appreciable radiation. The calculation involved is therefore very simple.

If H is the calorific value per cubic foot of gas,
V the cubic feet of gas burnt in period of test,
w the weight of water flowing in period of test, and
$t_2 - t_1$ the rise in temperature of the water,

$$H = w(t_2 - t_1)/V.$$

The calorific value of a liquid fuel is often found by the bomb calorimeter as used for solids. As the amount of oxygen which can be inserted in the bomb is only sufficient to burn a very small quantity of fuel, any loss of fuel by evaporation between weighing and its closure within the bomb may lead to considerable error. An application of the gas calorimeter has therefore been devised for the measurement of the calorific value of liquids. The arrangement is shown in fig. 35. The calorimeter F is of the Junkers type and is on the same principle as that of Boys, and is also used for gases. The water passes upward in the narrow annular space between two concentric cylinders and can be regulated by the valve T. The fuel is contained in the stove S of the Primus type, and consumed at the burner E. A small air pressure is formed in S by a bicycle pump operating through the valve V, and the fuel is forced out of a very small orifice at E. A preliminary heating of the burner by methylated spirits causes the fuel to vaporize as it emerges, and the subsequent combustion maintains the heat for the further vaporization. The stove is suspended from one arm of the balance B and the rate of burning can thus be obtained directly. The manipulation of the calorimeter and the calculations are the same as for a gas. The test can be continued until any required amount has been burnt and a corresponding accuracy attained.

9. INDICATED HORSE-POWER

The choice of an instrument for the measurement of the indicated horse-power of an engine, or of the periodic variation of any pressure, depends upon the frequency of the varying pressure. In order that the instrument should accurately record the variation it is necessary for it to be quick-moving and for the natural frequency of any part of it to be many times that of the pressure. The natural frequency can be increased by reduction in weight and by increase in the controlling force, but the latter factor also leads to reduced sensitivity. In order to increase the sensitivity and magnify small deflections, mechanical links can be used only at small speeds while optical methods must be used at high speeds.

An indicator for slow-speed steam- and gas-engines is shown in fig. 36. In the body of the instrument a small cylindrical chamber is cut with a piston acted on underneath by the steam or gas, and on the top by the spiral spring. The vertical movement of the piston is magnified by the links and

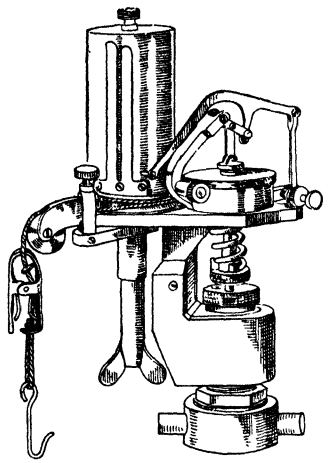


Fig. 36.—Dobbie-M'Innes Indicator

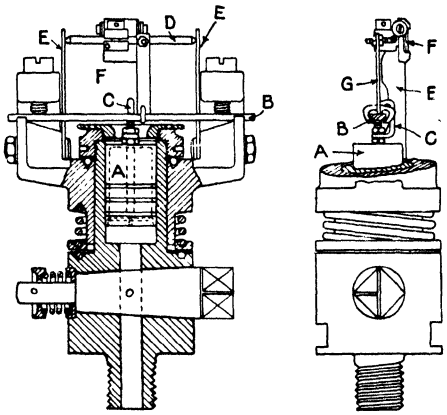


Fig. 37.—Hopkinson Indicator

the pressure is recorded by the vertical movement of the pencil against the drum. The reproduction of the engine stroke is obtained by an oscillation of the drum maintained by a cord attached to a suitable point of the engine. The records are made upon paper or thin card.

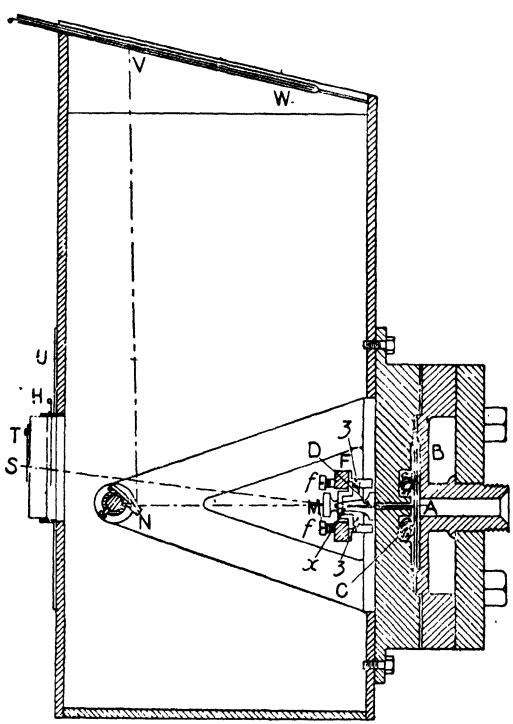


Fig. 38.—Diaphragm Optical Indicator

The moving parts of the Hopkinson indicator, which is designed for higher speeds, are much smaller. A section is shown in fig. 37. The gas pressure acts upon the piston A which operates against a beam spring B, the connexion being by the hook C. Above the spring a spindle D held between the uprights E carries a mirror F. A link G connects the spring excentrically to the spindle so that the vertical movement of the spring produces rotation of the spindle. A beam of light impinges upon the mirror, and the variation of pres-

sure and rotation of the spindle results in a vertical movement of the reflected beam. The piston stroke is represented by an oscillation of the upper part of the whole instrument about its vertical axis. The instrument is only accurate at moderate speeds owing to its natural frequency not being sufficiently high, and also owing to friction of the piston and the pivots of the spindle.

For very high speed work diaphragm indicators are to be preferred to piston indicators. The piston friction is eliminated and it is possible to increase the natural frequency to 2000 per second. With such a frequency the indicator diagram of an engine running at 1500 r.p.m. can be reproduced without indicator ripples. A compact form of optical diaphragm indicator is shown in fig. 38. The gas pressure acts on the diaphragm A and the deflection transmitted by the rod D causes the mirror M to tilt about an axis in the plane of the paper. A second mirror N oscillates in synchronism with the piston and a beam of light from the orifice S by reflection at M and N receives both pressure and stroke motion, and the diagram is recorded on the photographic plate VW. To prevent the reduction of its elastic properties by heat, the diaphragm is cooled by a rapid water circulation through the space B and by a moist wick C on the atmospheric side. A diagram obtained by this instrument on a 40-h.p. single-cylinder petrol-engine is reproduced in fig. 39.

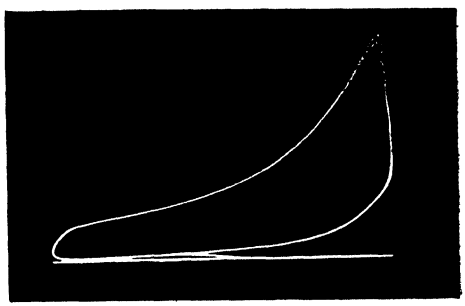


Fig. 39.—Indicator Diagram of Petrol Engine

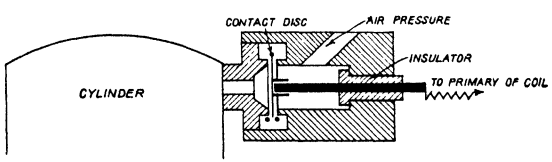


Fig. 40.—Disc Valve for Electrical Indicator

The necessity of having a high natural frequency is avoided in a new type of indicator developed at the Royal Aircraft Factory. There are also other designs, such as the Juhasz, which are based on the same principle and are less complicated but are also less accurate. Instead of constructing an instrument to follow the pressure during every cycle, the pressure is recorded at two points only during each cycle and it requires one or two hundred revolutions to obtain a complete diagram. A small disc valve is mounted in a container which will screw into a spare sparking plug of an engine as shown diagrammatically in fig. 40. A known pressure from an air bottle is intro-

duced behind the valve which will remain on the seating nearest the engine while the cylinder pressure is less than that of the air, and will transfer to the seating away from the engine when the cylinder pressure is the greater.

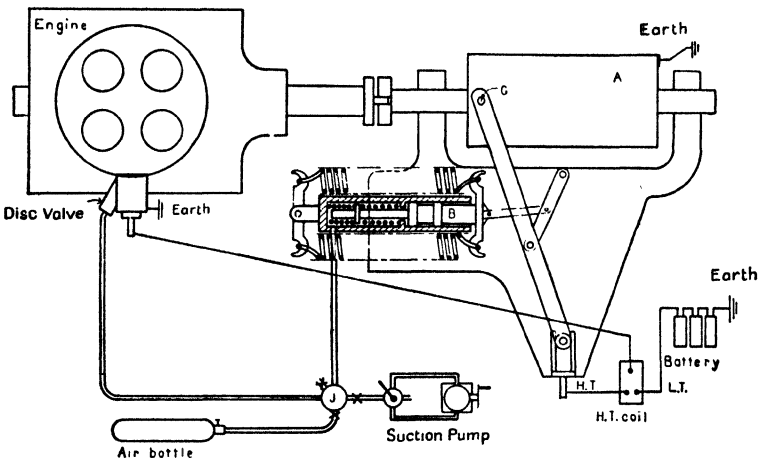


Fig. 41.—Recording Apparatus for Electrical Indicator

The disc will cross and recross during each cycle, and in its movement it breaks and remakes the primary circuit of an induction coil. A spark therefore occurs at each break in a spark gap of the secondary of the coil. The spark gap is formed between a point C and a drum A connected to the crank

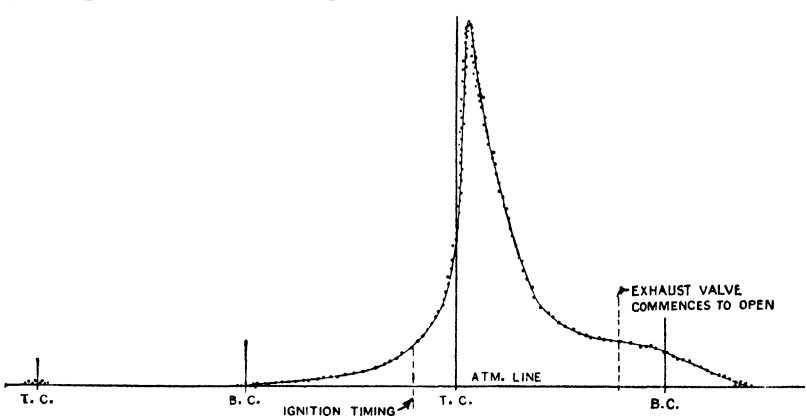


Fig. 42.—Indicator Diagram on Time Basis as recorded by Electrical Indicator

shaft of the engine and rotating at engine speed as illustrated in fig. 41. Thus the position of the spark along the circumference of the drum gives the time of its occurrence. The drum is covered with a black paper which the spark perforates, making the permanent record. The same air pressure

which is applied to the disc is also applied to a piston B in a cylinder parallel to the axis of the drum, and the sparking-point is attached to the piston. Hence the position of the spark along the generators of the drum records the pressure. By gradually changing the pressure the whole diagram is obtained. It is on a time basis, as shown in fig. 42, and requires to be replotted on a stroke basis in order to obtain the diagram in the usual form and for purposes of integration of the work.

10. BRAKE HORSE-POWER

In cases where there is a clear length of shafting the horse-power transmitted by the shafting can be found while the plant is under the usual conditions of operation. Thus the power transmitted by the propeller shaft of a marine engine can be obtained by measuring the twist of a portion of the shaft. In many other cases, however, there is no such clear length of shafting available. With reciprocating engines the brake horse-power can then be deduced from the indicated horse-power by subtracting the power required to run the engine light at the same speed. This method assumes that the frictional losses in the engine at a given speed are independent of the load. An alternative method is to run a special test of the engine in which the brake horse-power is used and measured in an absorption dynamometer, and to assume that the same horse-power is being produced under the same engine conditions when the engine is working its normal plant.

If T is the torque applied to a shaft whose diameter is d and coefficient of rigidity N, while the angle of twist produced in length l is $\theta°$,

$$T = \frac{N d^4 \theta}{584 l} = a\theta.$$

The horse-power transmitted when the shaft is running at n r.p.m. is given by the following equation, the lengths being in inches:

$$\text{Horse-power} = \frac{2\pi n}{12 \cdot 33{,}000} a\theta.$$

The value of the constant a can be calculated if the modulus of rigidity of the material is known, or it can be found experimentally by twisting the shaft when at rest. One end of the shaft is kept fixed and a known couple applied at the other by means of a weight at the end of an arm. The consequent torsion of the length l which is to be used in the power tests is then measured. The principle of the Hopkinson-Thring torsion dynamometer is shown in fig. 43. A disc A is fixed rigidly to the shaft at the point P and a sleeve B is fixed to the shaft at Q. The sleeve is flanged at C, which is near and parallel to the disc A. When there is a twist of the shaft between P and Q there will be a relative movement of A and C. The mirror M in contact with A and C registers this relative motion by its rotation of a beam of light directed upon it, the reflected beam being deflected through twice the angle of the mirror. The reception and reflection of the beam can take place only

once per revolution, but readings can be taken by persistence of vision. With some types of instrument two or four mirrors are placed round the circumference. The length of shaft required varies from 1 to 5 ft. according to the size of the shaft and the torque transmitted.

For the measurement of small horse-powers by an absorption dynamometer, some form of rope or band brake is most commonly used. Types of these brakes are shown in figs. 44 to 46. The power is used in turning a flywheel against the friction of the band upon its circumference, the amount of the friction being adjustable by means of the tension in the band. If F is the difference in the forces on the two ends of the band and s the length of band in feet which would surround the wheel,

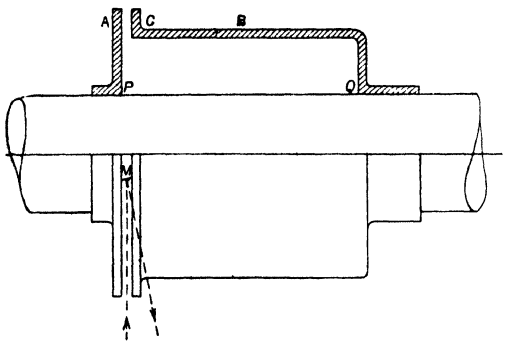


Fig. 43.—Hopkinson-Thring Torsion Dynamometer

$$\text{horse-power} = \frac{Fsn}{33{,}000}.$$

In the simple Prony brake shown in fig. 44 two blocks of wood shaped to the wheel take the place of the band, and are clamped to the circumference. The torque tending to rotate them and the clamping is measured by the

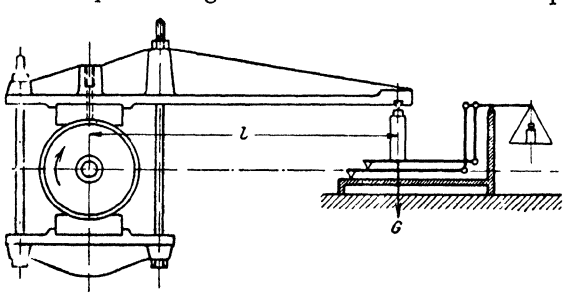


Fig. 44.—Prony Brake

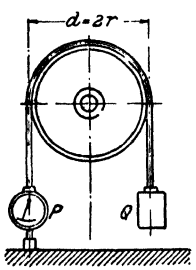


Fig. 45.—Band Brake with Spring Balance

force exerted on the weighing machine. The arrangement shown in fig. 45 acts to some extent as a speed regulator. The band is attached to the spring balance P and supports the weight Q, the direction of rotation being from Q to P. The friction exerted by the wheel on the band is equal to the difference Q − P. If the speed increases Q tends to rise, thus releasing the tension on the spring balance, and Q − P increases and restrains the increase in speed. An automatic arrangement of a more complicated character is shown in fig. 46.

When using the band brake the whole of the work done by the engine is dissipated as heat which is given to the band, the wheel, and the surroundings. The amount of work which can be thus expended without damage to the brake is limited. It can be increased by running a stream of water over the band, but a more satisfactory method is that employed in hydraulic dynamometers. The wheel runs within a stationary casing, and the force between the faces of the wheel and the parallel internal surfaces of the casing is transmitted by a stream of water passing between them. In the Froude brake, of which an end elevation is shown in fig. 47 and a longitudinal section in fig. 48, the parallel faces of the wheel and casing are scooped out and the force between the two is greatly augmented by the passage of the water backward and forward between them. The water is thus continually given kinetic energy which is shortly afterwards destroyed and converted

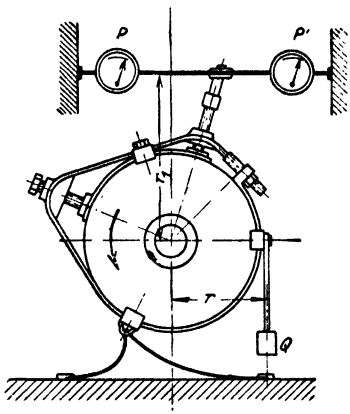


Fig. 46.—Brauer Automatic Brake

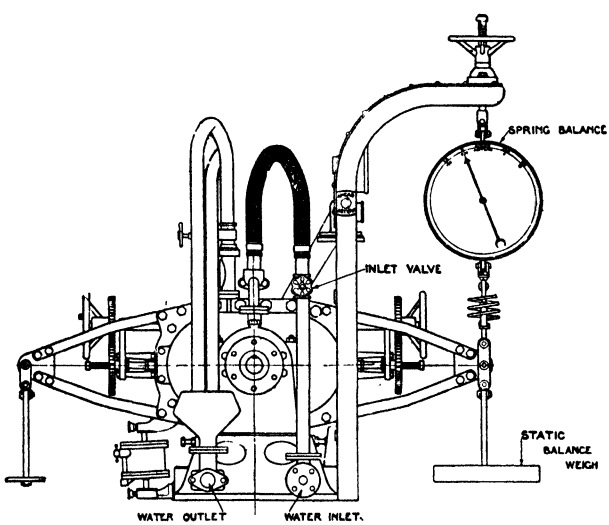


Fig. 47.—Froude Water Brake

into heat. The stream of water carries away the heat, and its amount is adjusted so that there is no large rise of temperature. The casing is supported on bearings, and the torque upon it is measured by that necessary to keep it from rotating. Brakes of this character can be made to absorb

and measure the horse-power of the largest of engines. The load can be adjusted by damping plates which can be moved in and out between the wheel and casing. thus reducing or increasing the size of the water passages.

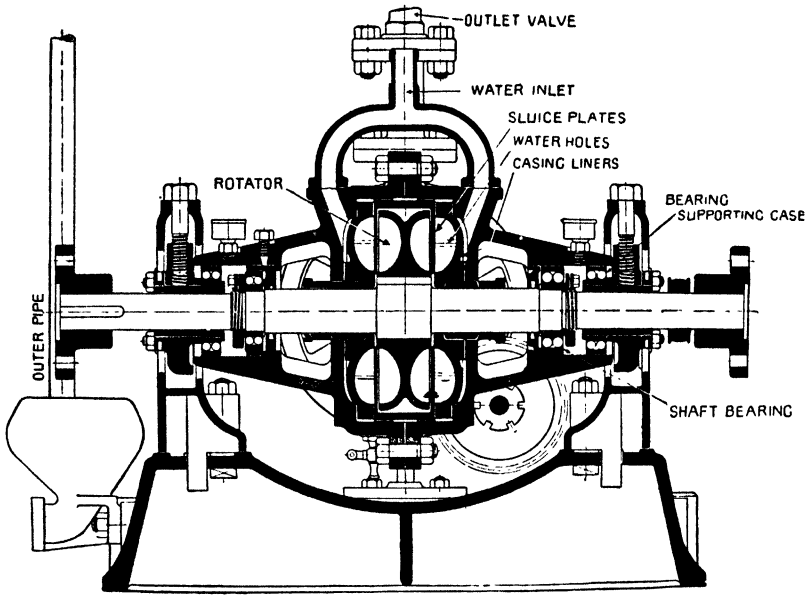


Fig. 48.—Longitudinal Section of Froude Water Brake

A convenient and accurate method of measuring the power of a small engine is that of the electro-dynamometer. The engine is used to drive a dynamo whose field magnets and casing are supported by bearings or rollers so that they are capable of a small rotation. The torque between the armature and the field is measured by the equal and opposite torque to be applied to the casing to prevent its rotation. This can be applied by a weight at the end of an arm as shown in fig. 49. The load is adjusted to give any required speed by altering the field and armature currents by means of external resistances. If a suitable supply of electrical energy can be drawn from the mains, the dynamo may be used as a motor to rotate the engine without its fuel supply, the torque between field and armature then measuring the friction in the engine.

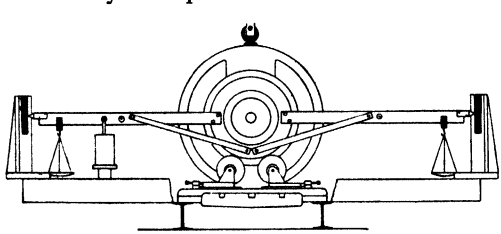


Fig. 49.—Electro-dynamometer

Values of the brake horse-power of an engine which are sufficiently accurate for many purposes can be obtained by means of the fan brake

shown in fig. 50. The two arms of the brake are clamped at the end of the engine shaft, and the plates are rotated with their areas perpendicular to their direction of motion. The power is thus dissipated in churning up the air, and is measured by the calibration of the fan on a motor or engine of known power. The range of speed and power can be altered by adjusting the distance of the plates from the axis of the shaft. It is most important that the

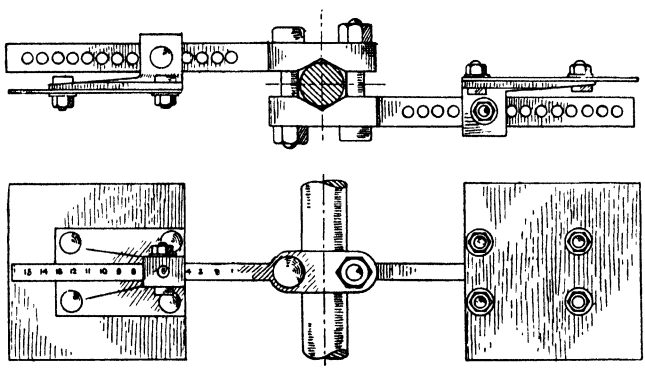


Fig. 50.—Fan Brake

calibration should take place under conditions similar to those under which the brake is used. The proximity of walls, floor, or parts of the engine itself to the blades during their passage through the air may seriously affect the power required to rotate them at a given speed and radius.

11. GAS ANALYSIS

The performance of a gas-furnace or of a gas-engine depends not only on the calorific value of the gas but also upon its constitution. The amount of air required for perfect combustion varies with the proportions of the various combustible constituents. The correct regulation of the air supply can best be attained by analysing the exhaust gases. Maximum combustion efficiency is obtained when the exhaust gases contain no combustible gas, i.e. no carbon monoxide, hydrogen, or hydrocarbons. The same conditions apply in the case of petrol and heavy-oil engines. In many high-temperature furnaces the presence of carbon monoxide is unavoidable as dissociation of carbon dioxide occurs, but it is equally important to obtain a knowledge of the constitution and properties of the flue gases. The furnace conditions can not only be deduced therefrom, but the value of the gases for exhaust heating or other utilization can be found.

The principal constituents of the exhaust gases are usually carbon dioxide CO_2, carbon monoxide CO, oxygen O, hydrogen H, and nitrogen N. In some cases there may also be some hydrocarbons, C_nH_m. Water vapour, H_2O, is also present but is usually condensed previous to analysis. A quantity of exhaust gas is passed into a burette over mercury and its volume measured.

It is then forced from the burette into an absorption vessel containing a solution which will absorb one of the gases mentioned. The process of absorption is assisted by shaking or rocking the vessel for some time, and the gas is then passed back into the burette where the loss of volume can be measured. A solution of caustic potash or soda is used for the absorption of carbon dioxide, cuprous chloride solution for carbon monoxide, and pyrogallic acid for oxygen. Hydrogen can be estimated by adding a noted volume of oxygen, passing an electric spark, and finding the diminution in volume due to the formation of water and its condensation. The amount of gas remaining after the other constituents have been removed is assumed to be nitrogen. For the detailed technique of chemical methods of gas analysis the reader is referred to separate books on the subject.

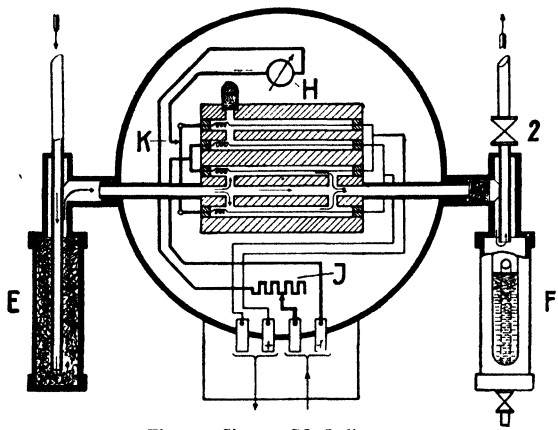


Fig. 51.—Siemens CO_2 Indicator

Various physical methods of continuously measuring the CO_2 and CO contents of flue gas have also been devised. The viscosity of carbon monoxide is similar to that of air while that of carbon dioxide is less, and this property can be used as an indication of the amount of dioxide present. The greater density of carbon dioxide can also be utilized. The thermal conductivity of the dioxide is less than that of air, while that of the monoxide is almost the same as that of air. An instrument based on this principle is shown in fig. 51. Two fine platinum wires are stretched along the axes of narrow tubes through which the flue gas is passing, and they are heated by an electric current. A similar pair of heated wires is surrounded by air. There being less conduction of heat from the wires in the flue gas, their temperature and resistance will be higher than that of the other pair of wires, the difference increasing with the CO_2 content. The two sets of wires are placed in adjacent arms of a Wheatstone bridge, and this difference in resistance is indicated by the deflection of a sensitive galvanometer. An instrument of similar construction but with thicker wires at higher temperature becomes an indicator of CO or H content if free oxygen is added to the gas. Owing to the catalytic action of the platinum surface combination of the carbon monoxide or hydrogen with the oxygen takes place although the temperature may be too low for flame production. Heat is liberated and the wire increases in temperature and in resistance, the effect being shown by the galvanometer deflection.

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