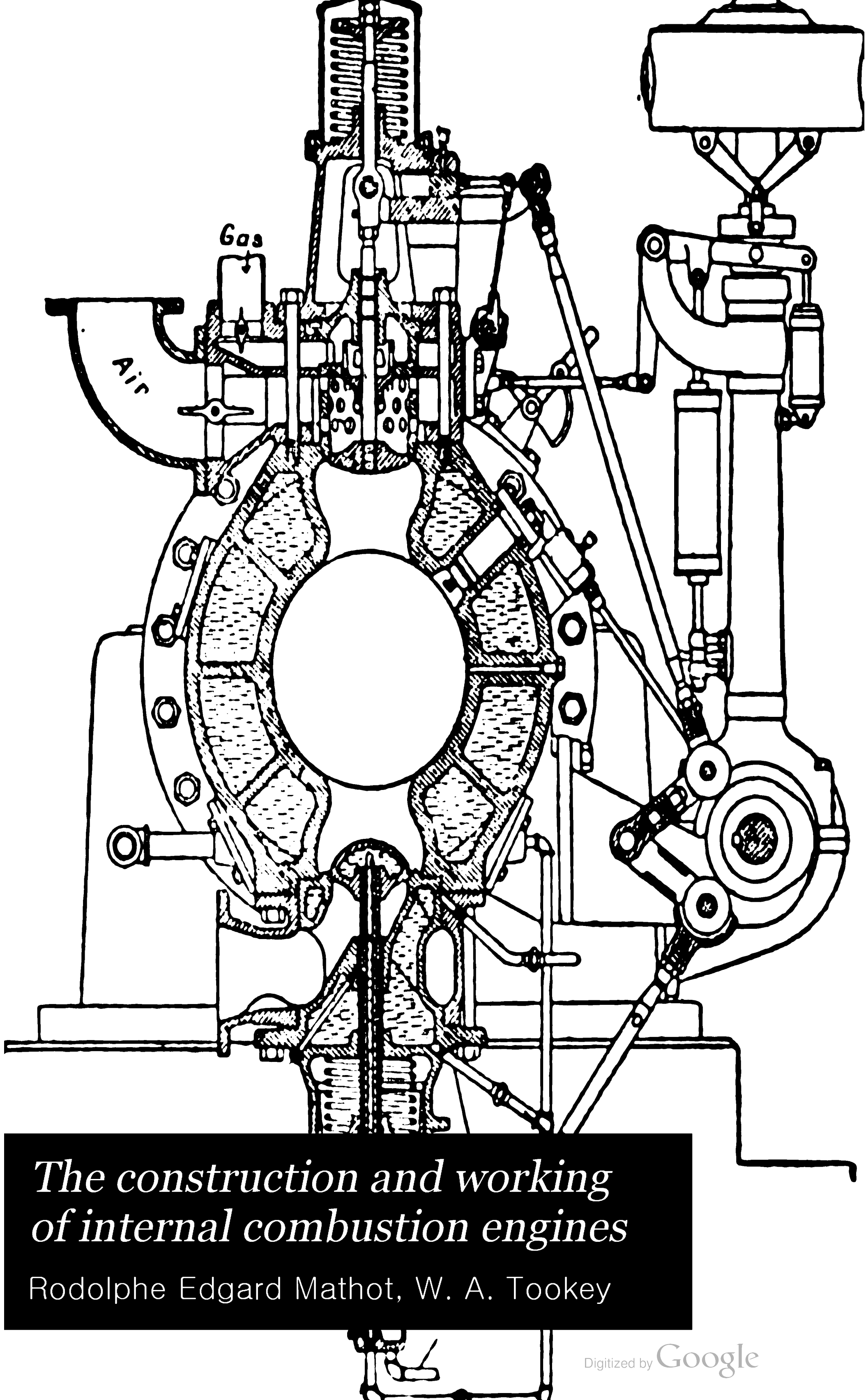

The construction and working of internal combustion engines

Rodolphe Edgard Mathot, W. A. Tookey

VFM

THE CONSTRUCTION AND WORKING OF INTERNAL COMBUSTION ENGINES

THE CONSTRUCTION AND WORKING OF INTERNAL COMBUSTION ENGINES

A Practical Treatise upon Methods of Construction, with Calculations for the use of Engineers, Manufacturers and Users, and a Critical Study of Present-day Types

BY

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TRANSLATOR'S NOTE

At the express wish of the Author, the translator has condensed or omitted some descriptions of little known types of European construction of no particular interest to British and American readers, and has included some new matter to record recent progress.

The original work has been entirely rearranged and recast in order to deal with the subject in a systematic manner. Thus, the first three chapters include the Author's remarks upon the subject generally and are principally concerned with the progress in the past, scope in the present, and possibilities in the future. Chapters IV. to XIII. have, been arranged to bring under review all matters concerned with design and construction. Chapters XIV. to XVI. similarly co-ordinate the Author's valuable hints upon the determination of power and efficiency, and upon the adequate classification of test results.

The lists of Gas and Oil Engine and Gas Producer Makers, and of various books, articles and papers published upon the internal combustion engine, have been brought up to date, separated from the context, and appear as appendices.

PREFACE

Although the internal combustion engine has been applied to industrial use only during the last fifty years, it can well substantiate the claim to have attained a degree of perfection closely approximating to that held by the steam engine of to-day. The studies and researches of numerous scientists, and the many experiments conducted by skilled engineers in charge of constructional workshops, have enabled certain general principles to be established and formulated, the application of which must be considered indispensable to the proper construction of internal combustion engines.

A list of the more important books that have been published in different countries, of valuable papers read before technical societies, and of informative articles in the technical press, is given in an appendix to the present volume. From these different contributions it would appear that the internal combustion engine has been dealt with by so many people, so frequently, and from so many different points of view, that the whole subject has been exhausted, and, therefore, a further volume could be merely a repetition of statements that have already been made several times over, and could not contain any fresh matters of interest, seeing that the most complete books are of somewhat recent date, and, consequently, record the state of progress prevailing at the present time.

Such an objection would be well founded if viewed from a purely descriptive stand-point, because this has been the position more frequently taken up by the various authors mentioned. But the objection cannot be sustained if the subject be examined with regard to the precepts of theory and of practice as far as experience permits at the present day. It is precisely from this point of view that the Author has written the present work.

He has reviewed the well-known arrangements adopted by the chief gas engine builders, that in a sense have become standardised, but, as far as is possible, he has avoided giving descriptions that have already been made public by several authors.

He has endeavoured to discuss the various mechanical details critically, and from a comparative point of view, and therefore has been compelled to give brief descriptions of the general features of the types adopted by some of the makers. Each time occasion has offered, he has given expression to his appreciation of various methods employed, some of which have been adopted as the outcome of the Author's conferences with the engineers connected with the principal constructional establishments. He has also dealt with the various details that go to make up the complete engine, and has given a series of practical formulæ for calculating their principal dimensions. In this manner, it is hoped that interesting, useful and original material has been provided.

In order to keep the size of the book within bounds, that portion allotted to descriptions of leading features and information of a general character has been condensed, with the knowledge that those requiring more information upon such points have already many valuable works to refer to.

The Author takes the present opportunity of expressing his obligations to those firms who have been good enough to send him information for incorporation in this volume, and for the manner in which they have permitted him to make a critical examination of their work. In so doing, they have presented a striking contrast to some of the smaller firms from whom it is practically impossible to obtain any particulars of what, according to the makers' claims, must be unsurpassed inventions, and upon the true value of which, alas! the public are precluded from forming an opinion.

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CHAPTER I

THE PROGRESS OF GAS POWER

WHILST the excellent qualities of town gas assured its acceptance as the best fuel for internal combustion engines by reason of the readiness with which it could be employed, the application of the gas engine, previous to the introduction of apparatus for producing low grade gas, was greatly restricted, owing to the fact that, for powers exceeding from 50 to 75 H.P., the cost of working became prohibitive. In order to compete effectively with steam engines, it has become a matter of paramount importance to produce a gas, cheaply and readily, capable of yielding good results from a simple and inexpensive plant.

Power-Gas Producers.—The low grade gas produced from the older forms of apparatus of the Dowson "pressure" type encouraged the utilisation of gas power in installations requiring 100 H.P. and over, but, beyond this figure, the complicated apparatus necessary for the production and purification of the gas, the capital outlay involved, and the amount of space occupied by the engine and producer, rarely showed to advantage when compared with a steam engine and boiler, especially as the latter could be used whatever kind of fuel was available—gaseous, liquid, or solid.

The above remark does not hold good in connection with pressure power-gas producers of the Duff, Mond, and similar types, equipped with apparatus intended for the recovery of ammonia and other by-products. With such installations it is possible to produce power with remarkable economy. Unfortunately, however, for remunerative application, experience has shown that the power developed should be at least 2,000 H.P., although Mr. Thomas Rigby, one of the competent engineers in the service of Messrs. Crossley Brothers, of Manchester, in a paper presented in 1907 to the *Engineering and Scientific Association of Ireland*, has declared that similar installations, but of a simpler character, with ammonia recovery, constructed by his firm, have now proved remunerative from 1,500 H.P. capacity.

"Suction" power-gas producers have solved the problem for general manufacturing purposes, and the processes of cleansing and washing the waste gases from blast furnaces and from coke ovens have proved, in an unexpected manner, the possibility of applying internal combustion engines for the highest powers demanded by the metallurgical industries.

The employment of certain low grade fuels, such as lignite and peat, have also given remarkably economical results, one effective H.P. per hour being readily obtained with $3\frac{1}{3}$ lbs. of peat, or with $1\frac{3}{4}$ lbs. of lignite. But the deposits of these fuels are sparsely distributed over the surface of the globe, frequently far away from industrial centres, so that it is of very rare occurrence that the full economy can be realised by the utilisation of the fuel in the neighbourhood of its production, that is to say, when high transport charges are not incurred.

In connection with the utilisation of these low grade fuels, it is of particular importance to realise that the cost of transportation depends not so much upon the fuel itself as upon the number of heat units that it furnishes.

The figures given above are rather higher than the average, as, from trials made by the *Saxisher Dampfkessel Revisions Verein* on two "Otto-Deutz" gas engines, of electric lighting type, about 160 H.P., the following results were noted:—

1st engine—Load 170 H.P. .	Consumption 575 grammes (say 1·25 lbs.)
2nd engine—Load 102·8 H.P. .	Consumption 600 grammes (say 1·32 lbs.)

The tests were carried out with peat briquettes having a calorific value of 9,180 B.Th.U. per lb.

Another trial, made with an 80 H.P. engine of the same manufacture, has given, for a load of 80 H.P., a consumption of 544 grammes (say 1·2 lbs.) of ligneous peat of 9,200 B.Th.U. per lb., costing, delivered on works, 9·4 shillings per ton (say 0·6 pence per H.P. hour).

The Gasmotoren Fabrik Deutz have given the utilisation of lignite a considerable amount of attention. Lignite contains a large quantity of water, generates a large proportion of carbon dioxide, and produces from 3 to 5 per cent. of tarry residue, containing about 63 per cent. of water, 17 per cent. of paraffin, and 20 per cent. of carbon and ash. There have been numerous difficulties to overcome, and the Gasmotoren Fabrik Deutz appear to have been successful. Their apparatus gives a practical efficiency of 88 per cent. and the purification is such as to ensure continuous operation for several months without cleaning the pipe connections or the engine cylinder.

The Gasmotoren Fabrik Ehrenfeld have also constructed a generator for burning lignite, and the following particulars give the result of a test made upon a 40 H.P. engine served by a generator fed with lignite briquettes:—

1. Fuel composition—

Water	14·1	per cent.
Ash	5·3	,,
Carbon	52·3	,,
Hydrogen	4·1	,,
Oxygen and Nitrogen	2·42	,,

Lower calorific value of fuel 8,100 B.Th.U. per lb.

2. Fuel consumption per B.H.P. hour, 800 grammes (say $1\frac{3}{4}$ lbs.).

3. Calorific value of gas at 32° F., and 29·9 inches of mercury = 118 B.Th.U. per cubic foot.

Liquid Fuels.—For central electric generating stations, steam power plants constitute a serious rival to gas engines, on account of their great reliability. Spirit engines have been adopted for vehicular traffic, and, owing to the marvellous qualities of these small engines, motoring and aeronautics have opened up new fields for the application of internal combustion engines in which steam power is entirely unsuitable.

The "Diesel" and "Trinkler" (Koerting) engines, with high compression pressures designed for utilising heavy or crude oils, have also brought a satisfactory solution in cases where it seemed steam power could never be displaced.

Benzol.—Amongst those products which are of a suitable nature for utilisation in internal combustion engines, benzol should be mentioned. Ninety per cent. benzol is a colourless liquid with an empyreumatic odour (that is to say, with a smell as of a slightly burnt substance) very fluid and perfectly neutral, composed exclusively of carbon and hydrogen with a slight trace of sulphur. The hydrocarbons belong to the aromatic series for the most part of the type C_6H_6. Its density at 15° C. (59° F.) is from 0·880 to 0·881.

Benzol distils between 80 to 82° C. (175 to 180° F.), and 118 to 120° C. (245 to 248° F.) at the normal pressure of 760 mm. (29·9 inches) of mercury. Its vapour tension at 59° F., is 2·04 inches of mercury. Freezing commences—appearance of the first crystal—at 20° F. The crystals form a greasy mass of laminated, square cornered pyramids. Its flash point is lower than 32° F.

Its ascensional power by capillary action is, other things being equal, at 59° F., 3½ times that of petrol and 2½ times that of ethyl alcohol. It burns completely away without residue, and, when properly mixed with air, gives a very hot blue flame which is peculiarly steady.

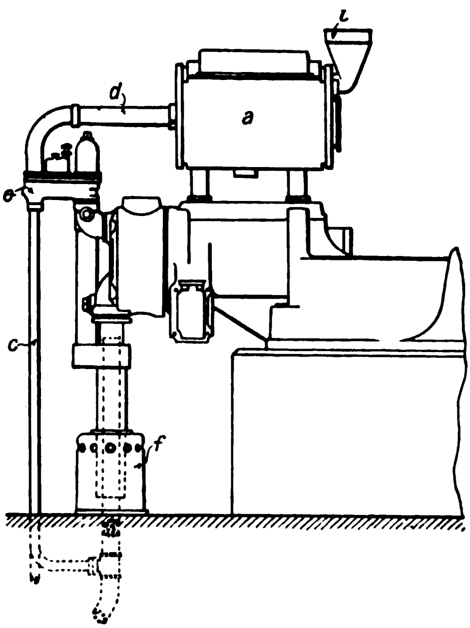


FIG. I.—1. Gasmotoren Fabrik Deutz Naphthaline Engine.

Benzol is not affected by dull red heat, but becomes decomposed beyond 930 to 1100° F. The molecular weight is 78 grammes. Its combustion in air, at constant normal pressure evolves 766,000 small calories or 766 large calories (for 78 grammes) equivalent to 9,960 calories per kilogramme, or 17,920 B.Th.U. per lb. gross. When the water produced by complete combustion is not condensed, the heat given out is 9,588·5 calories per kilogramme, say 17,250 B.Th.U. per lb.

Naphthaline.—The Gasmotoren Fabrik Deutz has been able to utilise naphthaline in engines. This substance is a solid by-product of the distillation of coal, becoming liquefied at 173° F., and costing in Germany only 0·4 pence per lb.

The Gasmotoren Fabrik Deutz naphthaline engine comprises a reservoir for naphthaline briquettes placed above the cylinder, around which the hot water passing from the cylinder is made to boil within an outer jacket or envelope. Water is fed into the funnel *i* (Fig. I.—1), and the vapour escapes from the double envelope of the reservoir *a* by the pipe *d*, and, after circulating round the carburettor *e*, passes away to the exhaust pipe by the pipe *c*. The melted naphthaline is conveyed to the carburettor *e* by a copper tube disposed within the tube *d*. The air required for combustion is drawn through the casing *f*, which encircles the exhaust pipe, and the heat thus furnished prevents the crystallisation of the naphthaline during the formation of the mixture in the carburettor.

The engine is first set to work with benzol or benzine, by means of a special carburettor, and at the end of half an hour the water in circulation becomes sufficiently heated to melt the naphthaline, and thereupon the engine is ready to work with the latter product. The consumption of naphthaline has been found to equal 0·66 lbs. per B.H.P. in a 10 H.P. engine.

Blast Furnace Gas.—The idea of utilising blast furnace gas to furnish power by means of gas engines should, apparently, be attributed to W. Lurmann, who propounded it in 1887, but its application to engines of large power has only occurred during the last seven or eight years.

Twelve or thirteen years ago, the first experiments were conducted simultaneously in Germany, Belgium, and England, and thus opened a wide field for the application of large engines. Although these early trials were made with only small engines, the technical information that the experimenters derived soon encouraged them to make further progress.

The *Société Cockerill* in Belgium, built a 200 H.P. single-acting, four-cycle engine, and for more than ten years this machine has been in regular operation in their works.

A further step was successfully made by the construction of a 600 H.P. engine on the Delamarre-Debouteville system by the Cockerill Co., and exhibited by them at Paris Universal Exhibition in 1900. This large engine was single-acting, and had a cylinder 1,300 mm. (51 inches) diameter and stroke of 1,400 mm. (55 inches). It was designed to give its power at 80 revolutions per minute, and, with an initial explosive pressure of about 325 lbs. per square inch, gave a total pressure of over 300 tons at the back of the piston.

Soon after several firms of repute commenced to build large engines destined to utilise the gases set free by the diverse reactions set up during the manufacture of iron, coke, &c., and the metallurgical industry, generally, was not slow to take advantage of this path to further progress, replacing steam engines and boilers by powerful groups of gas engines as opportunity offered.

A double-acting, four-cycle engine was built by Koertings' in 1898.[1] This engine worked for several years. Koerting Brothers, however, did not follow up this type of engine, for meanwhile, they had designed and constructed their two-cycle motor, which they patented and exhibited at the Düsseldorf Exhibition. Following the success obtained in 1902 by the firms Cockerill, Deutz, and Nürnberg, with their double-acting, four-cycle engines, Koerting Brothers again took in hand the construction of their engine of this class making extensive modifications to the original design.

It has been stated recently that out of fifty German ironworks, forty-eight have already given orders for blast furnace or coke oven gas engines. These orders represent nearly 360 units, aggregating about 420,000 H.P. The largest installation consists of 35,000 H.P., and about fifteen central stations in ironworks have from 10,000 to 12,000 H.P. In some instances gas producers have been installed, so that, in case of need, the engines can be kept at work when the blast furnaces are inoperative.

In collieries, and for coke oven purposes, the competition between gas and steam power has not been so keen, owing to the fact that the waste gases from the old style of ovens used in a great many works can only be utilised for firing steam boilers. In these installations, however, the number of gas engines at work, or in process of erection, amounts to thirty or thirty-five, equal to 45,000 to 50,000 H.P.

The greater number of gas engines used in blast furnaces and collieries are of the double-acting type, some two-cycle and others four-cycle. The latter are more usually adopted on account of their high efficiency.

An ordinary blast furnace producing 100 tons of iron in twenty-four hours gives off about 140,000 to 160,000 cubic feet of gas per ton of iron of 90 to 100 B.Th.U. per cubic foot, being equivalent to 560,000 to 640,000 cubic feet per hour. The heating of the blast absorbs about 45 per cent. of this quantity. Therefore, there remains for gas production about 77,000 to 88,000 cubic feet per ton of iron, or, say, 300,000 to 350,000 cubic feet per hour. This volume of gas in a

[1] The conception of a double-acting gas engine appears to be due to Griffin who took out an English patent, No. 4,080, of 23rd August, 1883, for a six-cycle motor, of which several examples are still at work.

In 1893, Dick Kerr & Co., Ltd., built a 90 H.P., two-cylinder, double-acting tandem gas engine of 12½ inches diameter, 20 inches stroke at 180 revolutions per minute.

M. Letombe built his double-acting engine in the year 1893.

steam power plant represents from 1,200 to 1,500 H.P. With gas engines, from 3,500 to 4,000 H.P. can be obtained from the same quantity, a difference of something like 2,400 H.P. The mechanical devices required to serve the blast furnaces absorb about one-fourth of this power. There remains, therefore, from 2,600 to 3,000 H.P. available for disposal.

These figures are sufficient to show why increasing favour is given to large gas engines in metallurgical works.

The following statements have been set forth by the Cockerill Co. to show the relation between the metallurgical apparatus and the disposable power that can be obtained by the use of gas engines.

Coke Ovens.—The disposable power in B.H.P. hours by gas engines is equal to a figure representing the production of coke in tons per week.

Blast Furnaces.—The disposable power in B.H.P. hours by gas engines (the blowing engines being operated by gas) is equal to the figure representing the production of iron in tons per month.

Such results, however, can only be obtained when the installation includes all the improvements of modern practice, amongst which the most important relate to the systems of washing the gas. This operation has latterly been the object of attention of large firms of gas engine builders as well as of those engaged in the production of iron and steel.

To separate the gas from the different impurities that it contains, such as dust, tar, and the chemical constituents that are prejudicial to the good working of the engine, and in order to reduce the temperature of the gas before its admission to the cylinder, very complete washing, purification, and cooling is necessary. These operations are carried out by means of fans, rotary washers, and similar apparatus which involve a consumption of water varying from 90 to 125 gallons per 1,000 cubic feet.

The amount of gritty dust is reduced by this method from 3 to 0·2 grammes, or about one-fifteenth of the original figure. The power absorbed by the fans and washing machinery depends upon the system employed as well as upon the quantity of impurities that have to be eliminated, and varies from 0·07 to 0·227 B.H.P. per 1,000 cubic feet, being less than 2 per cent. of the power recovered in the utilisation of gas.

Some very interesting information regarding the amount of heat capable of utilisation in steam engines and in gas engines for the steel industry have been derived from practice by the Maschinenfabrik Augsburg Nürnberg, and are reproduced below.

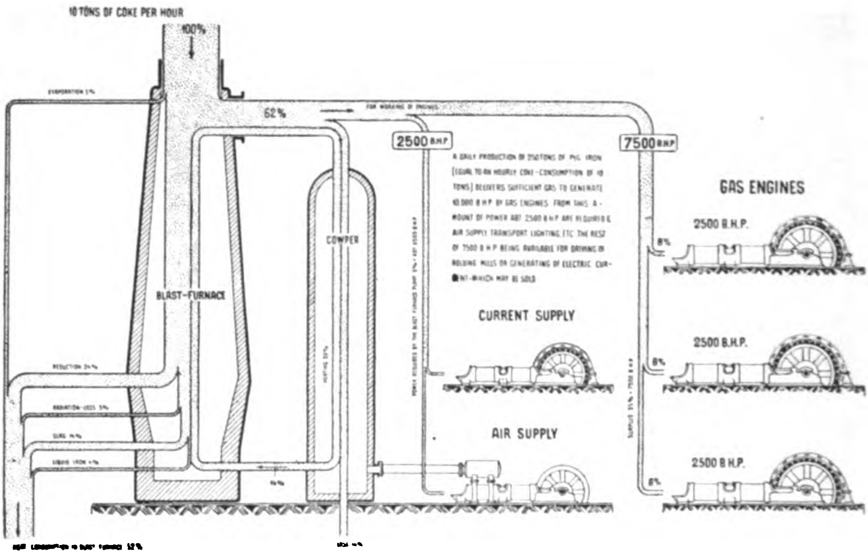


FIG. I.—2. Thermal Balance Sheet of Blast Furnace of 250 tons per day capacity.

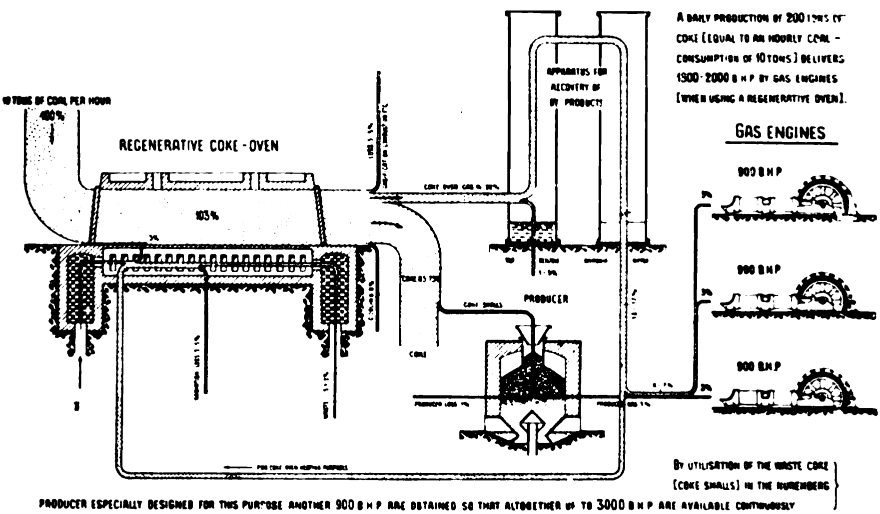


FIG. I.—3. Thermal Balance Sheet of Coke Oven of 200 tons per day capacity.

Figs. I.—2 and I.—3 represent a thermal balance sheet of a 250-ton blast furnace and of a coke oven producing 200 tons per day.

Fig. I.—2 shows that with gas from a blast furnace of this capacity, allowing a coke consumption equal to 10 tons per hour, gas engines of a total of 10,000 H.P. could be permanently served. The blast

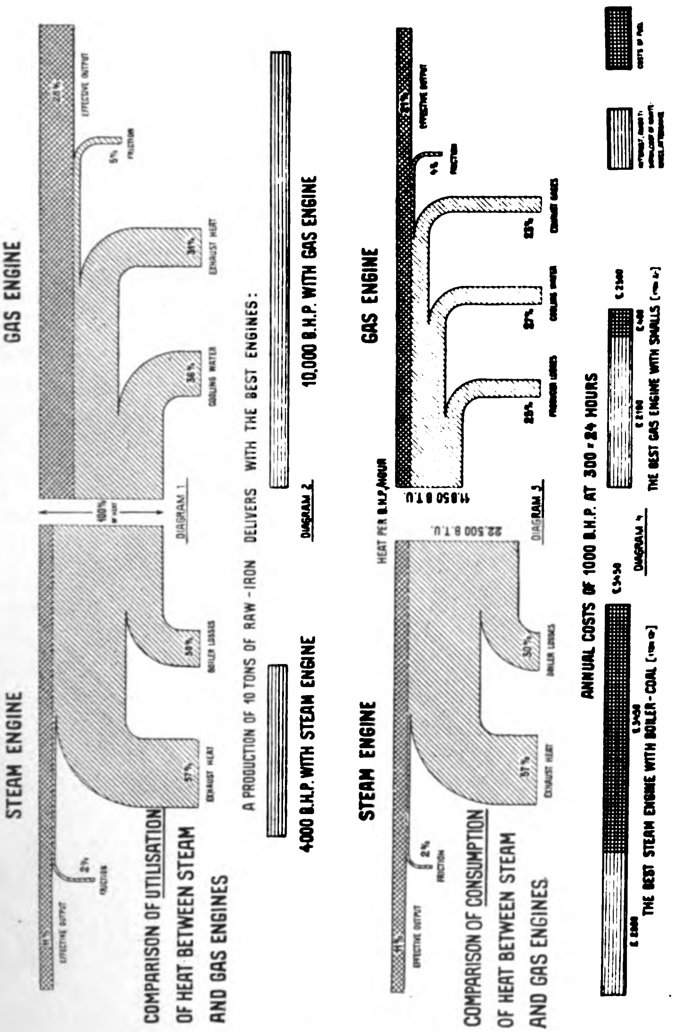


Fig. I.—4. Comparative Thermal Balance Sheets of Steam and Gas Engines.

furnace itself requiring about 2,500 H.P. for blowing, management, feeding, lighting, &c., 7,500 H.P. remains for the rolling mills, or for generation of electricity for other purposes.

With the gas from coke ovens, a production of 200 tons per twenty-four hours, consuming 10 tons of coal per hour, and with regenerators,

Fig. I.—3 shows that in permanent work, gas engines of from 1,800 to 2,000 H.P. could be served according to the quality of the fuel.

When the waste coke is gasified in a lean coal generator of the type made by the Maschinenfabrik Augsburg Nürnberg, it is possible to produce a further 900 to 1,000 H.P. by means of gas engines, so that, in this way about 3,000 H.P. are continuously available.

The undoubted economic superiority of the gas engine as compared with the steam engine is clearly shown from a comparison of the figures given in Fig. I.—4.

The construction of large gas engines has been greatly developed during recent years, and especially since their manufacture has been undertaken in the United States of America, upon lines inspired by German practice.

The table here given, which includes all the largest gas engines throughout the entire world, shows the relative output of these engines from builders in various countries. It only mentions those engines of more than 1,000 H.P., and has several gaps, owing to some firms not having furnished the particulars asked for.

Blast furnace gas is used for the largest number of these engines, and for a lesser number, coke oven gas. Natural gas, oil gas, and illuminating gas are employed only in some parts of the United States.

The general classification adopted by grouping the engines under the headings of the type of construction: Deutz, Koerting, Nürnberg, Oechelhauser, Cockerill, &c., is no longer rational, because a certain number of firms in Germany and elsewhere, building this or that engine under licence, have gradually proceeded, guided by the experience acquired in their own shops, to a more or less complete transformation of the original design, so much so, that, as a consequence, the characteristic features often disappear.

From the point of view of the relative number and aggregate power, Germany secures the first place with a considerable lead. However, a formidable impetus has been given to the construction of large gas engines in the United States, and it seems likely that Germany will soon be overtaken.

The backwardness of the English industry in this branch is indicated in the table.

TABLE GIVING THE WORLD'S OUT-PUT OF GAS

MAKER.	Number.	Single or Double-acting.	Total Output.	Blast Furnace Gas.
			B.H.P.	B.H.P.
Aschersiebener Machinenbau-A.-G. vorm. W. Schmidt & Co.	12	Double-acting, two cycle ...	17,600	16,500
Deutsch-Luxemburgische Bergwerks-u. Hütten-Aktien-Gesellschaft, Abt.	11	Double-acting, four cycle ...	14,100	12,100
Friedrich Wilhelms-Hütte, Mülheim a. Ruhr. Donnersmarckhütte, Labrze O.-S.	2	Double-acting, two cycle ...	2,000	2,000
Ehrhardt & Sehmer, G.m.b.H., Schleifmühle ...	39	Double-acting, four cycle ...	59,300	45,350
Elsässische Maschinenbau-Gesellschaft Mülhausen ...	29	Ditto ditto ...	38,000	36,000
Gasmotorenfabrik Deutz, Cologne-Deutz ...	15	Five single and ten double-acting, four cycle	20,800	18,000
Gutehoffnungshütte, Oberhausen ...	27	Double-acting, two cycle ...	24,400	21,400
Haniel & Lueg, Düsseldorf-Grafenberg ...	32	Double-acting, four cycle . .	45,000	38,500
Märkische Maschinenbau-Anstalt Ludwig Stuckenholz, Wetter a. Ruhr	8	Three single and five double-acting, four cycle	10,600	—
Machinenbau-A.-G. vorm. Gebr. Klein Dahlbruch ...	24	Double-acting, two cycle ..	38,999	38,999
Machinenbau-A.-G. vorm. Gebr. Klein, Filialwerk in Riga	4	Ditto ditto ...	5,326	5,326
Pokorny & Wittekind, Frankfurt a. M.-Bockenheim ...	4	Ditto ditto ...	2,200	1,000
Schüchtermann & Kremer, Dortmund ...	3	Double-acting, four cycle ...	3,600	3,600
Siegener Maschinenbau-A.-G. vorm A. und H Oechelhaeuser	29	Double acting, two cycle ...	41,410	41,410
A. Thyssen, Mülheim ...	43	Double-acting, four cycle ...	86,000	80,400
Maschinenfabrik Augsburg Nürnberg A.-G ...	129	Ditto ditto ...	203,870	161,570
Skodawerke A.-G. Pilsen ...	10	Double-acting, four cycle ...	13,650	13,650
Société Anonyme John Cockerill, Seraing (Belgium) ...	33	Two single and thirty-one double-acting, four cycle	42,164	40,164
Schneider & Cie., Creuzot (France) ...	9	One single, eight double-acting, four cycle	16,800	16,800
British Westinghouse Co., Manchester ...	1	Single-acting, four cycle ...	1,000	—
The Lilleshall Co., Ltd., Oakengates, Salop ...	2	Double-acting, four cycle ...	2,100	2,100
Mather & Platt, Manchester ...	2	—	2,400	—
Richardson, Westgarth & Co., Middlesborough (England)	6	Double-acting, four cycle ..	6,900	4,700
Allis Chalmers Co., Milwaukee, Wis. ...	46	Double-acting, four cycle ...	128,230	121,930
Snow Steam Pump Co., Buffalo, N.Y. ...	60	Ditto ditto ...	111,260	35,720
William Tod Co., Youngstown, O. ...	5	Ditto ditto ...	16,000	16,000
De la Vergne Machine Co., New York ...	26	Double-acting, two cycle ...	42,200	40,000
[illegible]ghouse Machine Co., Pittsburg ...	17	Double-acting, four cycle ...	39,800	33,000

ENGINES OVER 1,000 H.P. TO AUGUST 15, 1908.

GAS.				FOR DRIVING.				Other Purposes.	DELIVERED FOR.						
Gas	Producer Gas.	Town Gas.	Natural Gas.	Dynamos.	Blowing Engines.	Rolling Mills.	Trans-mission.		Belgium.	Germany.	England.	France.	North America.	Austria.	Other Countries.
.P.	B.H.P.	B.H.P.	B.H.P.	B.H.P.	B.H.P.	B.H.P.	B.H.P.	B.H.P.	B.H.P.	B.H.P.	B.H.P.	B.H.P.	B.H.P.	B.H.P.	B.H.P.
-	1,100	—	—	12,800	3,500	1,300	—	—	1,100	15,200	—	—	—	—	1,300
00	—	—	—	11,700	2,400	—	—	—	—	10,100	—	—	—	—	4,000
.	—	—	—	—	2,000	—	—	—	—	2,000	—	—	—	—	—
00	1,350	—	—	48,650	10,650	—	—	—	4,050	41,300	1,350	7,950	—	4,650	—
00	1,000	—	—	17,800	20,200	—	—	—	—	24,200	—	7,100	—	—	6,700
00	—	—	—	17,000	3,800	—	—	—	—	11,600	—	—	—	7,200	2,000
00	—	—	—	8,900	15,500	—	—	—	—	17,600	5,400	—	—	—	1,400
00	—	—	—	32,000	13,000	—	—	—	—	45,000	—	—	—	—	—
	—	—	—	3,000	4,600	3,000	—	—	—	10,600	—	—	—	—	—
	—	—	—	2,028	29,510	7,461	—	—	—	23,348	4,436	—	-	—	11,215
	—	—	—	—	4,104	1,222	—	—	—	—	—	—	—	—	5,326
	1,200	—	—	600	1,000	—	—	600	—	1,000	—	—	—	—	1,200
	—	—	—	3,600	—	—	—	—	—	3,600	—	—	—	—	—
	—	—	—	1,320	38,030	—	—	2,060	—	38,610	—	—	—	—	2,800
0	—	—	—	38,300	47,700	—	—	—	10,500	75,500	—	—	—	—	—
0	13,200	—	—	128,870	72,300	1,500	1,200	—	2,400	160,570	3,600	17,200	—	—	20,100
	—	—	—	7,500	3,750	2,400	—	—	—	—	—	—	—	13,650	—
0	—	—	—	—	—	—	—	—	28,664	1,200	—	6,000	—	—	6,300
	—	—	—	15,600	1,200	—	—	—	—	—	—	16,800	—	—	—
	1,000	—	—	1,000	—	—	—	—	—	—	1,000	—	—	—	—
	—	—	—	—	2,100	—	—	—	—	—	2,100	—	—	—	—
	2,400	—	—	1,000	—	—	—	1,400	—	—	2,400	—	—	—	—
	2,200	—	—	4,700	2,200	—	—	—	—	—	4,700	—	—	—	2,200
	6,300	—	—	98,230	30,000	—	—	—	—	—	—	—	128,230	—	—
	4,400	24,600	46,540	47,720	17,000	—	—	46,540	—	—	—	—	111,260	—	—
	—	—	—	1,700	14,300	—	—	—	—	—	—	—	16,000	—	—
	1,000	—	1,200	8,000	32,000	—	2,200	—	—	—	—	—	42,200	—	—
	1,000	—	2,200	7,600	32,200	—	—	—	—	—	—	—	39,800	—	—

CHAPTER II

GAS *VERSUS* STEAM ENGINES

GAS engines have still to conquer the vast field of manufacturing industries. It is in this particular domain that the contest between the gas engine and the steam engine will take place. During the course of long years, the steam engine has been endowed with valuable improvements both in design and workmanship, and, up to the present, has retained its preponderance in factories requiring motive power to the extent of between 200 and 1,000 H.P.

In the course of frequent visits to Switzerland, Germany, and America, the author has visited the principal firms who to-day specialise in the construction of large gas engines, and he has been struck by the paucity in numbers of gas engines of over 100 H.P. to be served by producer gas, although numerous gas engines of from 1,000 to 2,000 H.P. were everywhere under construction, destined to work with blast furnace or coke oven gas.

This state of things is certainly due less to the apprehensions that are aroused by gas engines and producers of 100 to 500 and 1,000 H.P. in the industrial field, than to the ever-increasing difficulty that is found throughout Europe in procuring poor coals at a low price suitable for gas production by "suction" plants.

The physical laws which govern the generation and application of steam as a motive force have long been formulated. They were soon disengaged from the obscurity which enveloped their interpretation and the science of thermo-dynamics has given them a definite sanction in many forms of application.

Improvements in construction have marched step by step with the progress of scientific theory to accomplish mechanical marvels. But steam is a fluid of a much less complicated nature than explosive mixtures. The steam during work obeys precise laws appertaining solely to physics, whilst the production of combustible gases and the manner in which they exist in the form of explosive mixtures in gas engines, apply as much to chemistry as to physics and mechanics.

From an economical point of view the special properties of the working fluids bring about advantageous or disadvantageous consequences according to the type of engine used. It is therefore impossible to speak of absolute superiority or inferiority between steam and gas engines. The one outweighs the other according to circumstances.

In a paper read before the Institution of Electrical Engineers in January, 1907, by Mr. C. E. Douglas, the choice between the two engines was considered in relation to the economy obtained by each system at different loads.

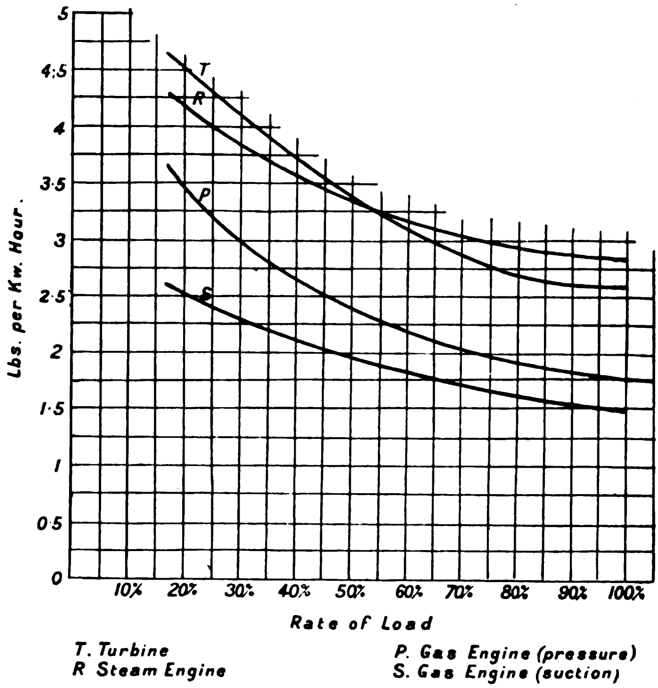


FIG. II.—1. Relative Fuel Consumption Curves of Steam and Gas Engines at various out-puts.

The diagram (Fig. II.—1), in which the ordinates represent consumption and the abscissæ the loads expressed as a percentage of the maximum power in kilowatts, shows clearly by the comparative curves for different loads, the corresponding consumption per kilowatt hour for turbines, reciprocating steam engines, "pressure" producer gas engines, and "suction" producer gas engines.

It will be noticed that at half load the turbine and the reciprocating steam engine have practically the same efficiency. The consumption of the turbine diminishing to a far greater extent than that of the reciprocating engine under increase of load, while, when the load is lessened, the reciprocating engine is more economical than the turbine.

The consumption of producer gas plants, "suction" and "pressure," remain practically in parallel relation throughout, except that towards low loads the "pressure" plant becomes more inefficient.

Apart from motive power derived from wind and water, the remaining sources may be classed in one or other of the following categories:—

1. Reciprocating stationary steam engines and boilers.
2. Steam turbines and boilers.
3. Semi-portable steam engines.
4. Internal combustion engines.

One of the principal considerations to take into account in respect of these installations is that relating to the cost of fuel. For installations of large size in districts where the fuel, delivered on site, is expensive, this consideration outweighs all others.

In many cases, however, perfect reliability is the most important question, and, at the present time, a decision upon this point must be given in favour of the steam engine, unless a sufficient number of gas engines are installed to act as reserve in case of failure.

Stationary Steam Engines and Turbines.—In these two categories the engines fed by the boilers are placed some distance apart. In installations of considerable size, a distance of about 40 feet usually intervenes. This, naturally involves loss by condensation and radiation which sensibly affects the efficiency of the combined plant.

For stationary steam engines of not less than 500 H.P., many makers are prepared to guarantee a steam consumption not exceeding 8 to 11 lbs. per I.H.P. The overall efficiency of such installations, judging from the results of the most favourable tests made by the author, or from those carried out by others, has never yet been less than 1·76 lbs. of coal per B.H.P. hour.

Turbines of more than 1,000 H.P. consume about 12 lbs. of steam per B.H.P. hour. From tests made in 1905 at the central electric generating station at Liège, with the turbine supplied by the John Cockerill Co., a consumption of 15·2 lbs. of superheated steam per kilowatt hour was noted, equivalent to 9·95 lbs. per B.H.P. hour.

Semi-Portable Steam Engines.—Semi-portable engines of high efficiency and low consumption are generally constructed of the compound condensing type, with superheaters, from 30 to 400 and

500 H.P. The consumption of such engines is about 1 lb. of coal (gross) per B.H.P. hour; 600 H.P. seems to be the practical limit for installations of this class, owing to the difficulty of indefinitely augmenting the dimensions of the boiler and its steam generating capacity. However, German makers have built semi-portable engines capable of developing about 1,000 H.P.

As regards fuel consumption, good progress has recently been realised. The following results have been sent to the author of a trial made in 1908 by Prof. Guthermuth, on a Wolf semi-portable engine:—

Length of test	7 hours.
Temperature of superheated steam before admission to high pressure cylinder	626° F.
Power developed	105 B.H.P.
Mechanical efficiency of engine	92·5 per cent.
Consumption of coal	1·04 lbs. per B.H.P.
Consumption of steam	8·7 lbs. per B.H.P.

Gas Engines.—The competition between gas engines and steam engines on the question of economy depends mainly upon the difference in the price of fuel. It being, of course, understood that an improved steam engine is compared to an installation of gas engine and producer, each of the best design and workmanship in respect of regularity and reliability of operation.

The majority of gas engine and producer makers of high repute now guarantee a consumption as low as 0·86 lb. of coal per B.H.P. hour, for work varying between 80 and 100 per cent. of the normal power, in installations of from 50 to 200 H.P. capacity.

The larger gas power installations are generally fed by "pressure" producers by means of a fan or boiler. The cost of fuel used in the latter must be taken into account in reckoning that of the generator, and generally amounts to from 15 to 20 per cent. of the fuel burnt in the producer.

The figures quoted for engines of 50 to 200 H.P., refer to the fuel consumption noted during full load tests under the most favourable conditions. In practice, these would be increased by 10 to 20 per cent., because of the losses incurred due to variations of load, firing of producer, &c.

Under similarly favourable conditions, a compound condensing steam engine fed with superheated steam requires at least 1·75 lbs. of semi-bituminous coal per H.P. hour.

Lean coals, containing from 6 to 8 per cent. of volatile matter, and 5 to 7 per cent. of ash, and screened to about ½ to 1 inch pieces, as generally employed for "suction" plants, usually cost about double the price of steam coal which contains from 15 to 25 per cent. of volatiles, and 12 per cent. of ash, of about the same average calorific value as the lean coal, namely, 13,500 B.Th.U.'s per lb.

The fuel cost per H.P. hour, for "suction" gas producer and engine is therefore practically the same as that for the up-to-date steam plant, the latter requiring double the weight of fuel but at half the price.

Taking into consideration that the consumption of lubricating oil for the gas engine is very little more than that needed for the steam engine, and that the initial outlay is practically the same for one as for the other, it is not to be wondered at that the manufacturer gives preference to the steam power plant, with its undoubted reliability and great elasticity of output.

As things stand, the future of the gas power plant for powers exceeding 100 to 150 H.P., depends entirely upon the possibility of using ordinary coal at normal prices, or some other cheap fuel, such as lignite or peat.

The heavy liquid fuel engines of the Diesel type work with a consumption of about 0·44 lbs. per effective H.P. hour.

Blast Furnace Gas Engines.—In connection with blast furnace gas engines, it is interesting to note the figures published in 1907 by M. Leon Greiner in a pamphlet entitled "Production economique de la force motrice dans les Usines Metallurgiques," published by H. Le Souvier, 174, Boulevard Saint Germain, Paris.

The Cockerill Co. estimate that the average cost of a central station, consisting of blast furnace gas engines, gas mains, washing apparatus, dynamos, foundation and buildings is £16 per kilowatt. The sum provides for an annual allowance for depreciation at 13 per cent., of £2·08.

The cost of working during the first six months of 1906—7 was 0·0653 pence per kilowatt. The average number of working hours per year is estimated at 4,380, corresponding to a load factor of 50 per cent.

The total cost per kilowatt is therefore, including depreciation, 0·183 pence.

The Cockerill Co. find that the working costs per kilowatt hour during recent years has been as follows:—

Year.	Production.	Working cost. Per kw. hour.
1900—1 steam only	1,789,781 kw. hours	0·88 pence
1901—2 gas and steam	1,930,740 „ „	0·776 „
1902—3 „ „ „	2,007,290 „ „	0·599 „
1903—4 „ „ „	5,387,612 „ „	0·296 „
1904—5 „ „ „	9,999,216 „ „	0·206 „
1905—6 „ „ „	14,915,919 „ „	0·163 „
1906—7 gas only	20,000,000 „ „	0·0653 „

Regularity.—With respect to regularity, mention may be made of the installation of two-cycle gas engines by the Siegener Maschinenbau, A.G., for driving three-phase, alternating current dynamos. These engines work with a degree of uniformity of $\frac{1}{160}$ to $\frac{1}{180}$.

Messrs. Mather & Platt, of Manchester, have lately installed a twin two-cycle engine of 600 H.P. in a Lancashire cotton mill. This is the first successful installation of large power gas engines for such a purpose. The chart reproduced in Fig. II.—2, was taken by a Moscrop recorder during a day's run, and shows the great regularity of this particular engine.

The principal considerations of a general character that have to be taken into account when making a decision between gas and steam power will now be discussed, and for this purpose some particular applications will be passed under review, and certain examples mentioned of existing large gas power installations.

Rolling Mills.—Some experiments have been made with a view to the utilisation of gas power for driving rolling mills. Their partial failure appears to be principally due either to the insufficiency of the power provided for the work, or to the lack of knowledge with regard to every-day working conditions.

It is very necessary to allow for the variations that occur in the quality of the gas obtained from blast furnaces from time to time. This gas is generally used for rolling mill engines, and sometimes is of about 100 to 110 B.Th.U. per cubic foot, while occasionally it is only of about 85 to 90 B.Th.U. In the latter case, the degree of inflammability of the mixture is sensibly affected, and if the engine is not fitted with some device for counteracting these variations, the power produced is materially lessened. Gas of 85 B.Th.U. per cubic foot is practically the lowest limit with which it is possible to produce combustion, owing to the fact that the combustible elements (hydrogen and carbon monoxide) are so diluted with inert gases that the oxygen cannot combine with them.

If this lower limit of 85 B.Th.U. be frequently found to exist, it is necessary to arrange for some method of enrichment, as, for example, by means of a certain proportion of producer gas of about 135 B.Th.U. per cubic foot.

Another consideration, peculiar to rolling mill work, is that without warning, and for relatively short periods, a very large amount of power is demanded, while usually it is necessary to arrange for the engines to be reversible.

A rolling mill may be driven either by a direct coupled steam engine, or by a gas engine; or by a dynamo taking current from a central electric station in which it is generated either by steam or gas engines.

The direct coupled steam engine has the advantage of assuring thoroughly reliable operation, owing to its great elasticity and capacity for overload, amounting on occasions to 50 per cent. of the normal output. It can be reversed with great ease, but it involves a high fuel consumption.

When comparing a direct-acting steam engine with a central electric installation and steam engine driving rolling mills only, allowance must be made for the cost of running the engines and dynamos for intermittent idle periods of from 10 to 30 minutes' duration.

The direct coupled gas engine is very economical with regard to fuel consumption. It can be readily served, and without great cost, by a gas main coming from the blast furnaces. To ensure reliability, however, it should be about 100 per cent. larger than actually required to deal with the normal resistances. This results in the charges for capital and depreciation being, in some cases, excessive. Besides this, the ordinary type of engine does not readily lend itself to reversing. Rotating always in the same

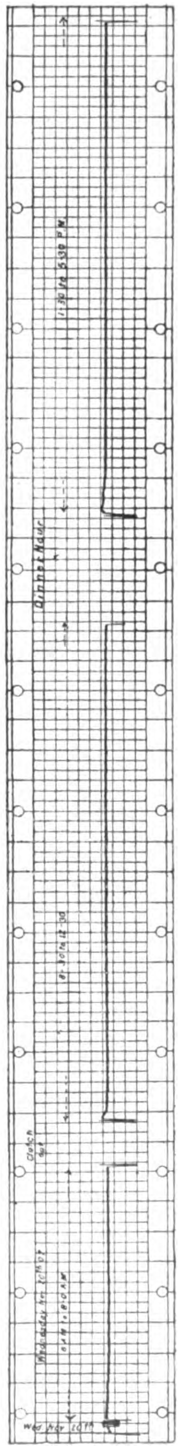


FIG II.—2. "Moscrop" speed variation record of 600 H.P. Mather & Platt two-cycle Gas Engine (Koerting type).

direction, it demands special mechanical appliances to be fitted to accomplish the reversing movements, which, in addition to causing further complication, involves loss in efficiency.

Indirect driving of rolling mills by a central electric power station, similarly involves a high capital outlay. If such a station serves rolling mills absorbing, for instance, 1,000 H.P. as an average, it will be found to be subjected to variations of load ranging between 700 and 1,500 H.P. or 100 per cent. Such conditions of working are therefore unsuitable. But if the central station supplies current for other purposes of a larger total capacity, and deals with a load of from 10,000 to 20,000 H.P., the load variation due to the rolling mill, will represent only 5 to 10 per cent. of the total power, and, in such a case, the cost of working per H.P. would be greatly decreased if blast furnace gas engines were installed to drive the dynamos.

Sundry Examples.—Under this heading authoritative figures will be given respecting several producer gas power installations known to the author.

The Pittsburg Plate Glass Co. has installed in its works at Crystal City (Missouri) an engine of 1,800 H.P. built by the *Allis Chalmers Co.*, and coupled to a 1,000 kw. alternating current dynamo. This engine is of the double-acting, twin-tandem type. The stroke is 42 inches and speed 107 revolutions per minute. It is served with "pressure" producer gas made from bituminous Illinois coal. The engine and its alternator work in parallel with other units. The alternator is fitted upon the main crank shaft between two sets of cylinders. The engine is part of an installation totalling 5,000 H.P. supplied by the same makers.

The Snow Steam Pump Co., of Buffalo, have installed 14 gas engines of 1,000 H.P., 2 of 4,000 H.P., and several of 500 H.P., for pumping natural gas from the shafts at low pressure, and delivering it into high pressure service mains. The same company has also supplied to the *Californian Gas and Electric Co.*, of San Francisco, for tramway service, 4 engines each of 5,400 H.P. direct coupled to 4,000 kw. dynamos. Three of these dynamos are 25 cycles, one is 60 cycles, and the voltage of the system is 5,500 volts. These engines are 4 cylinder double-acting, twin-tandem, four-cycle type and are illustrated in Fig. II.—3. Their principal dimensions are as follows:—

Diameter of piston—42 inches by 60 inches stroke.

Piston rods—15 inches diameter.

Speed—90 revolutions per minute.

The flywheel weighs 50 tons without reckoning the weight of the rotor.

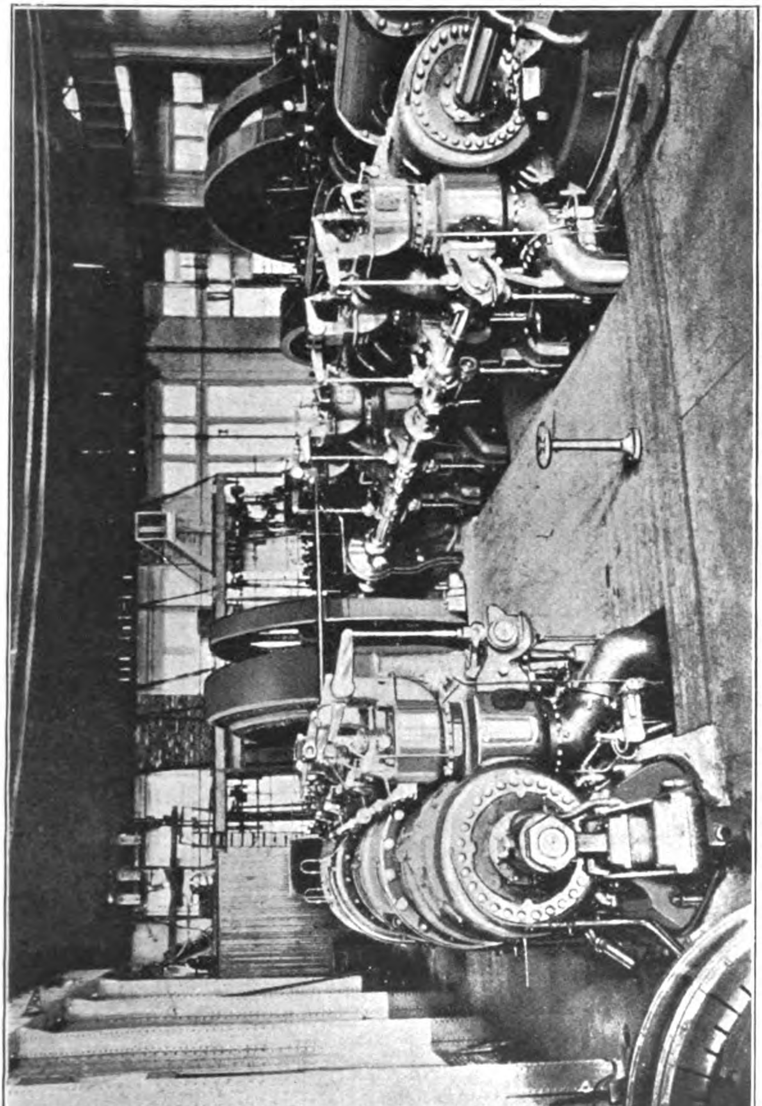


FIG. II.—3. Four-cylinder, double-acting, twin-tandem, four-cycle Gas Engine of 5,400 H.P., constructed by the Snow Steam Pump Co., of Buffalo, for Californian Gas and Electric Co. of San Francisco, direct coupled to 4,000 k.w. generator for tramway service.

The fuel used is oil gas of 600 B.Th.U. per cubic foot, containing as much as 60 per cent. hydrogen, produced by means of Lowe gas generators from crude petroleum of 20,000 B.Th.U. per lb. The petroleum is heated to about 300° F., and the gas produced is mixed with super-heated steam. The saturated mixture of petroleum vapour is fixed in the apparatus at a temperature of about 600° F., the average analysis being as follows:—

Hydrogen	H_2	58·4
Methane	CH_4	28·8
Hydrocarbons	CHn	3·4
Carbon monoxide	CO	4·4
Carbonic acid	CO_2	1·2
Nitrogen	N	3·8
Oxygen	O_2	Traces.

The *Western New York and Pennsylvania Traction Co.*, which owns about 85 miles of inter-urban lines in the vicinity of Olean (New York), several months ago gave proof of their confidence in gas engines by disposing of all the steam engines in their central station. The company now place entire reliance upon their central station at Ceres for the necessary power to work their trains. The station comprises two double-acting, tandem, Westinghouse engines, and one twin-tandem, double-acting, Snow engine, all of the same output, driving General Electric Co.'s alternating current dynamos, three phase, 300 kw., at 150 revolutions per minute.

The Snow engine has pistons 16 inches diameter by 30 inches stroke. The two cranks are at 90 degrees giving four impulses on the shaft per revolution. The distinguishing feature of these engines is that the combustion chambers are placed at the side of the cylinder with admission valves above, and exhaust valves below. The engines are fed with natural gas averaging 900 B.Th.U. per cubic foot, taken from shafts belonging to the company. The gas is fed into the main service pipes at a pressure of 200 lbs. per square inch, and passing through reducing valves the pressure falls successively to 50 lbs., and, finally, to 1·8 lbs., at which pressure it is admitted to the engines.

Days.	Period.	Average load.	Gas consumption.	B.Th.U. per B.H.P. hour.
	Hours.	Kw.	Per kw. hour cubic feet.	
Sunday	21	192·4	26·5	16,350
Monday	20	202·5	25·0	15,200
Tuesday	21	232·9	23·5	14,250
Wednesday . . .	19	248·7	22·5	13,500
Thursday . . .	18	280·5	22·0	13,250
Friday	20	256	22·5	13,500
Saturday . . .	19	272·6	22·0	15,300
Average	19·7	240·8	23·5	14,200

The table on p. 20 (from "Power and the Engineer") shows the efficiency of these engines in daily service over a period of one week with a very variable output, complete stoppages occurring between the hours of two and four in the morning when the engines are shut down.

The figures in the table are based upon an average calorific value of 900 B.Th.U. per cubic foot, and an efficiency of about 1·5 B.H.P. per kilowatt. The thermal efficiency per B.H.P. under such circumstances only amounts to about 18 per cent.

Messrs. Rowan & Co., of Glasgow, installed a gas power plant in their works in 1905, under the supervision of their consulting engineers, Messrs. Hal Williams & Bridges. In the first place, the total power absorbed by the existing machinery and required for new tools was precisely determined and found to be equivalent to about 600 H.P. One of the principal considerations was the great discrepancy between the maximum and minimum loads that prevailed, and particularly at night when reduced loads were to be dealt with, and also by the fact that several isolated workshops were to be supplied with power for overtime work.

It was decided to adopt gas engines in preference to steam power, and to sub-divide the 600 H.P. into four units of 150 H.P. each. It was arranged that the four engines should drive in parallel by belts on to one main shaft by friction clutches, and that three of them should each be belted to a direct current dynamo. The space available was much restricted and, in consequence, it was exceedingly difficult to arrange for the erection of new engines in place of the old without stopping the works. In order to avoid this, however, two engines were first erected and set to work, thus enabling the steam engine to be removed and the other gas engines to be put into position. As the headroom available was insufficient to accommodate an overhead crane, a special arrangement of suspended girders forming a runway had to be devised for use when it was necessary to remove the pistons for cleaning purposes, &c.

The engines were supplied by the *National Gas Engine Co.*, of Ashton-under-Lyne, and the producers by the *Dowson Economic Gas and Power Co.*, of London. The engines have cylinders of 22½ inches diameter by 30 inches stroke, and are speeded at 170 revolutions per minute. In order to obtain as slight a cyclical variation as possible with "hit and miss" governing, the fly-wheels were made specially heavy, each weighing about 12½ tons. Generally three engines work during the day, the fourth being set to work in the evening to generate current for electric lighting.

The four producer gas plants are placed in line and erected, each behind an engine. An equalising pipe is connected to each generator. Any engine is able to work with any of the generators, thus ensuring continuity of operation during the cleaning of any one of the generators during working hours. The four producers being connected to the same service main, the effect of the suction of the engine is equalised, and, in case of a breakdown to one of the sets, it can be isolated from the others.

Finally when one or two engines only are required for night work or for overtime, the four generators can be connected to the two engines. This considerably reduces the standby losses and facilitates starting to work next morning. In fact it is possible to start up the four engines without blowing up by means of the electric blowing fans. Each generator has its separate fan.

Under full load, the whole of the plant, producers, engines, and dynamos, are intercommunicated in parallel, so that, if an accident should happen to one of the units, the others would immediately take up the load and keep things going. Each unit is able to carry a heavy overload for long periods.

The guaranteed fuel consumption with Scotch anthracite was :—

At full load	1 lb. of coal per B.H.P. hour.
,, three-quarter load . .	1·125 ,, ,, ,, ,,
,, half load	1·25 ,, ,, ,, ,,

The quantity of cooling water for the engines was not to exceed five gallons per B.H.P. hour, and for the scrubbers and generators not more than one gallon per B.H.P. hour for both purposes.

Tests were carried out by the consulting engineers and the following figures obtained :—

	25th February, 1909.	*26th February*, 1909.
Average power developed by each engine, B.H.P. . . .	117·65	136
Average fuel consumption per B.H.P. hour, lbs. . . .	0·854	0·845
Cooling water consumption, gallons	4·2	3·64

The Scotch anthracite used was from the Barblues mines, and cost 18*s*. per ton at the time of the trials.

CHAPTER III

THE FUTURE OF GAS POWER

Throughout this work the author has endeavoured to give extensive consideration to the practical side of the subject of internal combustion engines, and it is for this reason, particularly, that in the appendices he has given complete lists of authors, engine builders, and manufacturers of auxiliary apparatus. The reader should, therefore, be in a position to obtain further and detailed information upon application to any of the specialists who are there mentioned.

He has entirely omitted all theoretical considerations, not because they are of little worth, but rather that experience has shown that, so far, theory has not greatly assisted the progress of the gas engine towards perfection. No theorist has discovered a cycle and a formula to satisfy all the extremely complex phenomena that the practical engineer has to accept, and that force him to exercise the more ingenuity in overcoming the obstacles that hinder further improvements. Fanciful generalisation and hastily-formed conclusions, have often resulted in misleading inventors instead of guiding them to the desired goal, and have been the cause of a great deal of money being lost, owing to the fact that gas engine theory and design, as far as present knowledge goes, is far removed from being an exact science. It should not be forgotten that Otto, who was a true inventor, and who first thought out the modern four-cycle gas engine, was absolutely ignorant of theory—he was not even an engineer.

It is worth while to summarise the improvements which must still be made to gas engines before they can claim to give the steadiness of operation and ease of control and maintenance that is so necessary, and which many factory owners have considered hitherto to be the exclusive monopoly of the steam engine. In accord with all other writers, the author claims that the gas engine is incontestably superior to the steam engine upon economical grounds, with regard to the quantity of fuel used to produce a certain amount of power. But low consumption is not the only question that the factory owner takes into consideration. He must, before everything else, protect himself from any risk of sudden breakdown, causing stoppage of industrial processes, which are always costly, and sometimes disastrous, as in the case of metallurgical works, electric light stations, refrigerating plants, ventilating appliances in connection with collieries, &c., &c., which cannot be allowed to stop even for a single moment.

The principal adverse criticisms to which gas engines are subjected are: (1) lack of overload capacity; (2) lack of regularity; and (3) the delicate nature of some of the mechanical devices which are indispensable for the operation of the engine. But the author would like to point out that the steam engine, when first brought out, presented similar weaknesses. More than a century and a half has elapsed since the first application of the steam engine, and, during that time, naturally, it has acquired a high degree of perfection, due to the experience gained during so many years of working. On the other hand, the first serious attempt to develop the gas engine dates only from the year 1860, the first industrial application dates from 1878, whilst the blast furnace and "suction" producer gas engines have only been on the market since the year 1895.

Lack of Overload Capacity.—It is frequently forgotten that a steam engine is never sold for the maximum power that it is capable of developing. For instance, a 100 H.P. steam engine is able to generate this power with an admission of steam during one-fifth or one-sixth of the piston stroke, a condition which corresponds to sufficiently economical working. As a matter of fact, however, by the increase in the duration of the admission period, or by supplying steam at a higher pressure, the power is also increased and as much as 150 H.P., or 50 per cent. more than the rated power, is within the bounds of possibility. It is clear, therefore, that the steam engine is essentially elastic with regard to output.

With regard to the gas engine, however, the custom is to sell it on the basis of the maximum power that it is capable of giving out. Sometimes, a margin of 5 to 10 per cent. is allowed to provide for contingent irregularity, either of mechanical adjustments or the working fluid. In such a case, there is evidently no available capacity for overloads. Nevertheless, there is nothing to prevent the choice of a gas engine powerful enough for it to be comparable to the steam engine.

The criticism relating to lack of overload capacity, therefore, should not be attributed as a fault of the gas engine itself; it rests with the puchaser to decide and to choose the power of the machine he may require to suit his peculiar circumstances.

Lack of Regularity.—The single-acting, four-cycle gas engine with only one impulse per cycle, or two revolutions, is evidently less regular in action than a steam engine in which an impulse is given for each half-revolution, but nobody can ignore the fact that any desired regularity can be obtained by means of a sufficiently heavy fly-wheel, and a sufficiently sensible system of governing. In this case, therefore, any lack of regularity should not be looked upon as a gas engine defect. It is only necessary to construct the engines adequately for the work that they are required to do. In the case of a steam engine a very great number of different types of engines are available for special duties. With gas engines, the number of types from which a selection can be made amounts to three only. The vertical engine, at a moderate speed, for electric lighting, &c., the high speed vertical engine for automobiles, launches, &c., and the slow-running horizonal engine for general industrial purposes.

Mention is made elsewhere of the many examples of two-cycle gas engines, which, as far as regularity is concerned, are at least comparable with, if not superior to, the best type of steam engine.

With regard to the criticisms relative to the delicate nature of some of the working parts, the gas engine, as also the steam engine, is subject to mechanical disorders, such as the seizure of badly lubricated portions, or breakage of some part or other, the dimensions of which have been incorrectly calculated, or which have been submitted to enormous strains. The gas engine, however, does include one particular detail that is not found in a steam engine, and that is the ignition device. In a later chapter, the author has dealt with this matter, and has mentioned the improvements which have been made to assure constant and regular operation of the ignition mechanism.

Low Grade Gas.—Difficulties of a different nature have been referred to in connection with engines using poor gas, due to the sooting up of parts, and particularly the inlet devices by imperfectly cleaned gas.

In large installations, where stoppages must be avoided at all costs, special means of washing and purification by centrifugal and other types of apparatus are installed. But, although by such means the amount of dust contained in the gas can be reduced to about 6 grammes per 1,000 cubic feet, there still remains a certain amount of tarry constituents which involve the greatest risk of interruption to the working of gas engines. This tar adheres to the walls of the supply-pipes, causes the valves and piston rings to stick, and may also become deposited in the cylinder, where it tends to cause premature firing. Some kinds of coke oven gas contain as much as 30 grammes of tarry vapours per 1,000 cubic feet.

The same tar trouble makes it obligatory to exclusively use lean coals for suction gas engines, so much in vogue at the present time, on account of their simplicity and economical operation.

Tar, being a direct distillation of volatile hydro-carbons which has escaped combustion and conversion into combustible gas, should properly be reduced within the generator itself. Various systems have been proposed, and, amongst others, up-draft producers, which burn the volatile matters as soon as given off and also distil the hydro-carbons in a separate portion of the apparatus, have been devised. So far the efforts of the inventors have not been attended with entire success. The caking of the fuel and the formation of cores, which upset the regularity of combustion, constitute the principal difficulties that must be overcome. The use of bituminous coal, therefore, has not yet been industrially solved in connection with suction gas producers.

The majority of producer plants designed up to the present, and particularly suction gas producers, have been brought out by manufacturers or mechanical engineers. These firms apparently have not specially considered the reactions which are produced during the process of gas manufacture. It is, however, possible that if the question were to be studied carefully by chemists, as is the case with blast furnaces, it would be possible to mix the combustible mixtures with some flux, so as to obtain, even with ordinary coal, a low grade gas of normal composition, free from tar, and from which the dusty residuals, without clinker, could be easily eliminated.

Gas engines should be particularly improved with regard to the ease of access and maintenance of working parts, such as the valves. They ought also to be made less sensitive to variations in the quality of the gas. The majority of applications that have been carried out are yet too recent to be of any real service in suggesting still further improvements. Nevertheless, in the writer's opinion several leading principles can now be formulated with regard to which it seems desirable that engine builders should undertake further researches.

1. To produce a system of governing of practically constant ratio of mixture admitted in variable quantity without involving a vacuum in the cylinder.

2. Increased compressions facilitated by water injection, for example, in such a way as to permit the utilisation of the poorest possible mixtures to burn slowly, but completely, at the commencement of the expansion period. In this way the mean pressure would be increased, permitting the reduction of cylinder dimensions and avoiding sudden explosions, which would be replaced by combustion

at nearly constant pressure during a certain period of the power stroke.

3. Automatic expulsion and complete scavenging of burnt gas, in order that the quality of the following charge of explosive mixture should not be unfavourably influenced.

4. Design of parts to allow free expansion of the portions in contact with hot gas, so as to permit engines to be worked at higher temperature and to increase the thermal efficiency by recovering a portion of the heat lost to the cooling water.

5. The design of details to permit automatic regulation of cooling water circulation in proportion to the work developed by the engine, so as to improve the efficiency under low loads.

6. The creation of vertical types of large power, both for marine and for industrial services, in order to reduce the space required and so to lessen the hindrance with regard to free expansion now existent in connection with the larger tandem horizontal engines.

7. A practical means of recovery of waste heat from the exhaust gas.

A certain number of these principles have already led to practical improvements, and the future will suggest others which it is impossible to foresee at the present time.

The ability and knowledge of scientists and constructors will, without doubt, soon overcome the present difficulties. The high efficiencies reported in connection with trials made on internal combustion engines by the best known authorities emphasise the future of gas engines.

Gas Turbines.

A number of experiments have already been conducted with a view to the production of gas turbines, but the author proposes to mention only the opinion of Mr. Dugald Clerk in his presidential address to the Junior Institution of Engineers (November, 1905). Mr. Clerk has given close study to the question, and his competency to express an opinion upon the matter is indisputable.

(1) It is impossible for a gas turbine to be subjected to the same high temperatures as are developed by the combustion of gaseous mixtures in gas engine cylinders. Such temperatures, which are of 1,500° C. to 1,600° C., and even 2,000° C., are practicable with reciprocating gas engines, because, on account of the "cycle," they are prevalent only for very short periods.

In a gas turbine, of which the characteristic feature is the continuous flow of gaseous flame, certain details must be subjected to the same temperature as the gas, but no material exists, as far as is known at the present time, that possesses the mechanical qualities necessary for the working parts of a prime mover of this type. It is therefore indispensable for the temperature of the gases to be considerably reduced before entering the turbine.

Mr. R. M. Neilson, in a paper presented in 1904 to the Institution of Mechanical Engineers (London), estimates that the temperature should not exceed 700° C., which is still a very high limit, for, from trials conducted with respect to the rapidity of oxidation of iron and steel at this temperature, it is certain that the turbine blades, under such conditions, would rapidly deteriorate.

(2) The necessity for lowering the temperature of the gas has brought about proposals to partially transform the thermal energy of the particles into kinetic energy by expansion of the mixture through an expanding jet of the Laval steam turbine type. The gaseous mixture is then compressed in a combustion chamber having a lining of refractory material, the gases escaping from this chamber by a nozzle of a suitable form to direct the gases against the blades of the turbine.

This type of turbine Mr. Clerk considers to be the most practical of those yet proposed; but to obtain a satisfactory efficiency demands (1) a rotary, or turbine compressor of high relative efficiency; (2) an expanding nozzle which shall ensure that free expansion is quantitatively equivalent to adiabatic expansion behind a piston; (3) a rotating turbine of such construction as to secure very high efficiency of transformation of kinetic energy of the moving gas into effective work available at the turbine shaft.

In order to calculate the overall efficiency that should be obtained from this class of turbine, and assuming for each of the three details an efficiency equal to the highest obtained in practice, Mr. Clerk has based his figures upon the hypothesis that the gaseous fluid, following the Joule or Brayton cycle, gives a theoretical efficiency of 48 per cent. To obtain this theoretical efficiency the compression should be carried to 140 lbs. per square inch; the maximum temperature would be 1,700° C., and the temperature at the outlet of the expanding nozzle would be 700° C.

For the compression it would be more advantageous to use a cylinder compressor, because the efficiency of such would be higher than that of a rotary compressor; but to do so would be equivalent to abandoning the advantages of the turbine principle. It is necessary, therefore, to assume an efficiency of 60 per cent. for the compressor; but in order to give the most favourable conditions for the turbine, an efficiency of 90 per cent. will be considered possible.

Similarly, the efficiency for the conversion of thermal energy into kinetic energy will be upon an assumption of 90 per cent., although this figure is much too high. The efficiency of transformation of kinetic energy into useful work upon the turbine shaft is certainly lower. It is recognised, in fact, that the mechanical efficiency of a steam turbine is lower than that of a reciprocating steam engine at high pressures. If the turbine gives more useful work from the energy in the steam, this is on account of the absence of the initial condensation that takes place in the reciprocating engine. On this account it is impossible to rely upon an efficiency of more than 80 per cent.

Using the numbers suggested, 90 per cent. for efficiency of compression, 90 per cent. efficiency of nozzle expansion, 80 per cent. efficiency of conversion in turbine, there would be, with a cycle having negative work equal to 0·4, the following efficiencies:—To get 0·4 of work in compression, 0·445 of work would have to be put into the compression. On expanding in the nozzle only 0·9 of the total energy of the flame gases would be obtained in the shape of kinetic energy, and of that 0·9, only 0·8 would be returned in the shape of available work by the turbine portion. From the turbine, therefore, the total work obtained would be $0·9 \times 0·8 = 0·72$, and deducting the negative work, $0·72 - 0·445 = 0·275$; that is to say, that the thermal efficiency of a turbine working on the cycle under consideration would be 27·5 per cent.

But even then, no account has been taken of any losses of heat due to radiation or carried away by the exhaust. Such losses, according to Mr. Clerk, would amount to 25 per cent. if an assumption be made upon the observations noted in connection with reciprocating engines. Thus the efficiency would be further reduced to 16·5 per cent.

A reciprocating engine, working upon the same Joule cycle of 48 per cent. ideal thermal efficiency, would give in practice at least 30 per cent. indicated efficiency.

(3) But the preceding solution is not the only one that has been proposed for the practical realisation of the gas turbine, suggestions having been made to make use of the greater part of the energy of the combustible mixture in a reciprocating engine and to utilise the exhaust gas in a turbine. One arrangement upon this principle has been tried by Mr. F. W. Lanchester. It would thus be possible to obtain a high efficiency in the turbine; but it could not be accepted as a solution of the problem of the gas turbine for large powers.

To avoid the difficulties resulting from high gas temperature, Professor Reeve has suggested the use of steam to provide the working fluid, the steam when produced to be heated by a very small quantity of combustible mixture of gas and air under pressure.

This would produce a turbine using highly super-heated steam, and thus be a compromise between the flame turbine and the steam turbine. The solution is interesting; but to obtain a higher efficiency than is possible with reciprocating engines it would be necessary that the temperature of the fluid at its entry into the turbine would be much higher than the limit that could be practicable for safe working.

CHAPTER IV

PRINCIPAL TYPES OF GAS ENGINES

Considerations of space render it impossible to refer even briefly to every maker of gas engines or to give details as to the type of engines constructed by them. It is intended, therefore, to review those which present the greater interest, but for the information of the reader a list is given in an appendix of all makers in the various countries as far as is known by the author. Here, as in the appendix, the names will be referred to in alphabetical order.

American.

Allis Chalmers Co., of Milwaukee, Wisconsin, have successfully installed some of the largest gas engines in industrial service in the United States. At the Homestead Works a series of four-cylinder, double-acting, twin-tandem engines comprise thirty-six double units of 4,500 h.p., but in the new works being erected for the Indiana Steel Co. at Gary, on the River Grand Calumet, about twenty-three miles from Chicago, the gas engine installation will be by far the largest in the world.

The sixteen blast furnaces will have a productive capacity of 500 tons of iron per twenty-four hours and be complete with all the improvements embodied in modern practice. From this number of furnaces about 45 million cubic feet per hour will be produced, sufficient to serve gas engines totalling 500,000 h.p. or thereabouts; 30 per cent. of this volume of gas will be employed in regenerative processes; 7½ per cent. will be used in the generation of steam; 5 per cent. will be lost in washing processes or utilised for auxiliary purposes; 12½ per cent. will serve gas-driven blowing engines for the furnaces; and the remaining 45 per cent. will furnish electric current in a central generating station.

The sixteen gas-driven blowing engines are each of 2,500 h.p. horizontal, twin-tandem type, with pistons 42 inches diameter and 54 inches stroke, displacing 3,000 cubic feet of air per minute at a pressure of about 18½ lbs. per square inch, and will work against 30 lbs. per square inch upon occasion.

In the central station seventeen horizontal, twin-tandem, double-acting engines are installed running at 83·3 revolutions per minute; fifteen are direct coupled to alternators and two to continuous current machines. The alternators are three-phase, 25 periods, 6,000 volts, and the continuous current machines will generate at 250 volts.

The seventeen engines are rated at 4,000 H.P., the dynamos are 2,000 KW. and will carry an overload of 30 per cent. They have been built by the Allis Chalmers Co. upon the lines of other similar units supplied by them to the Illinois Steel Co. to work in parallel. The engine frames weigh about 90 tons each, and the lower half of these

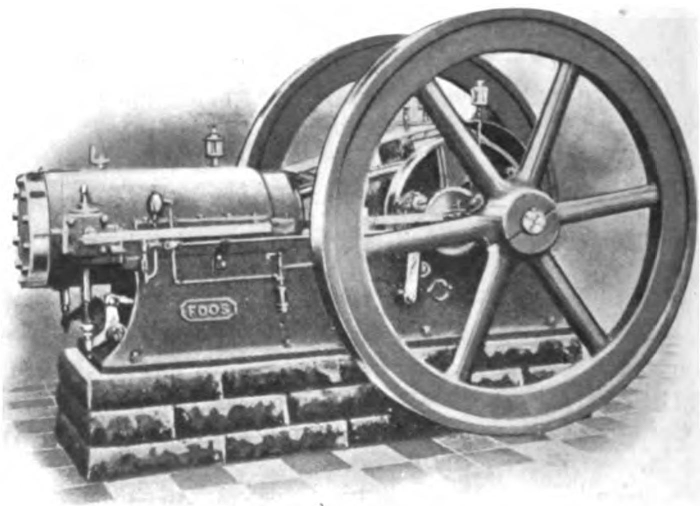


FIG. IV.—1. "Foos" Gas Engine.

frames are embedded in the foundations flush with the floor, giving freedom of access to the valves. The total weight of each engine is about 750 tons. The pistons are 44 inches diameter, and the stroke 54 inches long. The crank pin is 20 inches diameter; the shaft in the bearings, 30 inches diameter; and the fly-wheels, 23 feet diameter, weigh 91 tons.

Other references to this firm will be found on pp. 18 and 316.

Blaisdell Machinery Co., of Bradford, Pennsylvania.—A reference to the double-acting engine built by this firm is made on p. 329.

Buckeye Co., of Salem, Ohio, construct two-cycle engines, a description of which is given on p. 129.

De la Vergne Machine Co., of New York, have been for many years the licensees of the well-known Hornsby-Akroyd oil engine and have been responsible for many installations throughout America for all purposes. They have also developed the two-cycle, double-acting Koerting engine with success, the table facing p. 10 showing a total of 26 units over 1,000 H.P., aggregating 42,200 H.P.

Foos Gas Engine Co., of Springfield, Ohio, build engines up to moderate powers presenting a number of interesting features which

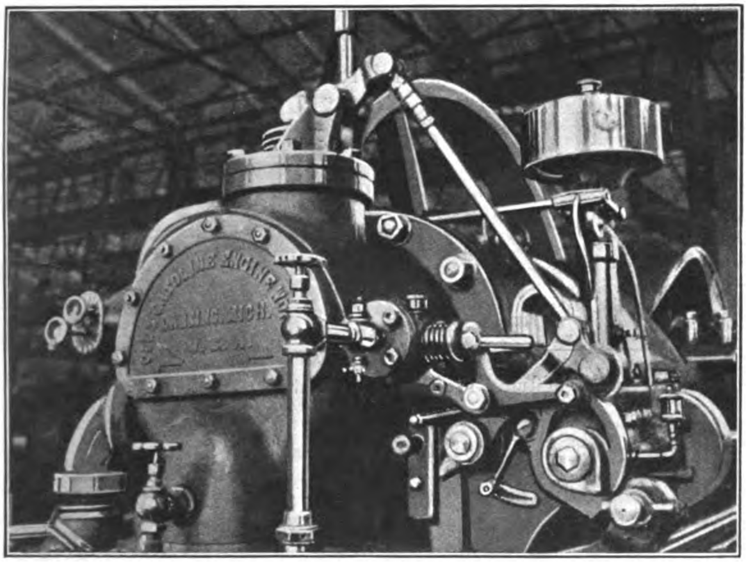


FIG. IV.—2. Olds Gas Power Co.'s 65 B.H.P. Engine, 16¼ inches diameter by 20 inches stroke, developing 85 H.P. on brake test at 215 revolutions per minute.

are alluded to on pp. 285, 319, 335, 354, 374, and 398. An illustration of a small engine is given in Fig. IV.—1.

S. M. Jones Co., of Toledo, Ohio, construct vertical gas engines to which allusion is made on p. 90.

Olds Gas Power Co., of Lansing, Michigan, have been licensees for the gas producers of the Pintsch Co. of Berlin and took up the manufacture of gas engines three years ago in consultation with the author. They have produced an engine combining utmost simplicity with highest efficiency.

The engine has a strong frame casting, with a separate breech end, provided with a large opening for removing the core after casting and for cleaning the water chamber when erected (Fig. IV.—2). Governing is effected by varying the volume of mixture admitted at constant ratio as determined by two valves, controlled by the governor and placed in the gas and air inlet passages respectively. The ignition system is very simple and compact and is illustrated on p. 157.

Riverside Engine Co., of Oil City, Pennsylvania.—This firm has departed from the arrangements and methods generally adopted by European makers in the construction of large double-acting engines. They have designed an engine of which the valve gear and operating mechanism and the form of cylinder differs widely from the principles that are sometimes alluded to as "American practice," with the exception of the side disc crank which seems to be preferred by

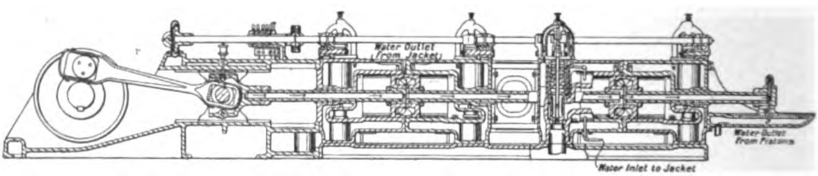
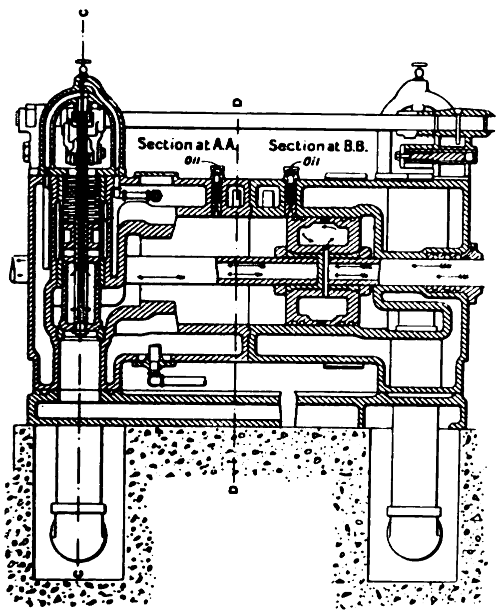


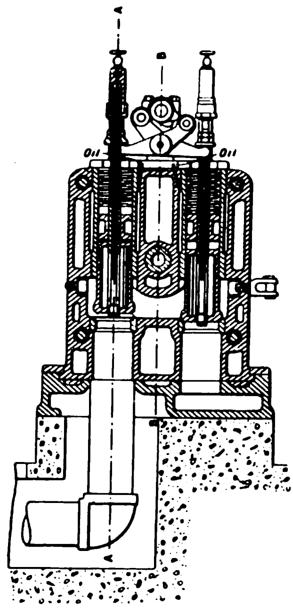
Fig. IV.—3. Longitudinal section of Riverside Engine Co.'s Tandem Engine.

American engineers over the cranked shaft that has become the standard practice of all European makers for engines of all sizes.

The first engine of this type, double-acting, with two tandem cylinders, has been supplied to the Watson Steelman Co., of Aldine, New Jersey, and includes several interesting peculiarities. The inner cylinders are cast with the outer casing, but both are in sections forming two trunks with a central joint over which the piston passes (Figs. IV.—3 and IV.—4). These trunks are held together by four powerful longitudinal bolts placed in each of the four corners of the water jacket, which is of the rectangular form shown in Fig. IV.—5. Each trunk or half-cylinder is fitted with a corresponding breech end with the valves, and is bolted in position upon a long sole plate having carefully-machined supporting surfaces, common to both of the tandem-cylinders.

Above the cylinders and upon the centre line the cam shaft runs longitudinally throughout the entire length and on either side the inlet and exhaust valves are placed, directly operated by short levers (Fig. IV.—5). These valves are identical and interchangeable, and their seats are arranged on the same horizontal plane in the breech casting and below the cylinder level. Equilibrium valves are used similar to those designed by Messrs. Crossley Brothers and referred to on p. 393. They are made in a cylindrical block well cooled internally and the height is sufficient to act as guides in the valve boxes. They can be readily removed from above after the operating levers have been dismantled, and the springs need be of a light section, since they





RIVERSIDE ENGINE CO.'S TANDEM ENGINE.

FIG. IV.—4. Longitudinal section (1 cylinder).

FIG. IV.—5. Transverse section (valve chamber, &c.).

are in equilibrium. No intermediate support is provided for the central portion of the piston rod.

The method of obtaining access to the pistons is of an original character. Instead of bringing these to the end of the cylinder, the half-cylinder, with its valves, cam shaft bearings, &c., is removed, and in this way the piston is inspected by the opening thus created.

The makers do not seem to have attached much importance to the shape of the combustion chamber, which is very irregular and presents a large cooling surface in proportion to its volume. It remains to be seen whether the dismantling of the complete half-cylinder to get at the piston is, in every-day practice, a simpler operation than the

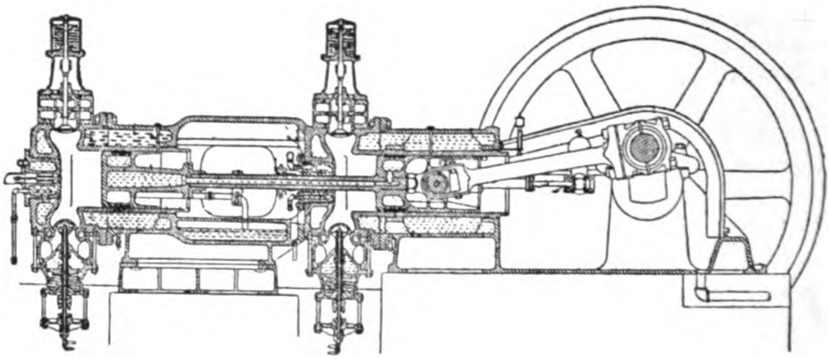


FIG. IV.—6. Longitudinal section of Struthers Wells Co.'s Tandem Engine.

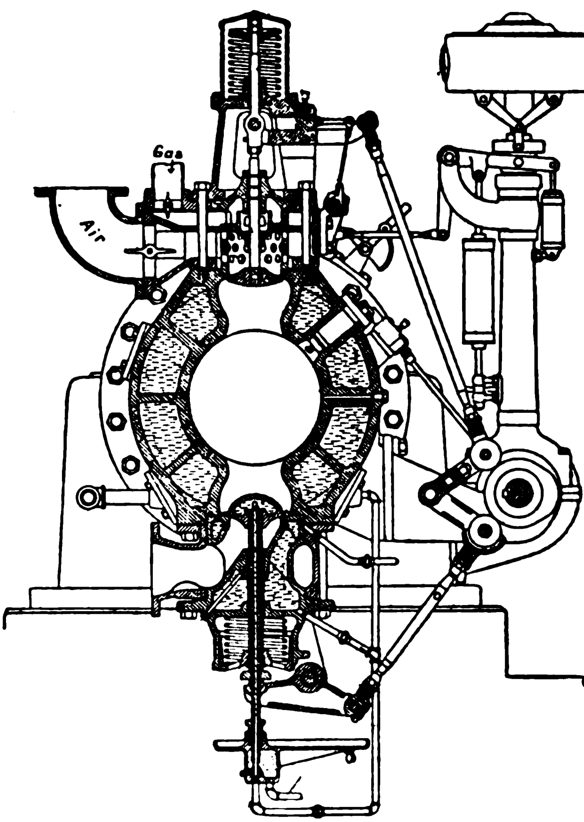


FIG. IV.—7. Transverse section of Struthers Wells Co.'s Tandem Engine.

removal of a cylinder cover. The future will decide also whether the universal practice of providing an intermediate support for the pistons and piston rod in a sliding crosshead can be neglected with impunity.

Snow Steam Pump Works, of Buffalo.—Several references to the large engines made by this firm are made elsewhere (see index).

Standard Motor Co., of Jersey.—This firm have specialised upon vertical engines for marine work, and reference is made thereto on p. 105.

Struthers Wells Co., of Warren, Pennsylvania.—The engine made by this firm deserves special mention as, rather than seeking to Americanise their design, they have decided to adopt the principles embodied in European practice. In Figs. IV.—6 and IV.—7, representing longitudinal and transverse sections of this engine, it will be observed that the inlet and exhaust valves are rationally disposed above and below the cylinder respectively. The liner is independent of the water jacket and is cast in one piece with the breech end carrying the valve boxes. The latter are arranged so that to obtain a tandem-cylinder engine from a single-cylinder engine it is only necessary to add the stuffing box, a second piston with its rod, a second cylinder, and an intermediate portion forming a water jacket. The valve gear is copied from the Güldner system and the exhaust is a combination of a separate box with water-cooled valve seat, while the valve itself is also water-cooled.

Westinghouse Machine Co., of East Pittsburg, Pennsylvania, are well-known makers of horizontal and vertical gas engines, and some of the features of such engines are mentioned on pp. 70, 80, 215, and 329.

Wisconsin Engine Co., of Corliss, Wisconsin, build the Sargent engine which is illustrated and described on p. 79.

BELGIUM.

Carels Frères, of Ghent.—For several years this firm have undertaken the construction of Diesel engines, and many successful installations have been carried out for both European and Colonial purchasers.

Société John Cockerill, of Seraing.—Throughout the present volume reference is made to the well-known engines built by Messrs. Cockerill, who were pioneers in the construction of double-acting engines and also of the large single-cylinder engines for blast furnace gas.

England.

The Anderston Foundry Co., Ltd., of Glasgow, specialise on high-speed vertical gas engines suitable for direct coupling to dynamos, &c.

Elijah Ashworth, of Colleyhurst, Manchester, pays special attention to horizontal gas engines of 50 B.H.P. and under.

W. J. Bates & Co., Ltd., of Denton, Manchester, have been established for some years, and produce horizontal gas and oil engines in general

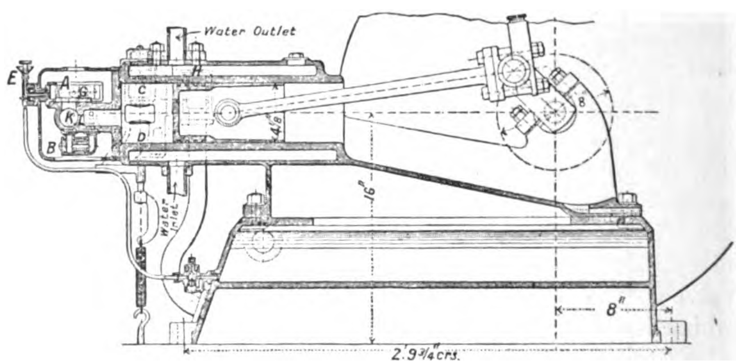


Fig. IV.—8. Britannia Engineering Co.'s Petroleum Engine.

accord with the majority of British engines. Recently they have revised their models with the view of adopting continental features of design and construction. The largest engine yet made has a 24-inch cylinder.

Wm. Beardmore & Co., Ltd., of Glasgow, are the licensees of the Oechelhauser engine, and have several of this type at work giving satisfactory service at their Parkhead Forge and elsewhere.

Blackstone & Co., Ltd., of Stamford, have earned a world-wide reputation for their oil engines for agricultural and colonial purposes, and have lately brought out a crude oil engine which has several distinctive features. Somewhat upon the lines of the Diesel principle the liquid fuel is injected only after the compression of the pure air, but instead of the latter reaching a pressure of about 500 lbs. per square inch the more moderate figure of 150 lbs. serves the required purpose. The oil is delivered to an injection valve, the latter being provided with two passages. In one of these the oil is first led to an ignition bulb kept hot during normal work by the heat stored up from the consecutive explosions, while the other passage leads to an orifice at the end of the combustion chamber so that upon subsequent admission of a charge of compressed air from a separate receiver at 400 lbs. pressure, the auxiliary fuel is forced into the combustion chamber for a longer or shorter period of the stroke according to the quantity of oil pumped by the governing mechanism. This oil enters the cylinder as spray at the moment when the charge just previously admitted to the ignition bulb flashes out into the combustion chamber.

Britannia Engineering Co., Colchester.—This firm specialise upon oil engines from ½ to 70 H.P. The general type of their engines is represented in Fig. IV.—8. The type of vaporiser employed is described on p. 171.

British Westinghouse Electric and Manufacturing Co., Ltd., of Trafford Park, Manchester, have greatly developed and improved their original type of vertical gas engine designed in America. An example of one of their present engines of 750 H.P. is illustrated and described on p. 92. Many gas-electric power stations have been furnished with Westinghouse engines both for private and public works.

Brown & May, Ltd., of Devizes, Wilts, have an extensive connection for their steam traction and other engines, and have for several years taken up the manufacture of oil engines for colonial and agricultural purposes in the smaller sizes.

Campbell Gas Engine Co., Ltd., of Kingston, Halifax, have built up a large business both for home and export trade in both gas and oil engines. Their vertical engines are described on p. 94.

Capel & Co., Ltd., of Dalston Lane, London, specialise mainly upon horizontal gas and oil engines of moderate powers. A revision of their models has recently been completed, and the general design accords with continental practice.

Crossley Brothers, Ltd., of Openshaw, Manchester, were the English licensees of the original Otto patents, and they so successfully exploited the invention that the numbers made and sold by them exceeded the output of the Gasmotoren Fabrik Deutz itself. All the improvements that marked the stages of gas engine progress were applied by "Crossley's," the supersession of the slide valve, scavenging, the adoption of separate cylinder heads, the application of multiple cylinders in different combinations, &c., &c. Until lately they have remained faithful to the system of hit-and-miss governing, being justly entitled to point to the satisfactory working of the very considerable number of engines they have made upon this principle.

Recently they have undertaken the manufacture of engines of 500 and 600 H.P., for which such a method of control is utterly unsuitable, and in Chapter XI., p. 254, a description is given of the unique and wonderfully sensitive yet simple device invented by their chief engineer, Mr. James Atkinson, and applied both to the large horizontal gas engines as well as to the multiple-cylinder vertical gas engine recently put upon the market.

The following is a statement of the output of the firm since it first commenced the manufacture of the old Otto & Langen rack type of engine:

Oil and spirit engines . .	5,750
Single-acting, single-cylinder engines	54,245
Single-acting, multiple-cylinder engines . .	225
Double-acting . . .	1
Total . .	60,221, aggregating 973,380 H.P.

Davey, Paxman & Co., Ltd., of Colchester, have been known for many years as makers of steam engines, and several years ago also commenced the manufacture of gas and oil engines and suction gas producers. An example of one of the largest engines made by them was exhibited in the Franco-British Exhibition in 1908, being a two-cylinder, single-acting engine with opposed cranks.

The Diesel Engine Co., Ltd., hold the English rights of the Diesel patent and have carried out many very successful installations for the home industries as well as for those in the different countries owning allegiance to Great Britain. The engines, however, have mainly been built in Germany or Switzerland.

Mirrlees, Watson & Co., of Glasgow, who for many years have built Diesel engines under licence, have recently erected a new works specially equipped at Hazel Grove, near Stockport, Manchester, carried on under the title of Mirrlees, Bickerton & Day, Ltd.

The Dudbridge Ironworks, Ltd., commenced the manufacture of engines in 1892 under the name of Humpidge & Snoxell. The history of the firm is one of continued progress and their engines have always been distinguished for simplicity and strength. Hit-and-miss governing has been the system of regulation adopted both for gas and oil engines, the general appearance of the engine being distinctive of the English type.

Recently an entirely new engine has been evolved, fitted with a governing arrangement with variable admission of mixture at constant ratio, and its simplicity of mechanism and pleasing general appearance is conspicuous. The leading dimensions of their 90 H.P. engine are:—Piston 17½ inches diameter by 25 inches stroke, running at 160 revolutions per minute.

In addition to ordinary industrial engines, they build vertical engines for launches and oil tractors.

Fielding & Platt, Ltd., of Gloucester.—This firm is one of the oldest engaged in the manufacture of gas engines. Mr. John Fielding was one of the first to employ superimposed inlet and exhaust valves upon the same axis and to govern the engine by the admission of graduated quantities.

Several powerful engines for use with town gas were exhibited at Antwerp in 1885. Messrs. Fielding make engines of various designs, some with overhanging cylinders with the breech end cast in one with the water jacket and horizontal inlet valve, and others with the cylinder supported upon side bearers, independent breech end, and superimposed valves.

Vertical four-cylinder engines are also built for producer gas.

L. Gardner & Sons, Ltd., of Patricroft, Manchester, are makers of both horizontal and vertical types of gas and oil engines. They have specially devoted themselves to the production of engines for use in motor-boats, while a number of multi-cylinder vertical engines running at 500 to 600 revolutions per minute have been supplied for direct coupling to dynamos.

Griffin Engineering Co., Ltd., of Bath.—Reference is made elsewhere to the vertical oil engines made by this firm and to the special form of vaporiser adopted by them. Recently an improvement in the actuating gear for rotary magnetos has been introduced in which by means of toothed wheels of elliptical shape, an increase in the velocity of revolution is obtained at the moment when ignition is desired.

Richard Hornsby & Sons, Ltd., of Grantham.—This firm have a world-wide reputation as the makers of the Hornsby-Akroyd oil engine which has been in successful service for many years. A few

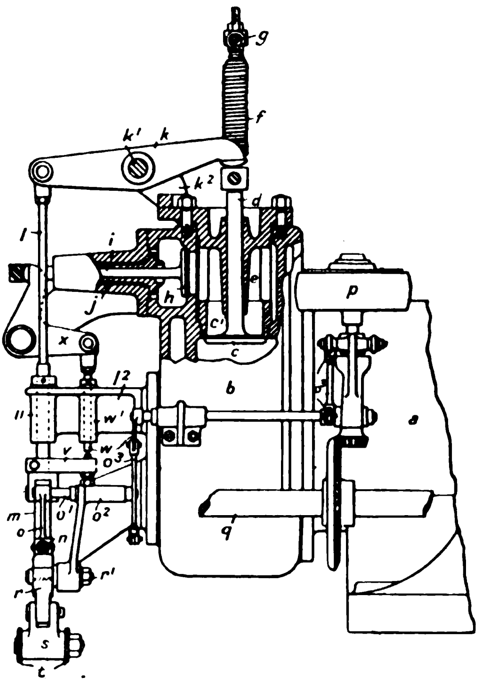


FIG. IV.—9. Hornsby-Stockport governing mechanism (variable admission).

years ago, by the amalgamation with the firm of J. E. H. Andrews, Ltd., of Stockport, they took up the manufacture of gas engines and have since greatly developed this branch of their extensive business, which also includes steam engines, both portable and stationary, and harvesting machinery.

Reference is made elsewhere to some of the special features of their gas engines, while within the last few months a new type of valve gear has been brought out which is illustrated in Figs. IV.—9 and IV.—10. As will be seen, the main inlet valve is placed in an inverted position on the top of the cylinder, while the gas valve is arranged horizontally at the rear. A single lever actuates both valves, a clamping bar *v* projecting from the push rod *l* to make contact with the gas valve mechanism by means of the rod *w* and the bell crank lever *x*. Upon the introduction of a charge into the combustion chamber *b*, the cam *u*, through the medium of the lever *t*, lifts the link *r*, thus causing it to raise the finger *m* and thereby the push rod *l* so as to lift the inlet valve *c* from its seat. When *l* has been raised to a predetermined

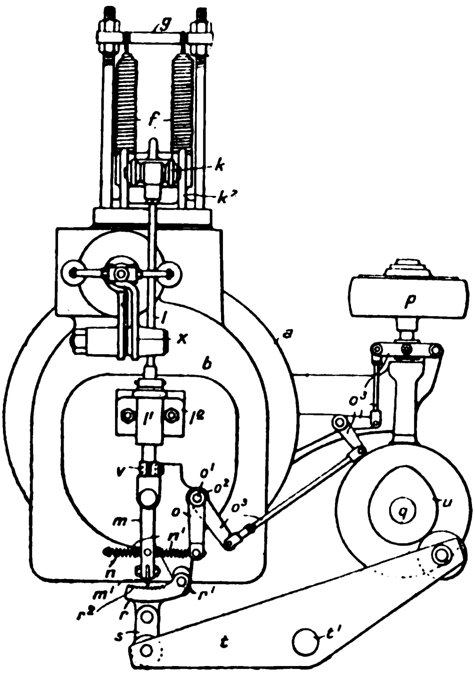


FIG. IV.—10. Hornsby-Stockport governing mechanism (variable admission).

extent, the tappet *v* strikes the lower end of *w*, so causing the gas valve *h* to open also. As the speed of the engine varies, the governor *p* moves the finger *m* relatively to the plunger rod *l*, so that more or less lost motion has to be made up before contact of the engaging points is made. Thus both the time of opening and the amount of lift given to the valves is varied in accordance with the load on the engine, but the two valves are equally affected, ensuring constant ratio of mixture.

National Gas Engine Co., Ltd., of Ashton-under-Lyne.—This well-known firm, which includes Mr. Dugald Clerk, F.R.S., upon its

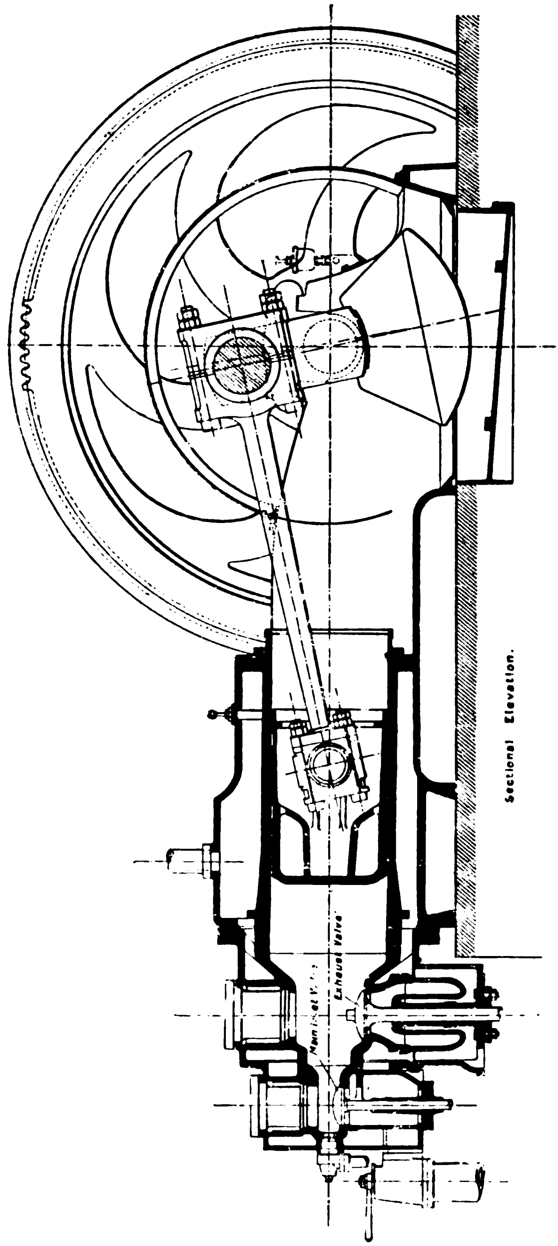


FIG. IV.—11. 100 H.P. "National" Gas Engine.

directorate, has taken a high place amongst leading gas engine makers for the high standard of excellence and extreme simplicity that characterise their productions. A section of one of their latest designs for a 100 H.P. engine is shown in Fig. IV.—11, and on pp. 161 and 270 reference is made to the system of valve gear and ignition adopted in this engine.

Premier Gas Engine Co., Ltd., of Sandiacre, near Nottingham.—This firm have adopted a positive method of scavenging the products of combustion from the cylinders of their engines at the end of the exhaust strokes. The air piston works within a cylindrical portion of the main frame to which the water jacket of the motor cylinder is attached by a flange. The underside of the casing is planed and rests upon the foundation. The combustion chamber is independent and overhangs the cylinder. By the provision of a stuffing-box in place of the end cover, the engine is readily converted into a tandem double-cylinder engine.

The positive scavenging is effected in the following manner:—Assuming that the engine is commencing its first out-stroke, the section given in Fig. IV.—12 shows the admission valve *E* open, the exhaust valve *H* is partially closed and the gas valve *G* quite closed. The latter opens at a more advanced position of the piston. Air is admitted by the pipes *S B* and the flap valves *F*. One portion of the air enters by the opening *D* into the cylinder *N*, and another portion enters the motor cylinder *Z* by the passage *C*, the ports *P*, and the valve *E*.

The gas is fed by the valve *G* and the openings *P*, where it mixes with air. During the compression stroke the air in the front cylinder *N* is compressed, but on account of the large volume of the compression chamber the pressure only reaches 4 or 5 lbs. per square inch, falling practically to that of the atmosphere at the end of the expansion period in the cylinder *Z*.

Just before the completion of the expansion stroke, the exhaust valve *H* is opened and the cylinder pressure is reduced and the succeeding stroke expels the products of combustion in the usual way, meanwhile the air behind the piston *A* is compressed until the crank reaches the position shown in Fig. IV.—12. At this point the admission valve *E* opens and the air from *A* passes through the ports *PP* and the inlet valve *E* into the space *Z* as indicated by the arrows. This air, the quantity of which increases as the volume of the space *Z* diminishes, sweeps all the remaining residual gases from the space and replaces it with cold air. A considerable excess of air is furnished

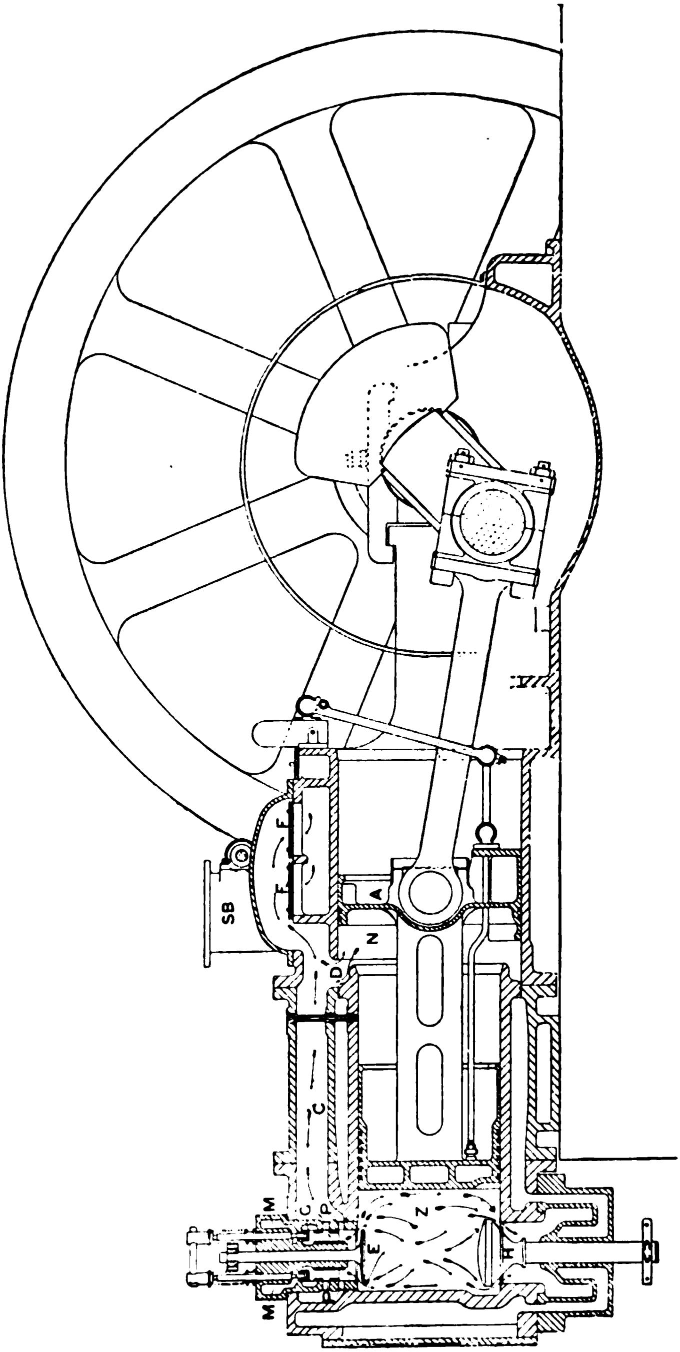


FIG. IV.—12. "Premier" positive scavenging Gas Engine.

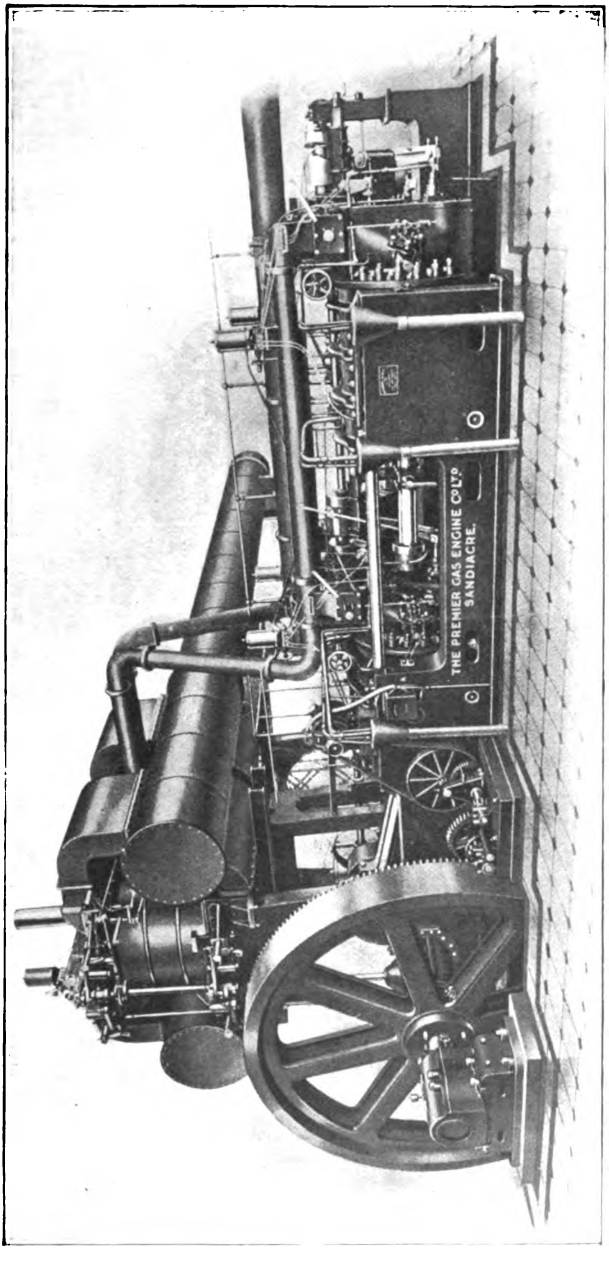


FIG. IV.—13. 2,000 H.P. "Premier" Gas Blowing Engine.

by the piston *A* to ensure complete scavenging. Not only is the space *Z* thus filled with cool air but the valves and internal surfaces are cooled and prevented from getting overheated.

A large number of these engines have been supplied, and in

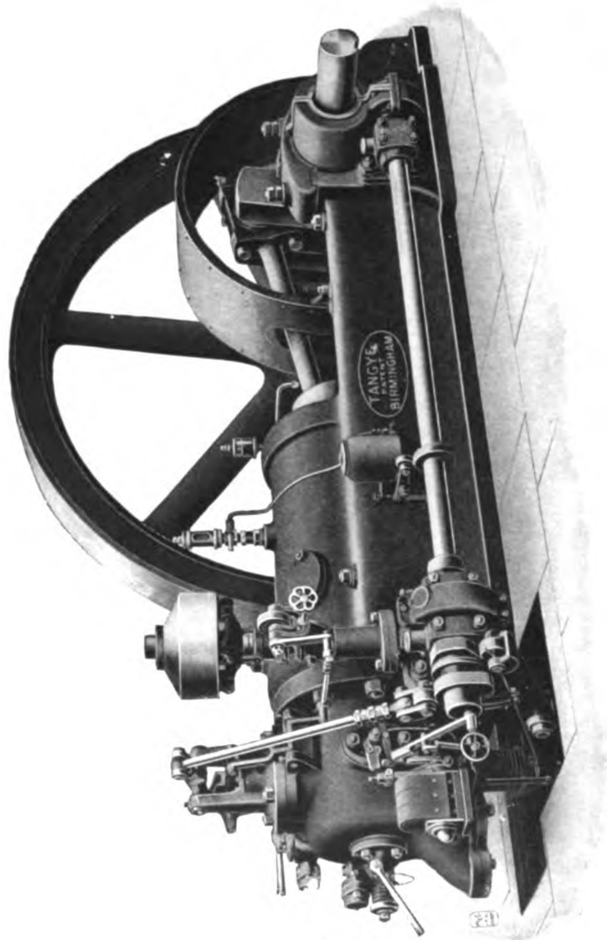


FIG. IV.—14. Tangye "T" type Gas Engine.

Fig. IV.—13 a view is given of a 2,000 H.P. gas-blowing engine for blast furnace gas as installed in the works of Sir Alfred Hickman, Ltd.

Tangyes Ltd., of Birmingham, have been makers of gas engines for many years, and the gas engine department of their immense works now occupies a considerable area. The designs of their earlier engines conformed generally to the British style of construction with overhanging cylinder and hit and miss governing, but the careful designing of the combustion chamber enabled them to obtain very efficient and economical results. In September, 1905, the author had the privilege of making exhaustive tests of three engines rated at 8½, 20, and 29 B.H.P. respectively, and the thermal efficiencies based upon the number of B.Th.U. contained in the coal at full loads were as much as 20·7, 22·9, and 24·1, which for such small engines must be considered remarkable. In April, 1909, a further test was made by the author upon an engine of new design, the general features of which can be seen from the Fig. IV.—14. Space will not permit of the full figures to be given here, but the following summary must suffice: The engine had a piston 16½ inches diameter with stroke of 23 inches, and ran at a speed of 190 revolutions per minute. The trial at the rated load of 68 B.H.P. was continued for ten hours. Five hours' test was made of the engine at 81 B.H.P., and an overload test of 88·75 B.H.P., for sixteen minutes.

The mechanical efficiencies at the various loads were 84·2, 86·3, and 90·2 respectively. The consumption of anthracite amounted to only 0·72 lb. for the rated H.P., and 0·665 for the maximum load test, both on the basis of B.H.P. hour. The gross thermal efficiencies based upon the number of B.Th.U. in the coal being 0·248 and 0·269 respectively per B.H.P. hour. The latter figure is the highest that the author has had the opportunity to ascertain up to the present, this being due as much to the efficiency of the producer as to the high average pressure and high mechanical efficiency of the engine.

Messrs. Tangyes build engines either right or left hand sc that two can readily be coupled to form a unit of considerable power.

GERMANY.

Benz & Co., Mannheim.—This firm took up the manufacture of gas engines in 1883. At that time they placed two-cycle engines upon the market, but when the Otto patents expired these were abandoned in favour of four-cycle engines. Horizontal engines are made for powers varying between 2 and 150 H.P.

Governing is by variable volume of mixture at constant ratio by means of a butterfly controlled by the governor. For the smaller engines 2 to 8 H.P. hit and miss governing is adopted. The manufacture of the engines is very carefully carried out, and the principles that have been mentioned as indicating the features of good construction are observed.

Vertical two-cylinder engines are built of 40 H.P. or 80 H.P. with four cylinders to run at 300 revolutions per minute, specially designed for pinnaces.

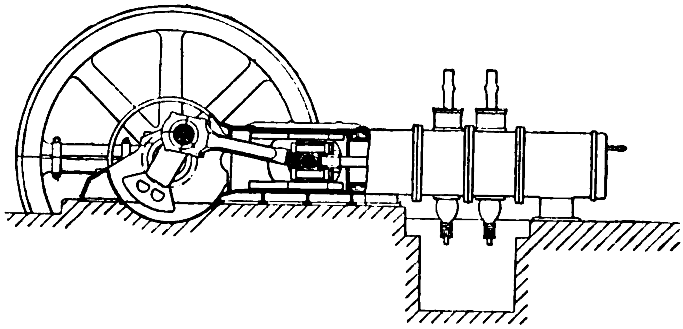


FIG. IV.—15. Dingler Gas Engine, part sectional elevation.

During the period 1883 to 1907 they have built 9,500 engines, aggregating 20,240 H.P., or a mean power of 21·3 H.P.

Dinglersche Maschinenfabrik, A. G., Zweibrücken (Alsace).—The Dingler engine differs considerably from the usual arrangements

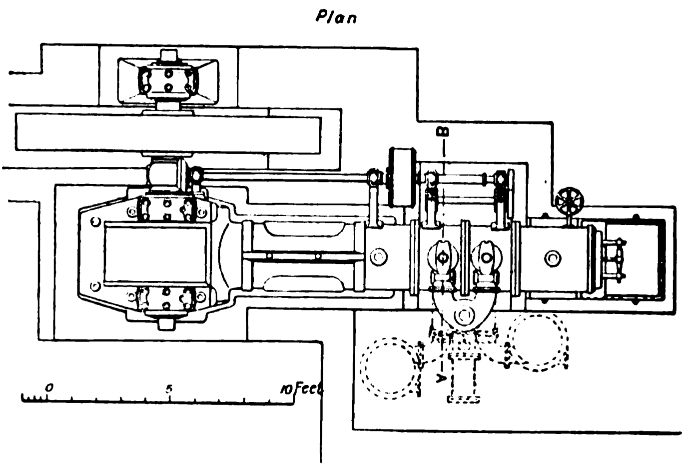


FIG. IV.—16. Dingler Gas Engine, plan.

adopted by other makers of large engines. (Fig. IV.—15 and 16.) It comprises two open-ended cylinders joined together at the combustion chamber. As shown in Fig. IV.—15, each of the cylinders contain a piston, the two being connected by one rod. The explosion takes effect alternately on the internal face of one or other of these pistons. The rod common to the pistons is made in sections and moves in a sleeve which passes through the common partition of the two portions of the cylinder. This partition, as well as the jacket of the double cylinder, is water-cooled. The valves are arranged as in ordinary single-acting engines.

The system gives free expansion to the casing and the internal cylinder which are secured only at one end. The cylinders being

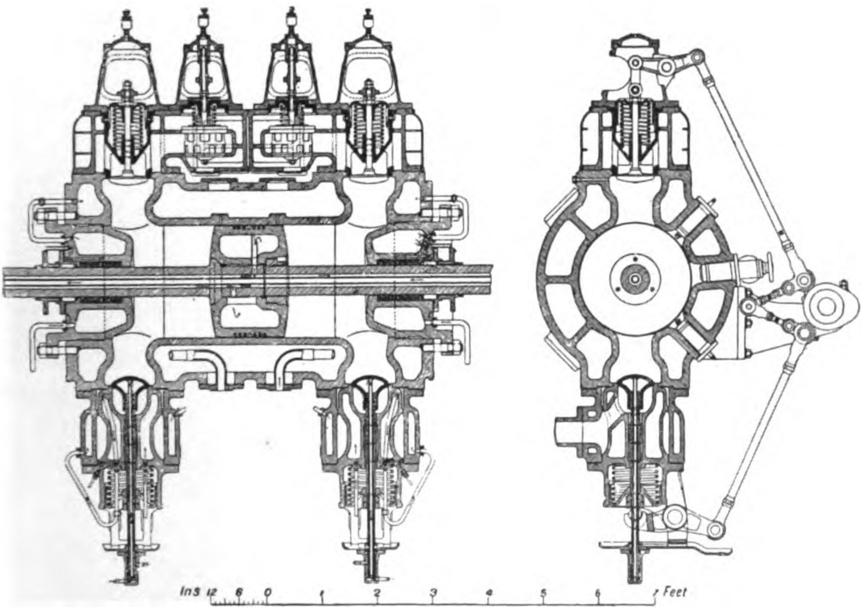


FIG. IV.—17. Ehrhardt & Sehmer, 700 H.P. Engine.

open, inspection and maintenance is as simple as in the ordinary single-acting engines.

Ehrhardt & Sehmer, Schleifmühle (Alsace).—This firm who make a speciality of material for metallurgical works is well known as being makers of large engines, and during the space of about five years have made about sixty engines, together representing 69,790 H.P. These are double-acting engines with twin or tandem-cylinders. Fig. IV.—17 shows a longitudinal and transverse section through the inlet valve and cylinder of a 700 H.P. tandem engine. The Ehrhardt & Sehmer engine is distinguished for the features of good construction with regard to cooling, control, &c., common to this type, mention of which has been made in the course of this present volume.

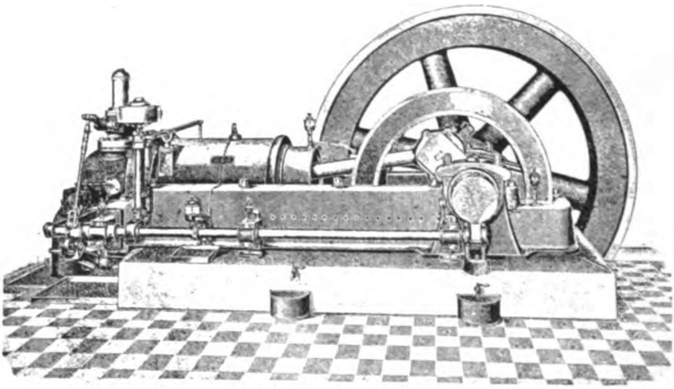


FIG. IV.—18. Schmitz Engine.

Gasmotoren Fabrik Aktien Gesellschaft, Cologne, Ehrenfeld.—This company build the Schmitz engine originally designed by its founder and improved by the engineers who have succeeded him. The

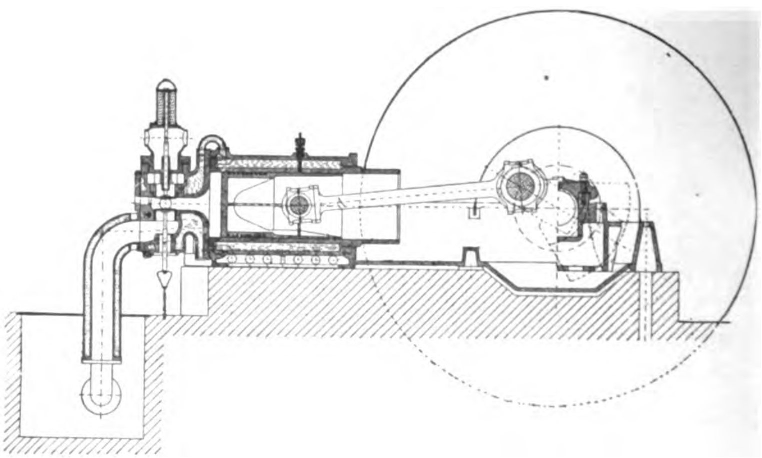


FIG. IV.—19. Sectional elevation of Schmitz Engine.

maximum power of the engines made is 190 H.P. for one cylinder or 380 H.P. for one twin engine. Since its establishment the works have turned out about 60,000 H.P. Above 20 H.P. the engines are fitted with the system of valve gear described on p. 226.

As shown in Figs. IV.—18 and 19, the engines made by this firm are in agreement both in detail and appearance with the standard practice obtaining in Germany, which, in the author's opinion, is bound to become general.

Gasmotoren Fabrik Deutz, Cologne, Deutz.—The Otto Deutz Co.'s services in connection with gas engines are well known. Continuing the traditions of its illustrious founders, Otto and Langen, they have made the four-cycle engines, to which the inventor has given his name,

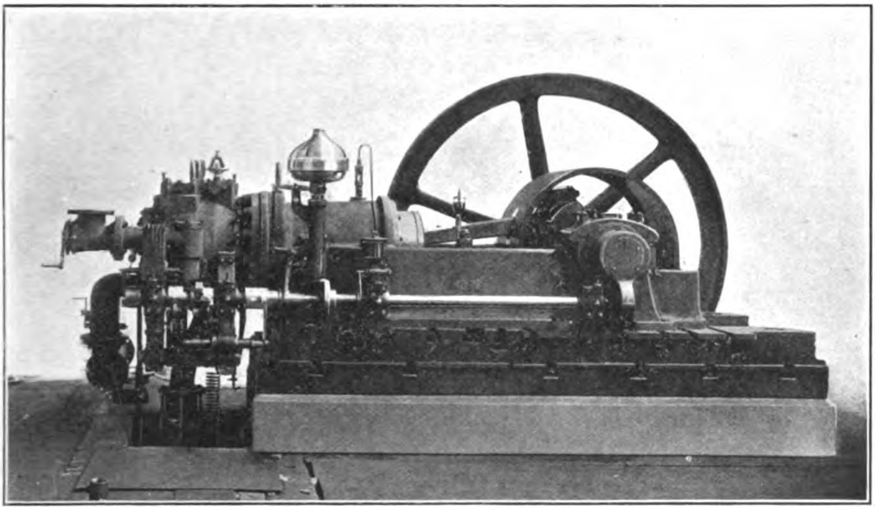


FIG. IV.—20. Otto-Deutz, single-cylinder, four-cycle Engine.

for more than forty years. Since its formation this company has produced nearly 70,000 engines.

The first engines built, of course, were small single-cylinder, horizontal and vertical types. Since 1895 large engines for blast furnace gas have been constructed with one, two, or four cylinders, single-acting, twin or double-twin, of 125, 500, and 1,000 H.P. and over. (Figs. IV.—20, 21 and 22.)

The four-cylinder 1,200 H.P. engine exhibited at Düsseldorf in 1902 was much admired as being the most powerful engine made in Germany at that time. It was built to drive a blast furnace blower delivering 35,000 cubic feet of air hourly at a pressure of 7 lbs. per square inch. The total weight of this fine engine was 219 tons, of which the fly-wheel weighed 19 tons. This weight is enormous as compared with that of steam engines against which low-grade gas engines must compete. These large single-acting engines weigh about 400 lbs. per H.P. Their price was necessarily high and the space

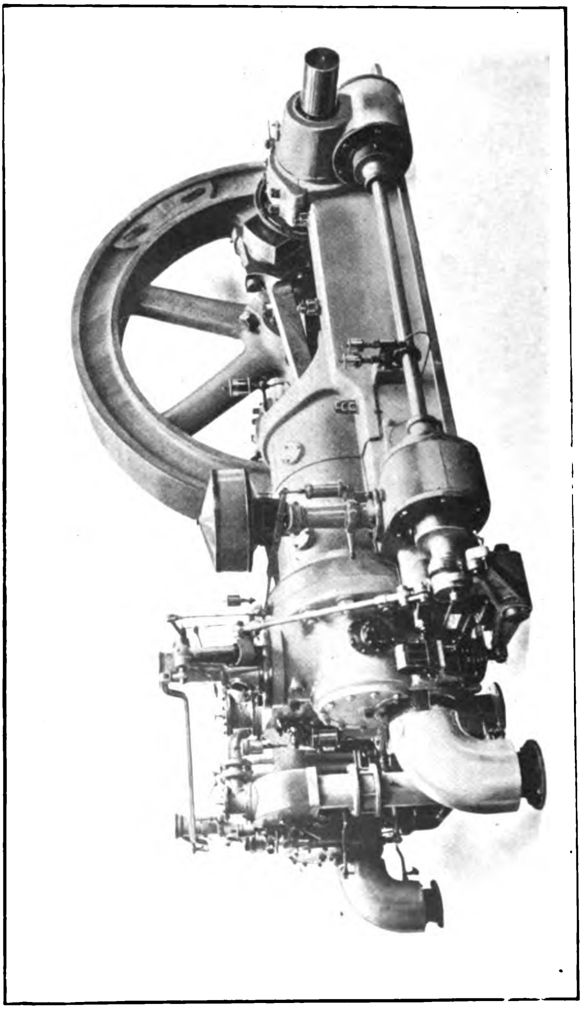


FIG. IV.—21. Otto-Deutz, double-cylinder Engine.

occupied excessive. An economical solution was imperative in this connection as otherwise the application of large engines would have been greatly impeded. This solution has been found in the double-acting engine which is now the standard style of construction of the Gasmotoren Fabrik Deutz. The single-acting type is retained for

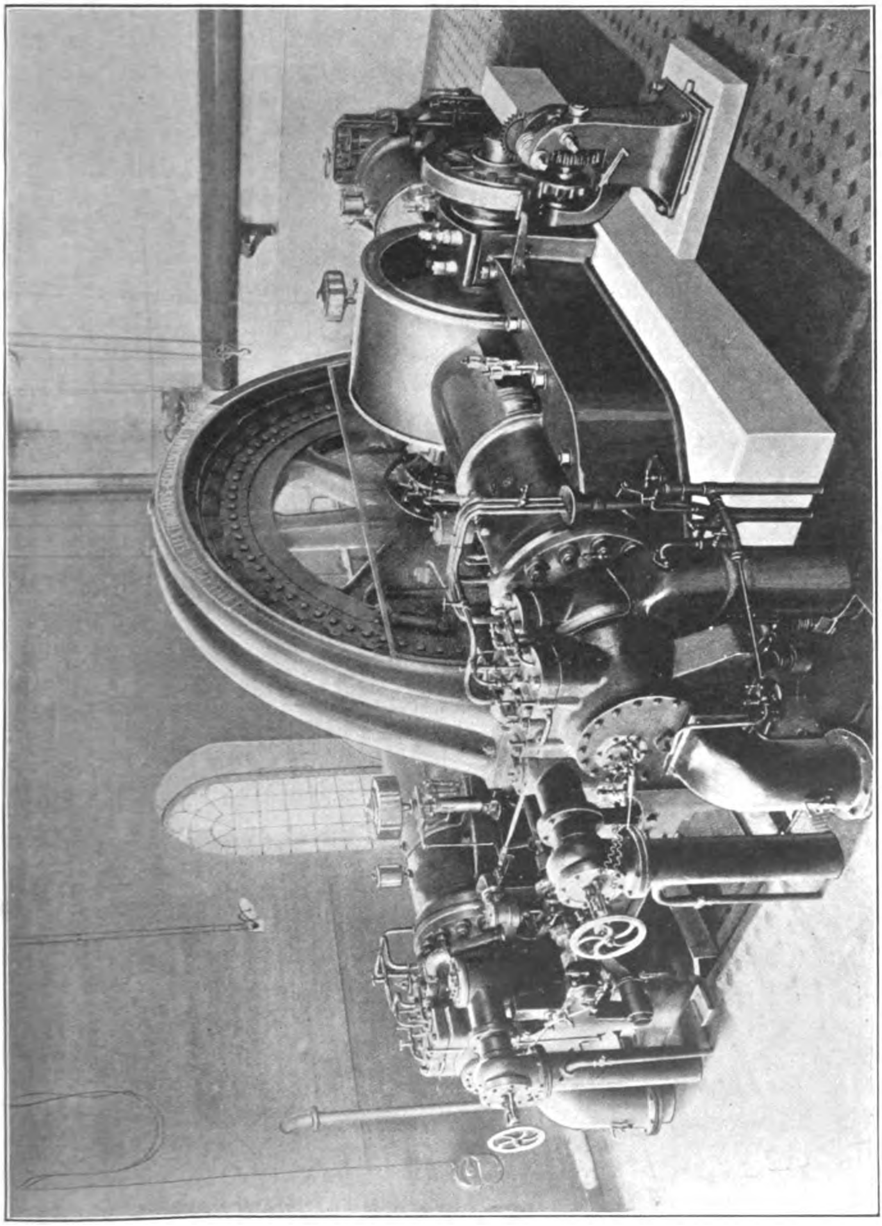


FIG. IV.—22. Otto-Deutz, four-cylinder (double-twin), single-acting Engine.

small powers, and for a little more than two years these have been fitted with a uniform system of valve gear described on p. 220.

For engines of 120 to 150 H.P. in one cylinder the mixture valve is separate from that of the gas in order to make the latter more accessible for cleaning. A similar arrangement is followed in the large double-acting engines.

For 200 to 300 H.P. the former type of engine was with twin-cylinder, either side by side or opposed, or with two cylinders *vis-à-vis*. The latter, however, has been abandoned by the chief makers in Europe, as experience has shown that it involves excessive wear. (See p. 71.)

Since 1901 the Otto-Deutz Co. have taken up double-acting engines, giving preference to the four-cycle type that is more generally used. In 1902 they set their first 200 H.P. double-acting engine to work in their central electric station where it served as an experiment. Since that date it has been used in their general service. Meanwhile they have built eighty-two engines of this type representing 47,500 H.P. either in their own works or those of their licensees.

The Gasmotoren Fabrik Deutz make every kind of internal combustion engine, including small spirit and alcohol engines, marine type engines, benzol engines, &c. Recently they have undertaken the manufacture of Diesel engines, the original patents of which have expired. (See p. 90.)

Since the foundation of the firm the number of engines produced, including those of Langen & Wolf in Vienna, who are licensees, are as follows:—

	Number.	Total Power.
Alcohol engines . . .	859	6,934 B.H.P.
Oil and spirit engines . .	12,384	87,128 ,,
Single-acting gas engines:		
Poor gas . . .	3,405	116,383 ,,
Town gas . . .	28,547	199,653 ,,
	45,195	410,098 ,,
Double-acting engines . .	82	47,400 ,,

If the engines built by Messrs. Crossley Brothers, Ltd., of Manchester, were added the total figures would be 105,498 engines, aggregating 1,430,878 H.P.—irrespective of the number made by the Otto Gas Engine Works of Philadelphia, U.S.A.

Görlitzer Maschinenbau Actien Gesellschaft, Gorlitz.—This company builds engines from 50 H.P. to the largest powers. The smaller engines are single-acting and the larger double-acting. In 1907 the company was re-organised and entirely new designs adopted. Special study has been given to the form of stuffing boxes.

Güldner Motoren Gesellschaft, Munchen-Giesing.—This firm was founded several years ago by M. Güldner, whose excellent book "Calcul

FIG. IV.—23. Güldner vertical Gas Engine.

et Construction des moteurs a combustion," is referred to elsewhere, and in accordance with the principles enunciated in that volume in connection with vertical engines (Fig. IV.—23) this type has been adopted, with one or more cylinders in preference to horizontal.

The remarkable results obtained by Professor Shrötter from a Güldner engine have been mentioned on p. 526, and on p. 223 a description has been given of the system of valve gear applied.

Haniel & Lueg, Düsseldorf.—This company build single-cylinder engines from 150 to 300 H.P., and single-acting tandem engines from

340 to 2,000 H.P. The double-acting engines are of such dimensions that 1,000 H.P. can be obtained from one cylinder, 2,000 H.P. from a tandem engine, and 4,000 H.P. from a twin-tandem engine. The consumptions guaranteed by these makers are as follows :—

Working load or overload	= 9,100 to 9,700 B.Th.U. per B.H.P. hour.
$\frac{3}{4}$-load	= 11 per cent. more.
$\frac{1}{2}$-load	= 15 per cent. more.
Cylinder oil consumption	= 0·0015 to 0·002 lbs. per B.H.P. hour.
Bearings, &c.	= 0·00066 to 0·0011 lbs. per B.H.P. hour.

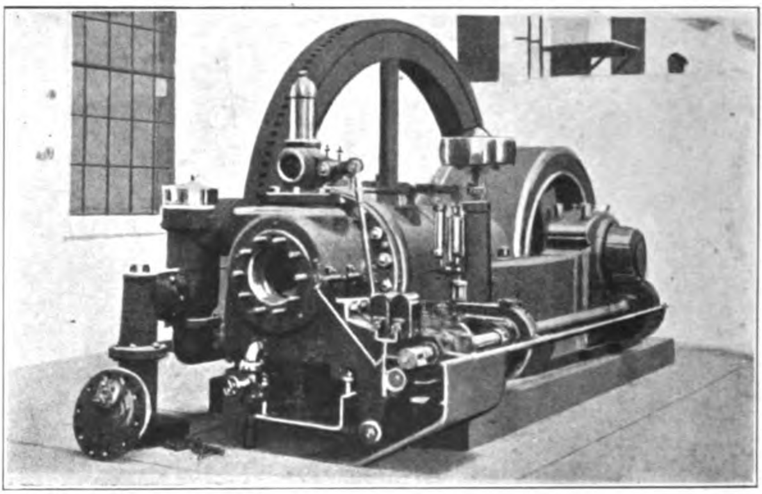


FIG. IV.—24. Koerting four-cycle, single-acting Engine.

Up to the present time thirty-four engines representing 55,000 B.H.P. have been made, including three units of 4,000 H.P. each.

Gebruder Koerting Acktien Gesellschaft, Kortingsdorf.—Messrs. Koerting Brothers, of Hanover, began to build gas engines in 1881 and producers in 1889. Up to 1896 this firm had produced about 3,500 engines equivalent to 15,000 H.P. Since that date the Koerting shops have made 7,200 new engines. In recent years 50,000 H.P. in two-cycle engines have been supplied. Along with the two-cycle type four-cycle engines have been developed, the sale of which has reached 100,000 H.P., while several thousand H.P. is in progress.

Messrs. Koerting build four-cycle, single-acting engines representing about twenty-three different sizes from 2 to 200 H.P. (Fig. IV.—24).

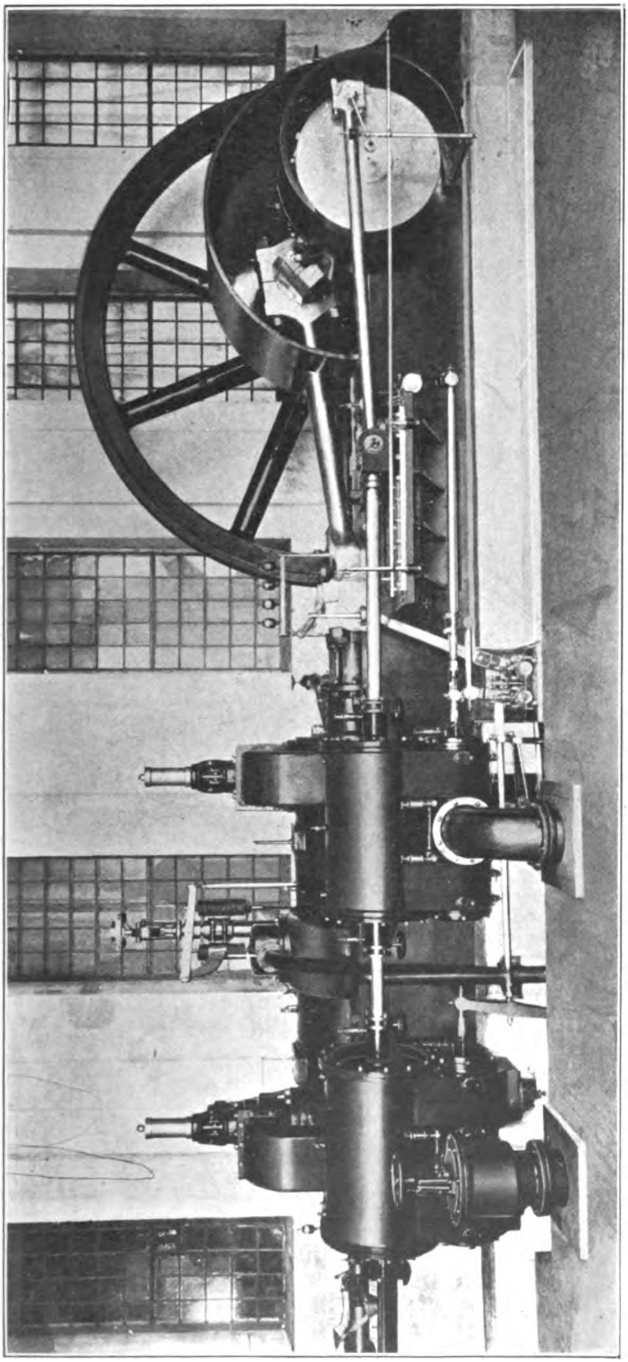


Fig. IV.—25. Koerting two-cycle, double-acting Engine.

The latter has a cylinder 24·8 inches diameter by 35·4 inches stroke, and runs at 150 revolutions per minute. They also build twin engines up to 400 H.P. on the same lines, while recently they have constructed a four-cylinder 600 H.P. engine. This engine forms part of an installation in which the gas used is impure and contains a large proportion of tar. The design permits partial cleaning without interrupting the normal operation. In this case it is preferable to four-cycle double-acting engines and to two-cycle engines in which the dismantling and cleaning of the cylinders and valves is carried out with some difficulty.

According to the dimensions given for the 200 H.P. engine the output is obtained with a mean pressure at 77 lbs. per square inch, assuming a

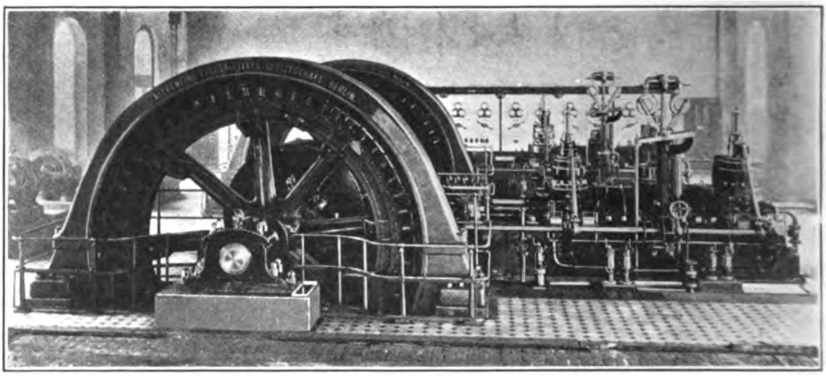


FIG. IV.—26. Two 500 H.P. Koerting Engines, forming part of 7,500 H.P. installation at Gutehoffnungshütte, Oberhausen.

mechanical efficiency of 80 per cent. As a matter of fact the Koerting single-acting engine often reaches 85 to 86 per cent. efficiency.

Double-acting engines, both two-cycle and four-cycle types, are generally made for outputs exceeding 500 H.P., but the firm also make two-cycle engines of about 300 H.P. for direct coupling to pumps. A series of these engines working with town gas and producer gas are installed in the Berlin Waterworks.

Finally, they build the engines with either one cylinder, Fig. IV.—25, or two twin cylinders. The Koerting Co. is one of those who have carried out the more important installations with regard to size. Special mention may be made of the 7,500 H.P. central station at the premises of Gutehoffnungshütte, Oberhausen (Rheinland). Fig. IV.—26 shows two of these 500 H.P. engines.

Koerting engines were the first explosion engines employed in America to work with blast furnace gas, and this installation was the more remarkable owing to the fact that the total power amounted to 42,000 H.P. The installation was equipped by the De la Vergne

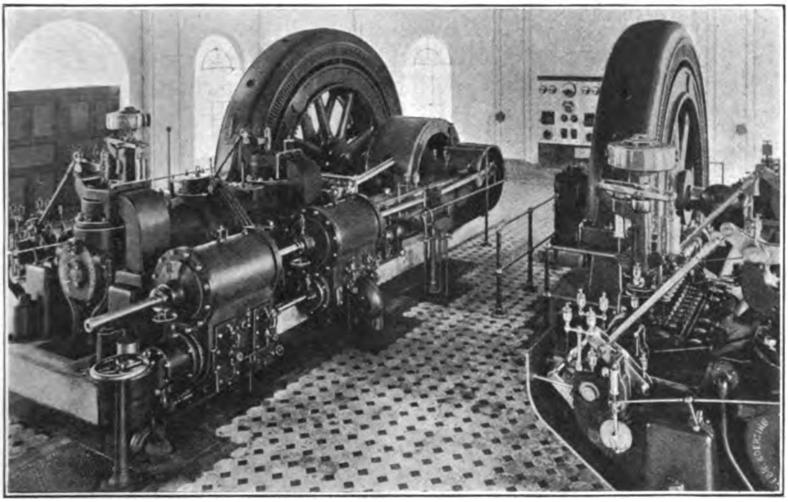


FIG. IV.—27. Koerting 300 H.P. double-acting, two-cycle Engine, using gas from lignite.

Refrigerating Machine Co., of New York, for the Lakawana Iron and Steel Co., of Buffalo, N.Y., and consisted of ten double-cylinder

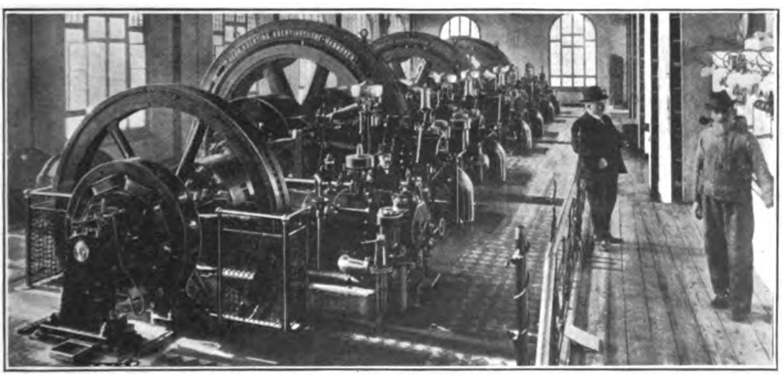


FIG. IV.—28. Six 300 H.P. Koerting four-cycle, single-acting, twin-engines at Julienhütte Electricity Generating Station.

1,000 H.P. engines and dynamos for electrical service, and six 2,000 H.P. engines with blowing engines coupled to the crank shafts. This installation was at that time the largest in the world.

Several large installations have also been made for utilising lignite, which is very abundant in many parts of Europe and America. The lignite is treated in special pressure gas producers.

Fig. IV.—27 is an illustration of a 300 H.P. double acting two-cycle engine fed with gas produced from lignite. Fig. IV.—28 represents a central electric station at Julienhütte, Germany, where coke oven gas is used. This consists of six four-cycle engines, single-acting, of 300 H.P. of the twin type with fly-wheel armature.

The installations of Koerting two-cycle engines have been carried out by the parent company and by the undermentioned licensees, and comprise 214 engines aggregating 182,240 H.P.

Siegener Maschinenbau Akt. Ges. Vorm. A. & H. Oechelhäuser, Siegen, Germany.

Maschinenbau Akt. Ges. Vorm. Gebr. Klein, Dahlbruch, Germany.

Donnersmarckhütte Oberschlesische Eisen und Kohlenwerke Akt. Ges., Zabrze.

Gutehoffnungshütte Aktien-Verein für bergbau und Hüttenbetrieb, Oberhausen, Rheinland.

Erste Brünner Maschinenfabrik Bes. Brünner, Oestenreich.

Ganz & Tarza, Budapest.

The De la Vergne Refrigerating Machine Co., New York.

Mather & Platt, Ltd., Salford Iron Works, Manchester.

Fraser & Chalmers, Ltd., Erith, London.

Leflaive & Co., Saint-Etienne, France.

Maschinenfabrik Augsburg Nürnberg, Nürnberg. — This firm has always been engaged in the construction of powerful steam engines, and has gained well merited renown in this connection. The experience and practice thus acquired, and the use of large and improved plant, has enabled them to build large gas engines under the best conditions.

They quickly recognised that the employment of single-acting gas engines of increased dimensions and multiple-cylinders was only a temporary solution for the provision of engines of sufficient power to supply modern requirements. The space occupied, the enormous weight and low efficiency of this type, were so many defects, that, above a certain power, it was abandoned in favour of double-acting engines. However, for small engines up to 175 H.P. the Nürnberg Co. still build the single-acting type, and when required for even double this power, by placing two cylinders side by side. The photograph reproduced in Fig. IV.—29, is of a single-acting engine of 60 H.P., the construction of which was first begun in 1899. From the general appearance, strong, simple and yet massive, design, the characteristic features of German construction will be recognised.

The single-acting Nürnberg engine has been the outcome of many important improvements, but it is in connection with the construction of double-acting engines that this firm excels. These engines are built up with one, two, or four cylinders according to the power required. In one cylinder the output reaches 1,500 H.P.; in two cylinders, 3,000 H.P.; in four, up to 6,000 H.P.; the latter engine

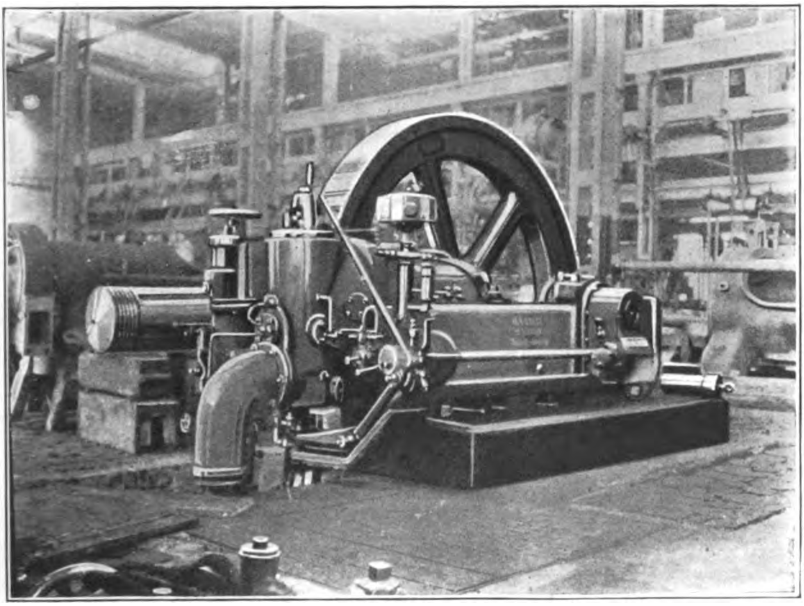


FIG. IV.—29. Nürnberg 60 H.P. single-acting, four-cycle Engine.

being two tandem engines arranged side by side on the same crank shaft.

The Nürnberg Co. has supplied an installation of 22,500 H.P. to the Rombach Steel Works for electrical service and for the blowers. A 1,000 H.P. tandem engine in this installation is shown in Fig. IV.—30. Fig. IV.—31 refers to two twin engines, each of 1,100 H.P., running at 100 revolutions per minute and fed with blast furnace gas, for generating continuous current for the electrical service of the Micheville Steel Works (France).

Amongst the largest engines mention may be made of the 3,600 H.P. twin-tandem engine forming a portion of the group of 9,100 H.P. at the

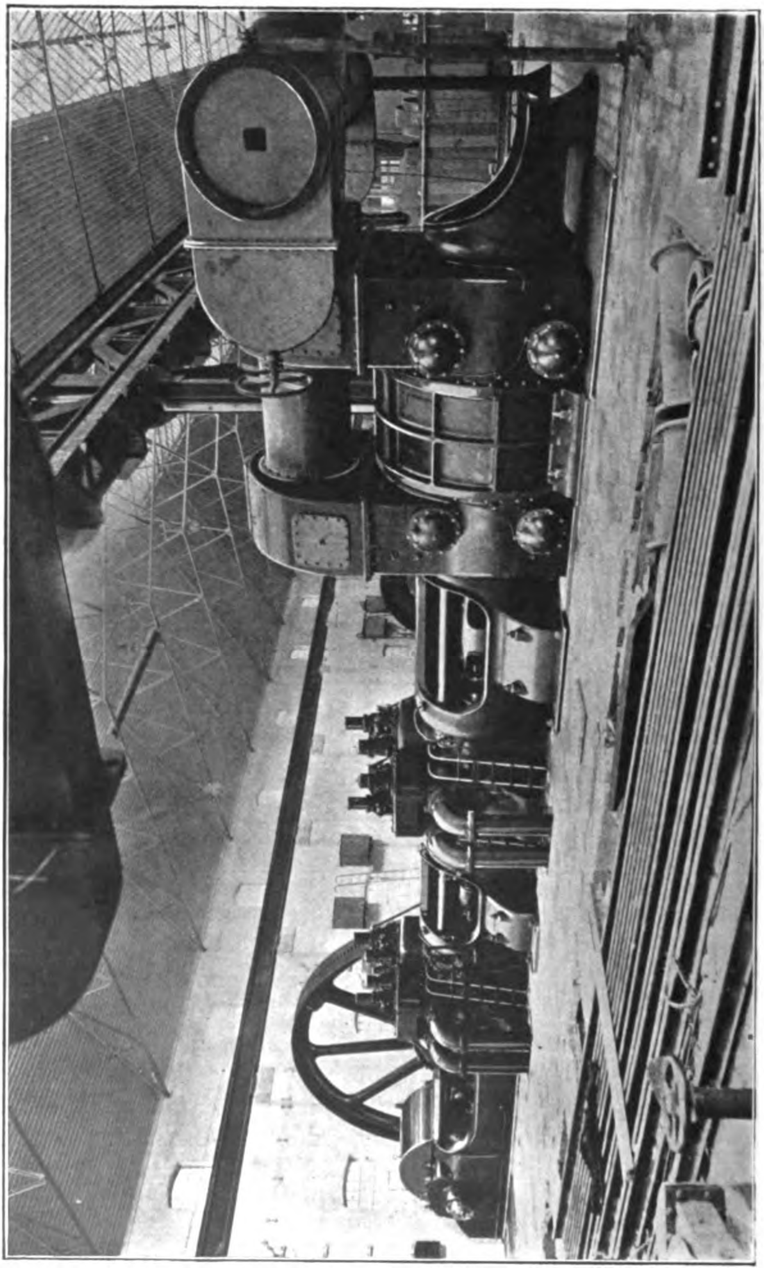


FIG. IV.—30. 1,000 H.P. Nürnberg tandem blowing Engine at Rombach Steel Works.

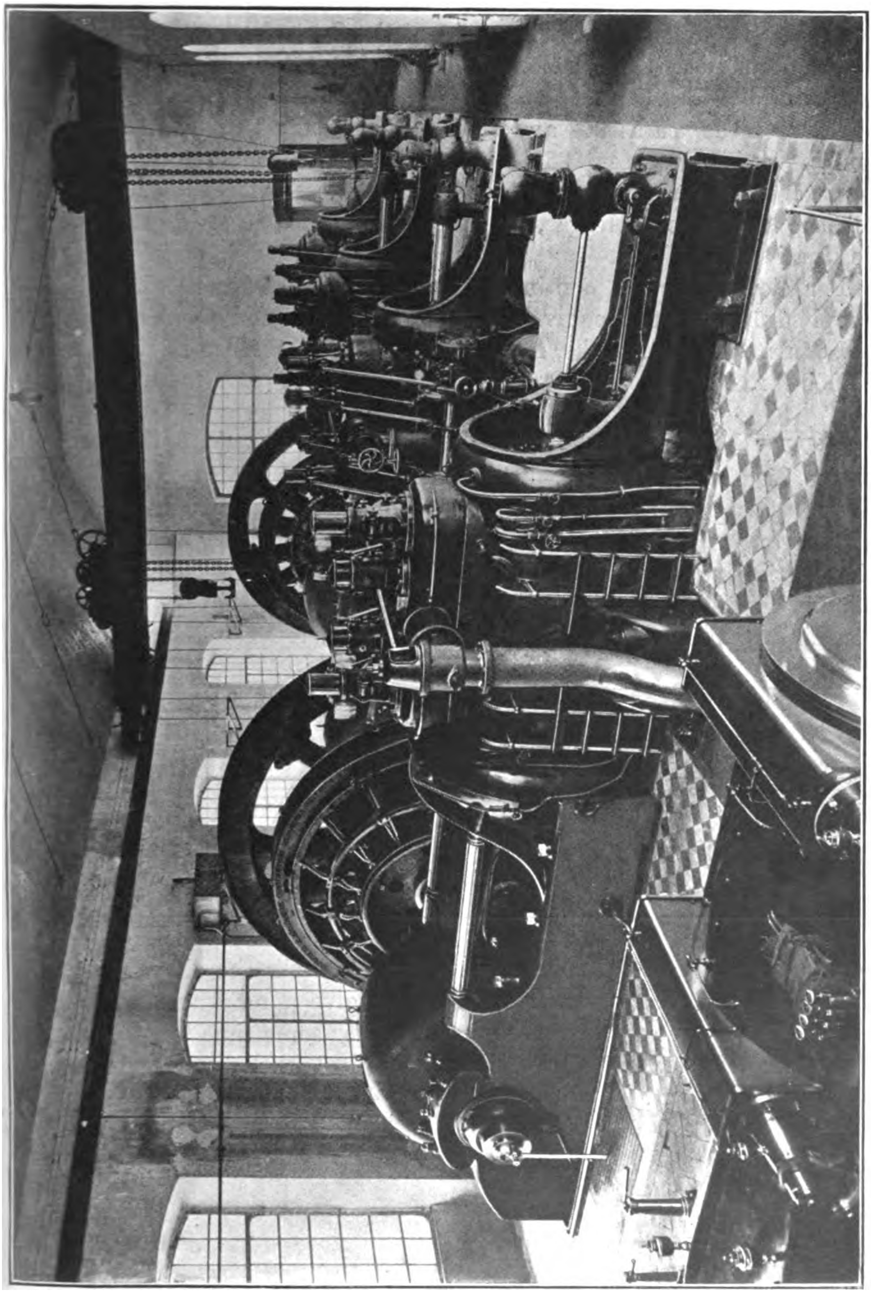


Fig. IV.—31. Two 1.100 H.P. twin-cylinder Nürnberg Engines and Dynamos at Micheville Steel Works.

works of Schalker-Grüber & Hüttenverein, Gelsenkirchen. Also the installation of 12,000 H.P. in six units at the Central Electricity Works of the Sociétad de Gasificacion Industrial, Madrid, served by Duff gas producers.

Between the years 1903 and 1907 the Nürnberg Co. have supplied, and have in progress, 239 engines of their double-acting type, representing a total of about 300,000 H.P., served by either blast furnace gas, coke oven gas, or by producer gas, according to their destination.

For electric service, continuous or alternating current, they have supplied 187 engines, equivalent to 216,000 H.P., and for blowers in blast furnaces, steel works, rolling mills, &c., 52 engines representing 84,000 H.P.

Siegener Maschinenbau Aktien Gesellschaft. Vorm. A. & H. Oechelhäuser, Siegen (Westphalia).—This firm make, in addition to the Siegen-Koerting engine, steam engines, blowing engines, compressors, &c. The arrangement of details, and particularly of the valves, ensures their engines working with imperfectly purified gas, and even with gas coming from a blast furnace the engines are capable of working for several days. The current types are as follows :—

Type.	Stroke.	Revolutions per Minute.		Output B.H.P.	
		Blowing Engines.	Electric Service.	Normal.	Maximum.
	Inches.				
Light .	43·3	100	107	500/600	630/770
Heavy .	43·3	100	107	560/700	720/900
One .	49·2	90	95	800/970	1000/1250
Three .	55·12	85	90	900/1150	1800/2300

The twin double-cylinder engines have double the above powers. Thus a twin double-cylinder engine may give a maximum of 4,600 B.H.P. The engine speed can be decreased for some time to 24 revolutions per minute, and the engines can deal with an overload of 35 to 40 per cent. Up to the present 47 engines have been built, representing 50,000 H.P.

CHAPTER V

HORIZONTAL GAS ENGINES

It was originally in Germany and then in England that the construction of gas engines made the most rapid progress. America afterwards was the country that produced most. Each construction has a peculiarity—some feature in its design—which reveals its country of origin.

The German engine is characterised by its robust appearance and its massive under-frame (Fig. V.—1). It always presents the appearance of an extremely well-finished piece of engineering work, as all the working parts that can be machined are habitually polished, showing that the makers take a real pride in making their engine a thing of beauty. The price, naturally, is increased thereby, but the working life of these engines is sensibly augmented by it.

As an instance of longevity it may be remarked that certain engines of the Otto-Langen rack type have been in constant service for nearly forty years.

English builders have adopted a different point of view,—that of building engines cheaply and in numbers, for the most part of small industrial sizes. These engines opened a fertile field of study and experience suitable for leading the makers forward, with advantage, to the construction of larger sizes, but it must be said that few makers have profited by the data thus placed within their reach.

English engines intended for service with illuminating, or town, gas, are, in general, characterised by the cylinder being overhung, as shown in Fig. V.—2. They compete in ingenuity in so arranging the working parts that the mechanical devices are effective and simple. The valve movements and method of control are generally obtained by cams and positively moved levers. The governor itself is reduced to its simplest form, for in "hit and miss" governing, which is a feature of the majority of English engines, its duty is merely to displace, or to cause a slight deviation of a small piece of metal which is interposed or not between the point of contact of the governor lever and the gas valve spindle, according as the necessity arises for the latter to open or remain closed.

One cannot be but astonished that English makers have permitted

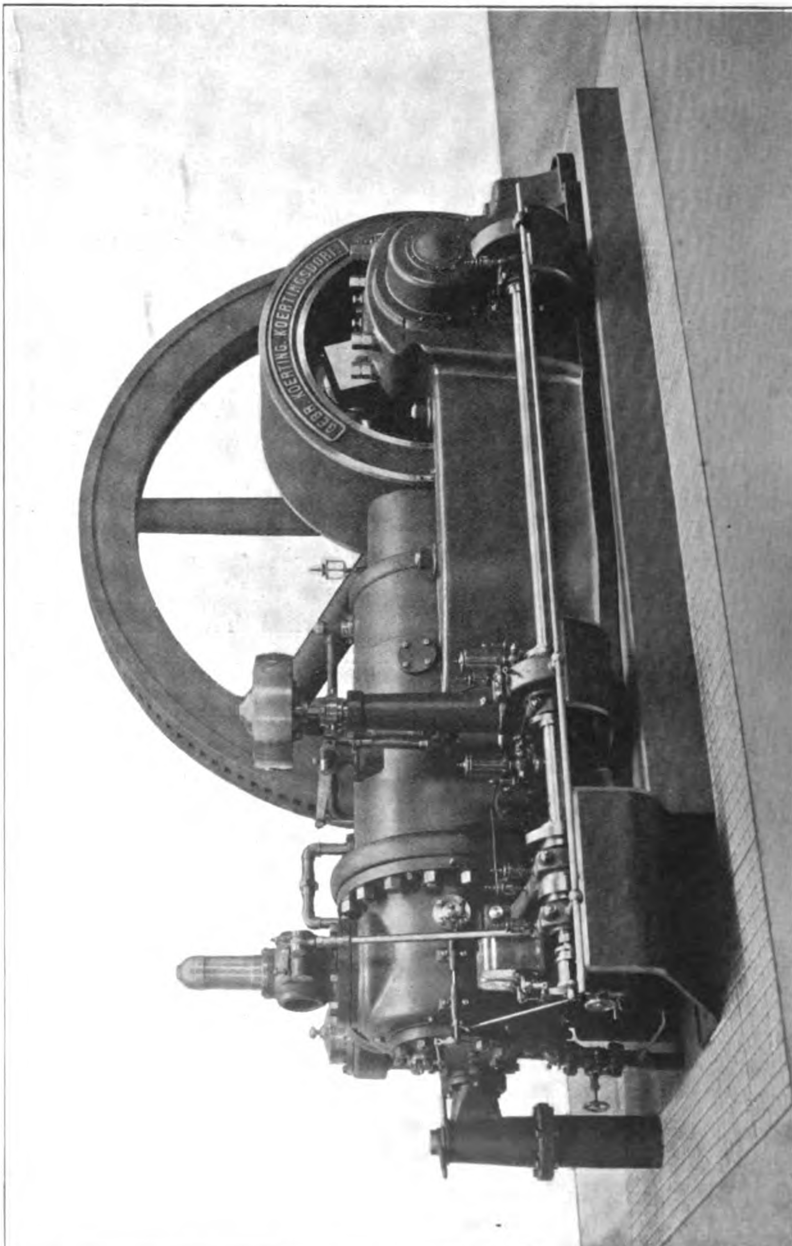


FIG. V.—1. Gas Engine typical of German construction.

themselves to be out-paced by the Germans in matters concerning the design of modern and improved types of engines and in the building of engines in large units. Mr. H. A. Humphrey, in his treatise upon large gas engines, recognised the fact and attributed it chiefly to the feelings of prudence on the part of his compatriots who have not desired to make adventitious experiments in this matter. Continental builders have not been possessed by the same fear, and success has crowned their efforts.

American constructors have been induced, by the peculiar economical

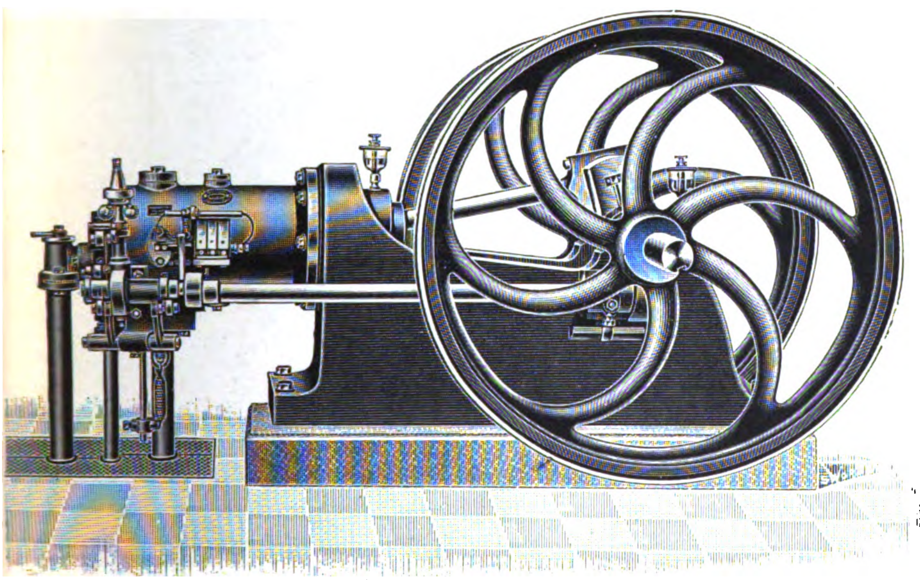


FIG. V.—2. Gas Engine typical of British construction.

conditions of their country, to establish types of gas engines which are more in accord with English than with German practice. The prime considerations which have, until now, outweighed all others in America, as far as working is concerned, are simplicity and ease of manipulation and maintenance. Users pay but little heed to the consumption whether of natural gas, benzine, or petroleum. They desire, before everything else, an engine which has but few complications, and which may be placed under the care of any kind of workman who may have only extremely vague notions about engine mechanism, if, indeed, he has any at all. Makers have been compelled to build

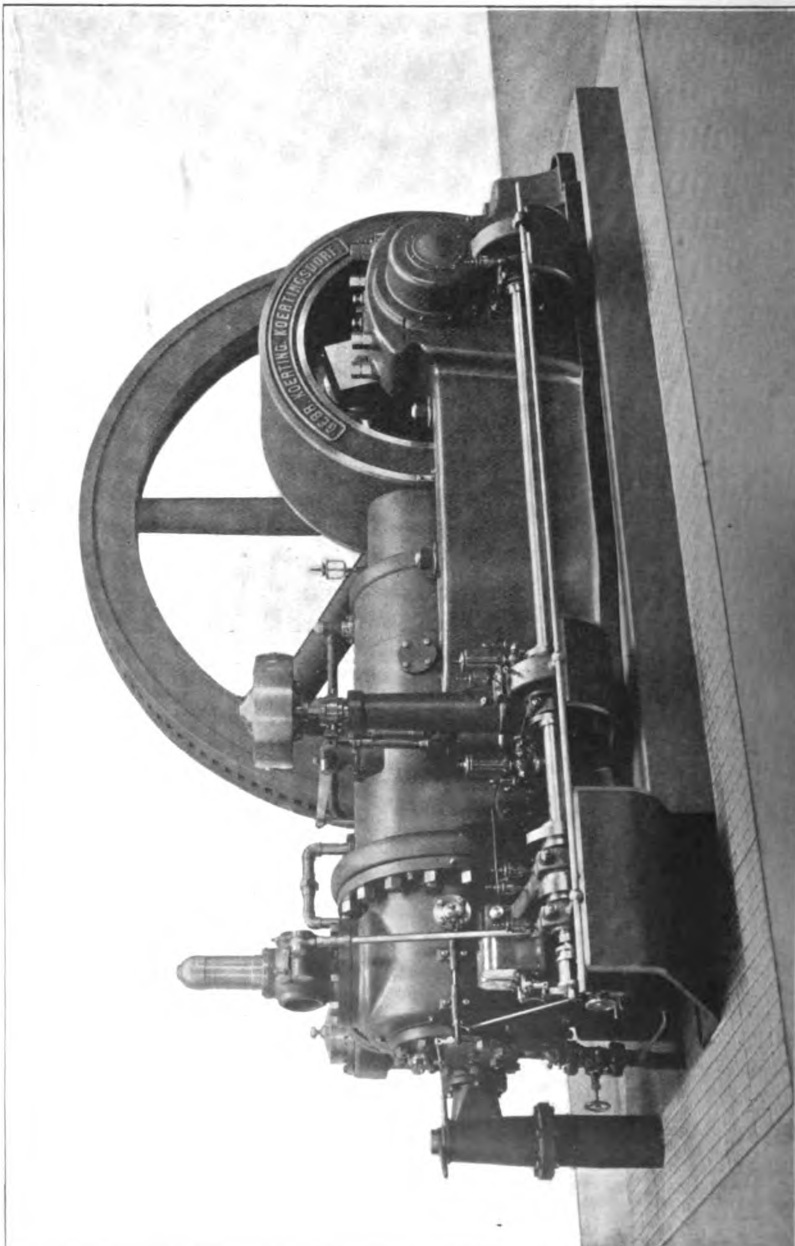


Fig. V. 1. Gas Engine typical of German construction.

themselves to be out-paced by [illegible] in the [illegible] design of modern and improved types of engines and in the [illegible] of engines in large units. Mr. H. A. [illegible] large gas engines, recognised [illegible] feelings of prudence on the part of [illegible] desired to make adventurous experiments in [illegible] builders have not been [illegible] crowned their efforts.

American constructors have been [illegible]

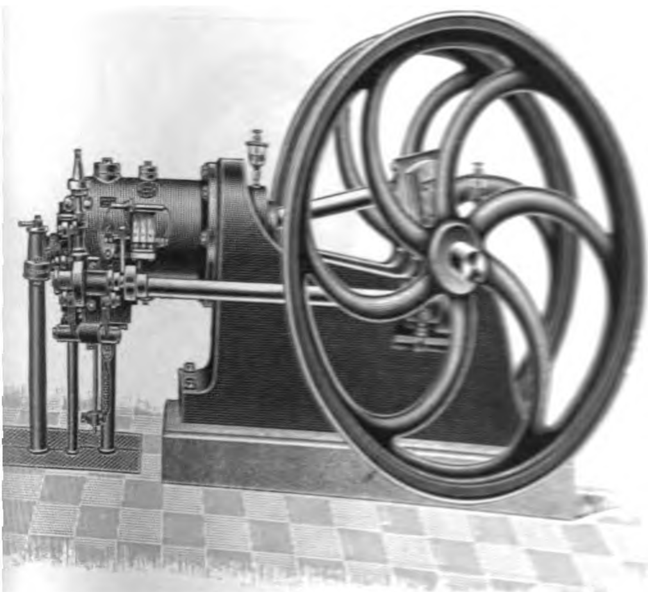


FIG. V.—2. Gas Engine [illegible]

conditions of their country, to establish types of gas engines which are more in accord with English than with German [illegible]. The prime considerations which have, until now, [illegible] in America, as far as working is concerned, are [illegible] of manipulation and maintenance. Users pay [illegible] consumption whether of natural gas, [illegible] desire, before everything else, an engine which [illegible] tions, and which may be placed under the care of any kind of workman who may have only extremely vague notions about engine mechanism, if, indeed, he has any at all. Makers have been [illegible]

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their engines in conformity to this desire and have standardised their types on the general lines of the example shown in Fig. V.—3.

Several makers, however, have realised the necessity of following the methods adopted in Europe, and particularly for the construction of large engines. The Allis-Chalmers Co., Bessemer Gas Engine Co., Olds Gas Power Co., Westinghouse Co., and Snow Steam Pump Co.'s engines are all of modern type, some being constructed with important simplifications, such as, for instance, the use of a side crank in place of the usual central double web crank.

It is unnecessary to dwell upon the Swiss type, as this is based

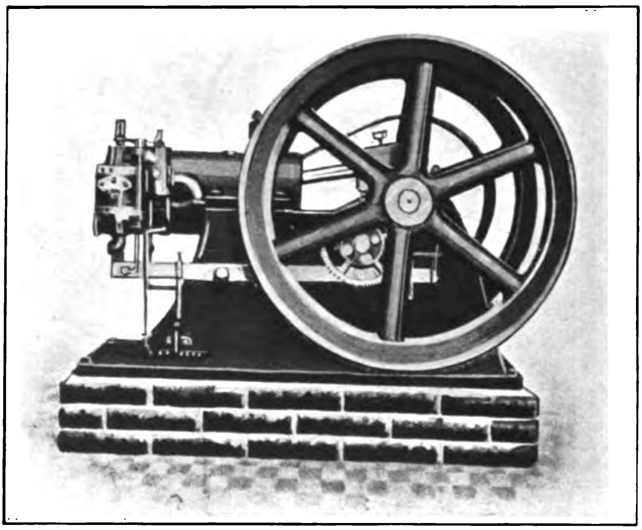

FIG. V.—3. Gas Engine typical of American construction.

upon the German designs. Nor do the French and Belgian engines call for remark, for, speaking generally, these appear to have lagged behind the trend of progress in other countries.

In the author's opinion the German type seems likely to preponderate. It has been recently adopted by several large English makers, including Kynoch, Robey, Tangye, Stewart, Stockport, &c. The new design of Campbell engine is also similar. In America, the Olds Gas Power Co., Bessemer Gas Engine Co., Muenzel Co., Struthers-Wells, and other firms have brought out new engines upon the same model.

Various Arrangements of Cylinders.—Up to 100 H.P. producer gas engines are built usually with one single-acting cylinder. In installations of 150 to 200 H.P. the same system applies, but rather more rarely, because, as much from the point of view of regularity as for reliability of working, it is preferable for the same output to employ engines with two cylinders.

Reduction of weight, especially as regards fly-wheel and frame, gives preference to an engine with two cylinders over one with a single-cylinder for corresponding powers. Some makers sell a "double" engine at a lower price than for a "single," for powers ranging between 100 and 150 H.P. In all respects, therefore, it is rational to give preference to double-cylinder engines.

Two cylinder engines, either twin or tandem, present a very appreciable advantage when installed in new factories, where, at the start, the total capacity of the plant is not required. One cylinder only can be used at the outset, and will work under the best conditions as to efficiency, the second cylinder being coupled up at a later period when full power is wanted.

Above 300 H.P. a single-cylinder double-acting engine is generally considered the most suitable. The weight is reduced to 300 to 400 lbs. per H.P., and the space needed is considerably less than for single-acting engines.

In Europe, the two-cylinder "twin" engine soon displaced the type with opposed or *vis-à-vis* cylinders, because the latter gave a great deal of trouble in practical work. The crank heads working on the crank-pin and the main bearings appeared to suffer more particularly. Besides which, in the same direction of rotation, one of the pistons, during the power stroke, produced a reaction normally downwards in one cylinder while the second piston caused a reaction upwards in the other cylinder, thus giving rise to shocks which became accentuated with wear.

The double-acting type is generally constructed with one cylinder up to 400 to 500 H.P., and even as far as 800 to 1,000 H.P. For the largest engines, from 3,000 to 4,000 H.P., it is then sufficient to arrange for four twin-tandem cylinders. Nevertheless, it is unusual to find an engine of 800 to 1,000 H.P. with one double-acting cylinder. It is preferable for consideration of weight and regularity to make use of two twin cylinders, or, better still, two tandem cylinders.

Figs. V.—4, 5, and 6, show the various arrangements of cylinders both for four-cycle and two-cycle engines with the corresponding power stroke diagrams for each. These diagrams indicate the impulses of the cycle under consideration, as would be represented by the records obtained by means of indicators, and developed in the order of the succession of the impulses.

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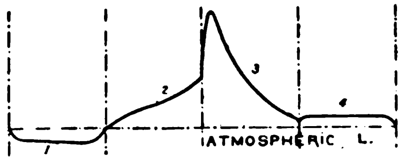
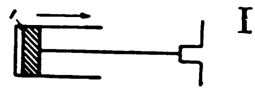
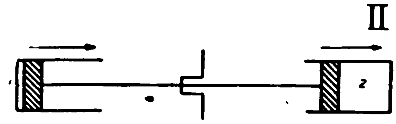


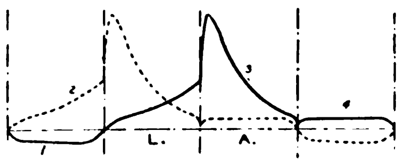


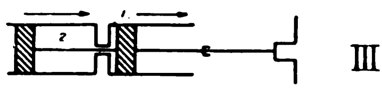
One cylinder, single-acting.

SECTION B.

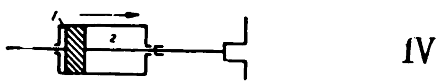


Two cylinders, *vis-à-vis*, single-acting.



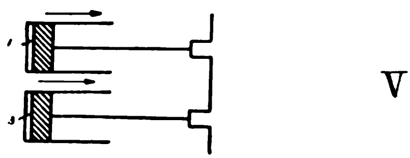


Two cylinders, single-acting, coupled (Dingler).

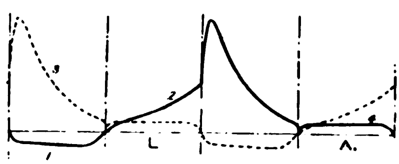


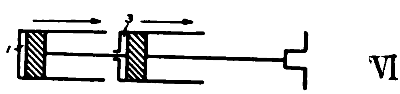
One cylinder, double-acting.

SECTION C.

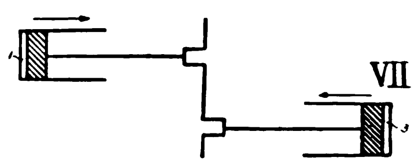


Two cylinders, twin, single-acting.





Two cylinders, tandem, single-acting.



Two cylinders, single-acting, opposed on separate cranks (180°).

FIG. V.—4. Characteristic features of various cylinder combinations with relative frequency of impulses (four-cycle engines).

For multiple-cylinder engines, the same diagram has been made to include the cycle of each power stroke, so that the different impulses occurring during the same period and for one complete cycle can be differentiated.

The different arrangements of cylinders that realise the same advantage as regards regularity are batched together.

In Fig. V.—4, Chart I., section A, corresponds to a single-acting, four-cycle engine, the different strokes being indicated on the diagram: (1) suction, (2) compression, (3) explosion and expansion, (4) exhaust. The engine piston is represented as being in the position corresponding to the commencement of the first stroke. One explosion is produced for two revolutions of the fly-wheel.

Section B, Charts II., III., and IV. of Fig. V.—4, correspond to three arrangements of double-acting engines. On the diagram the full line shows the cycle of one of the power strokes, and the dotted line that of the second. It will be noticed that, in this case, two explosions occur in succeeding strokes, and are separated from the two following explosions by two resistance strokes, namely: (*a*) suction, dotted line, and exhaust, full line, marked (4); and (*b*) suction, full line, marked (1) and compression, dotted line, marked (2). There are, therefore, two successive explosions in one revolution of the fly-wheel, and two negative strokes in the following revolution. The pistons are shown in the position for making the strokes 1 and 2 (suction and compression, respectively).

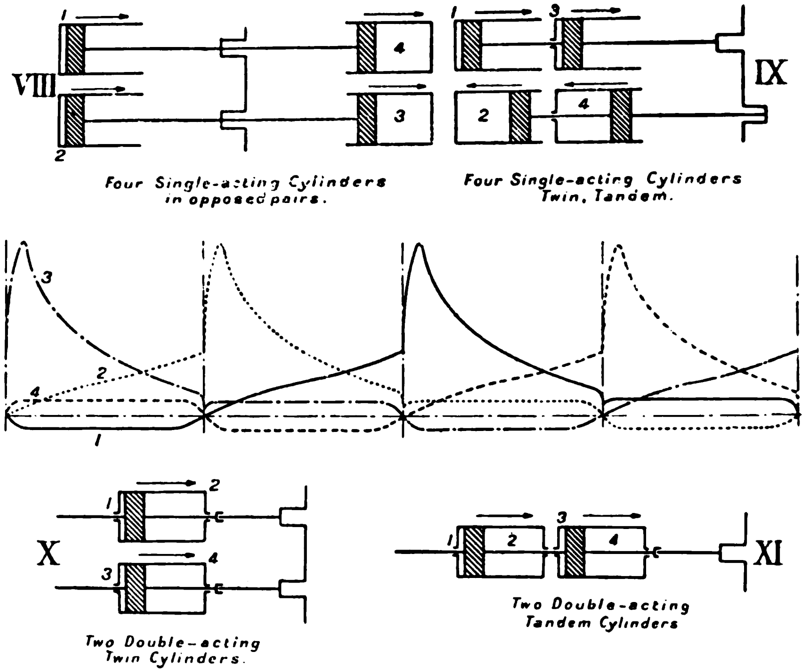
From these examples it will be seen that a single-cylinder double-acting engine has no advantage, as far as cyclic regularity is concerned, over that with two opposed single-acting cylinders on one crank.

Section C, Charts V., VI., and VII., Fig. V.—4, also shows three arrangements of double-acting engines, but in these each explosion is separated from the next by one resistance stroke. An explosion is thus obtained for each revolution of the fly-wheel, and, therefore, the regularity is greater than in the preceding section.

The twin engine, by this comparison, possesses the same characteristic as the tandem engine. But the latter has greater advantages from a constructional point of view, because it is simpler, and involves but one crank shaft, one connecting-rod, and one side shaft. On the other hand, it necessitates a stuffing-box at the back of the foremost cylinder.

In the twin arrangement (V.) it is customary for one engine to be left-handed and the other right-handed, so as to be symmetrical, with one fly-wheel placed between the two cylinders common to both. To

SECTION D.



SECTION E.

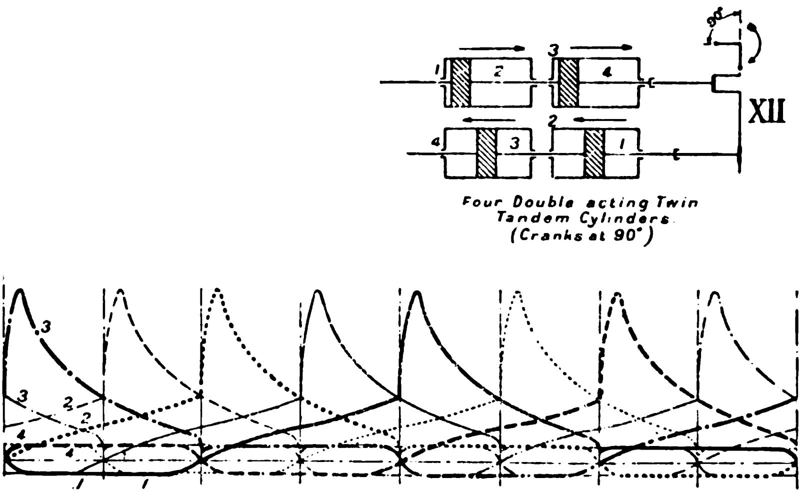


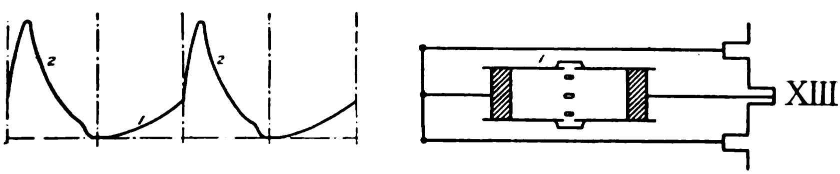
FIG. V.-5. Characteristic features of various cylinder combinations with relative frequency of impulses (four-cycle engines).

facilitate supervision it is a good plan to place the side shafts and the valve gearing towards the interior, but in this case, to permit erection, the gear-wheels driving the side shafts, which are fixed on the crank shaft, should be in two pieces, as well as the fly-wheel and the pulley. The governor is fitted on one of the cylinders, and, by means of a rod, controls the other cylinder also.

The arrangement shown in Chart VII. is very cumbersome and for this reason is very rarely adopted.

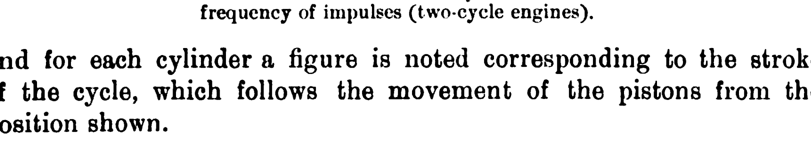
In section D, Charts VIII., IX., X. and XI., Fig. V.—5, an impulse is given on each stroke of the cycle, that is to say, at each half-revolution of the fly-wheel. The four cycles are depicted in the diagram,

SECTION F.



One cylinder, single-acting, two-cycle (Oechelhäuser).

SECTION G.



One cylinder, double-acting, two-cycle (Koerting).

FIG. V.—6. Characteristic features of various cylinder combinations with relative frequency of impulses (two-cycle engines).

and for each cylinder a figure is noted corresponding to the stroke of the cycle, which follows the movement of the pistons from the position shown.

The arrangement VIII. has been abandoned. There was no advantage in setting the cranks at 180° from the point of view of balancing the moving masses, except that it modified the sequence of the strokes of the cycles.

In the Chart IX., the regular sequence of strokes for the diagram represented, demands the fixing of the two cranks at 180°. These two arrangements of single-acting cylinders are cumbersome, complicated, and costly to construct. Preference, therefore, must be given to the double-acting arrangement.

With regard to Charts X. and XI., the latter is preferable if sufficient

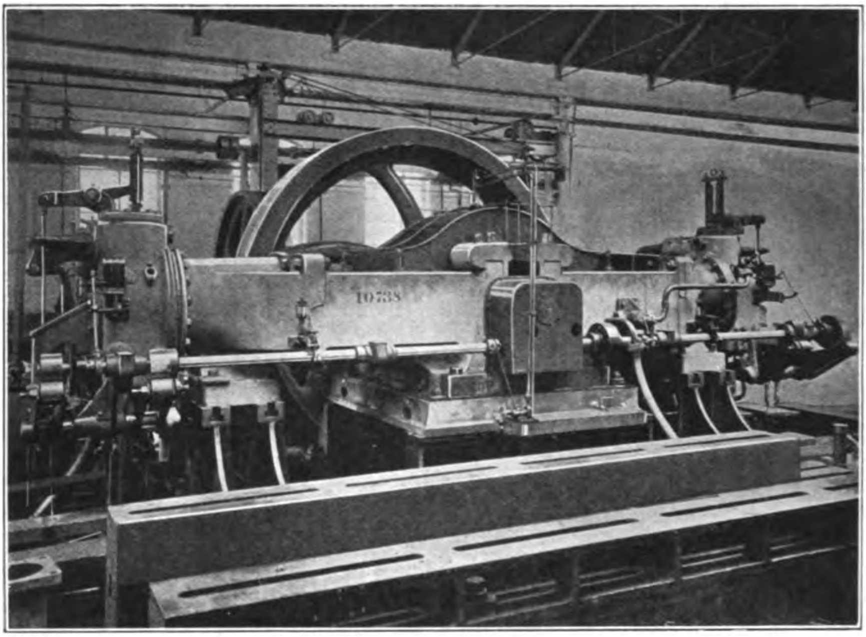


FIG. V.—7. Two-cylinder, *vis-à-vis*, single-acting, four-cycle Hornsby-Stockport Engine.

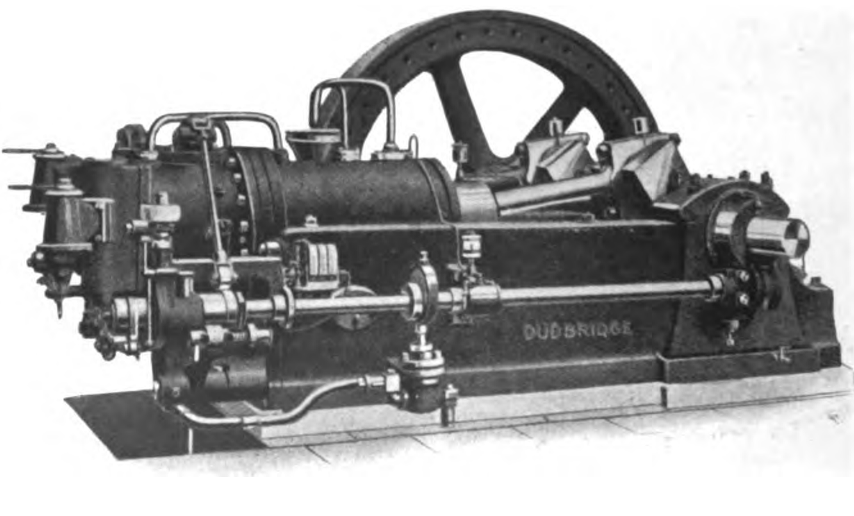


FIG. V.—8. Two-cylinder, twin, single-acting, four-cycle Dudbridge Engine.

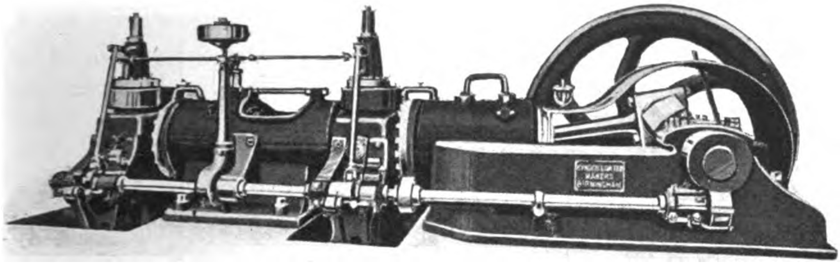


FIG. V.—9. Two-cylinder, tandem, single-acting, four-cycle Kynoch Engine.

length be available within the engine room, for the same reasons as those detailed in connection with Charts V. and VI.

Section E of Fig. V.—5 is scarcely ever applied except for powers exceeding 2,000 H.P. It has one evident advantage from the point of view of regularity, for, with the cranks fixed at 90°, it gives one explosion for every quarter-revolution of the fly-wheel.

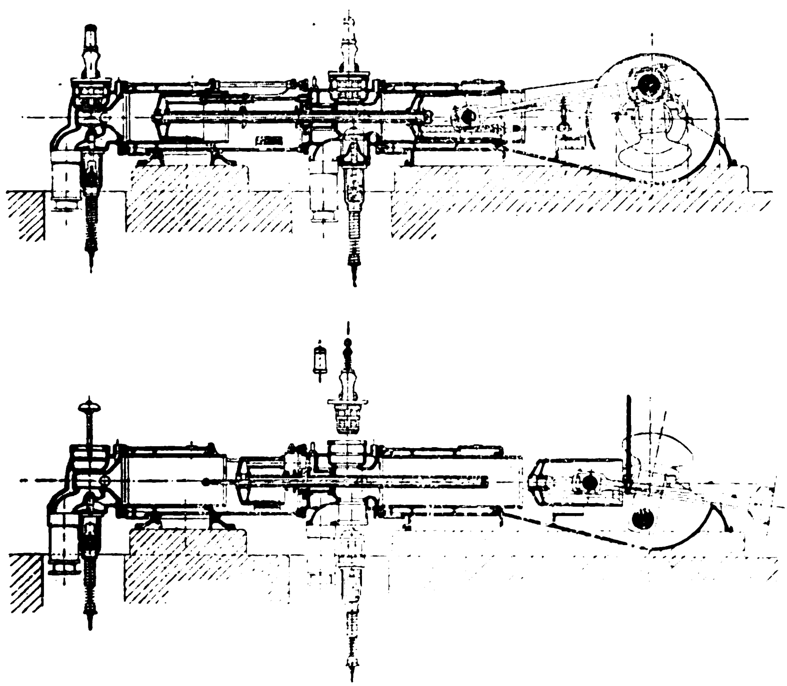


Fig. V.—10. Two-cylinder, tandem, single-acting, four-cycle Winterthur Engine.

COMPARATIVE WEIGHTS AND POWERS OF LARGE GAS ENGINES.

Description of Engine.	B.H.P.	Number of Cylinders.	Diameter of Cylinder.	Stroke of Piston.	Weight of Flywheel.	Total Weight, Engine and Flywheel.	Revolutions per Minute.	Cyclic Regularity.	Total Weight per B.H.P.
			Inches.	Inches.	Tons.	Tons.			Lbs.
Cockerill four-cycle .	600	1	51·18	55·12	32·47	125·0	90	$\frac{1}{30}$	466·5
Deutz " " .	600	4	26·0	33·5	11·81	98·4	150	$\frac{1}{130}$	367·3
" " " .	1,200 I.H.P.	4	33·0	41·0	19·0	219·0	135	—	408·8 I.H.P.
Soest " " .	350	2	25·6	33·5	26·0	76·0	140	—	486·4
Koerting (d.-a.), two-cycle .	550	1	25·0	43·3	11·81	68·8	100	$\frac{1}{80}$	280·5
" " " " .	600	1	27·56	47·24	15·24	90·0	95	$\frac{1}{160}$	335·7
" " " " .	500	1	25·59	44·0	12·3	70·0	100	$\frac{1}{160}$	313·3
Thornycroft vert. petroleum	120	4	8·0	8·0	—	1·24	900	—	23·25

Sections F and G, Fig. V.—6, correspond to the two-cycle engines, Oechelhäuser and Koerting, respectively.

The Oechelhäuser engine has a single cylinder giving one explosion every revolution of the fly-wheel, whilst the Koerting engine (double-acting) gives two. The latter, as regards regularity, is therefore similar to those of the section D. With the Oechelhäuser engine, four cylinders coupled and opposed must be employed. With the Koerting engine its multiple cylinders would be coupled or arranged in tandem.

In Chapter XIII., coefficients are given for the calculation of the weights of fly-wheels in order to obtain a suitable cyclic regularity for each of the Charts I. to XIV.

In Fig. V.—7 is illustrated a type of *vis-à-vis* Hornsby-Stockport engine. The Dudbridge Ironworks Co. arrange their engines with two cylinders side by side upon the same base as shown in Fig. V.—8 in which is represented a 140 H.P. engine with the fly-wheel placed on the end of the crankshaft. Several German firms build their single-acting engines up to 500 H.P. on a similar plan, combining two groups of "twin" cylinders (four pistons) with the fly-wheel between.

Kynoch, Ltd., of Birmingham, build single-acting tandem engines of 200 H.P. and over, with the cylinder arranged as shown in Fig. V.—9.

The Winterthur Co. have brought out a tandem type of single-acting engine, 300 H.P., shown in Fig. V.—10. The lower cut shows how the pistons, connecting rod, &c., are removed.

The Wellmann-Seaver-Morgan Co., of Cleveland, U.S.A., build a tandem engine, double-acting from 100 H.P., upon the Sargent principle.

The figures on p. 78 are given by Mr. Horace Allen in his book "Gas and Oil Engines," p. 198, and it is interesting to note the great variation in total weight per B.H.P. for the engines there referred to.

CHAPTER VI

VERTICAL GAS ENGINES

The application of vertical gas engines hitherto has been somewhat restricted. Indeed, it is only in America that this form has been adopted to any great extent. That this should be so is not remarkable when one considers that, with a vertical engine, the space required is reduced to a minimum, and that, in view of the cost of building sites in American towns, such consideration is there of the greatest importance.

Until recent years gas engines in the United States of America were ordinarily served with either natural gas or town gas, except when very large units were installed, when they were fed with pressure producer gas or by blast furnace gas. It seems that it is owing to this fact that American gas engine builders have not experienced the serious troubles that are sometimes occasioned by vertical engines when fed with power gas, such as is produced from suction gas-producers using lean or non-bituminous coals.

Suction gas plants are becoming more and more in favour by reason of their advantages, and owing to the increasing demand for this type of installation a large number of new firms have commenced to build gas engines and producers. Several of these firms have hesitated when making a decision as to the type of machine they should construct, finding it difficult to make up their minds whether to build horizontal or vertical engines. In the result, the majority have made their decision in favour of the horizontal type. Many firms have built vertical gas engines as well, and after making tests and noticing their daily operation in the works where they have been installed, have apparently arrived at the conclusion that the vertical engine, except in certain cases, is unable to show any advantage when compared with horizontal types for stationary purposes, when the total output exceeds from 25 to 30 H.P. per cylinder.

Amongst English firms who have for several years made a speciality of vertical type engines are :— British Westinghouse Co., Ltd. ; Campbell's Gas Engine Co.; Crossley Brothers, Ltd. ; Fielding & Platt, Ltd.; L. Gardner & Sons; Hindley & Sons, Ltd., and Tangyes Ltd.

In America, the Westinghouse Co. have specialised upon vertical engines, although they have now started to build horizontal engines also. In Germany and Switzerland only a few firms build vertical engines, either exclusively or in addition to horizontal types. Of these MM. Bachtold et Cie. ; Engelhardt et Cie.; Güldner Motoren Gesellschaft; Hallesche Maschinenfabrik und Eisengiesserei; Gasmotoren Fabrik Deutz; and the Winterthur Co., are the principal makers.

The Diesel, Koerting, and Banki engines are of special construction for liquid fuels.

The partisans of the horizontal type of gas engine justify their preference by the following principal reasons :—

Construction.—Vertical engines of modern type for industrial purposes with inverted cylinders are usually built with two, three, or more cylinders. They thus become complicated and expensive to build. The cam shaft demands special gearing, while the crank shaft bearings require special arrangements for taking up wear.

Examination and inspection of the working parts is much more difficult than in horizontal engines, owing to the valves, sparking plugs, &c., being placed at the top of the engine out of reach of the attendant, unless a ladder and platform are provided. The lubrication of the piston and its pin cannot be supervised.

Maintenance.—With vertical engines the maintenance of working parts in good order is a much more difficult matter, and especially with respect to the combustion chamber and the back of the piston. Indeed, to remove the piston, an operation which is absolutely necessary from time to time when producer gas is used, is usually a matter of difficulty. Producer gas is only partially cleaned, and is always liable to contain dust and tarry matters, and consequently frequent cleaning of the internal parts is essential. The piston rings are apt to become set fast, and therefore useless, occasioned to a large extent by the dirt thrown up by splash lubrication.

Lubrication.—Lubrication of the cylinder has been effected hitherto by means of the splashing of the oil contained in the crank case by the rotating crank. This method of lubrication is bad, because, instead of allowing the oil to remain still and thus to encourage the impurities to become deposited, it is, on the contrary, kept continually in motion. Clean oil is only admitted to the frictional surfaces and in the bearings when the old oil has been taken away.

Splash lubrication is still less suitable in connection with producer gas engines and of the type governed by the admission of a variable volume of mixture of constant composition, a method frequently employed by the best makers of gas engines.

In engines served by suction gas producers a partial vacuum is always produced behind the piston during the admission stroke and it necessarily follows that this vacuum is greatly increased when the engine is working under light loads. As the frictional surfaces of the cylinder always receive an excess of oil, projected by the crank, there is, at each suction stroke, a tendency for the oil to be drawn to the back of the piston. This tendency is further augmented by the relative higher pressure produced in the enclosed crank chamber by the forward stroke of the piston. This excess of oil becomes partially burnt and adheres to the interior of the walls, forming carbon deposit, which extends even to the back face of the piston, ultimately giving rise to premature ignition and back firing.

The excess of oil occasioned by splash lubrication gives greater trouble with vertical gas engines that are served by producer gas, than when gas of a higher calorific power is used, because, in the latter case, the high temperature resulting from the combustion burns and vaporises the excess of oil, up to a certain limit, without causing carbon deposits. In horizontal gas engines, if an excess of lubricating oil be accidentally permitted, its ejection from the back of the cylinder may be arranged for by means of a small blow-off valve fitted at the lowest part of the cylinder.

The author has personal knowledge of installations of vertical gas engines and producer gas plants, using anthracite fuel at a low working cost, where the amount of oil consumed costs nearly as much as the fuel, occasioned by the losses caused by splash lubrication. It can be accepted as an actual fact that, in an ordinary horizontal modern gas engine of good design, perfect lubrication can be obtained with a consumption of two or three grammes (0·125 oz.) of good oil per B.H.P. per hour.

Cooling.—With vertical gas engines it is difficult to cool the cylinder methodically, because the water enters at the lower part of the cylinder, which is the coldest portion, and rises towards the upper part, which has the highest temperature. It thus follows the reverse direction to that taken by the cooling water in horizontal engines, in which it enters at the base of the breech end and flows away from the top of the cylinder jacket. In this way the water attains a temperature sufficient to prevent an excessive quantity of heat being taken from the gas while burning and expanding in the interior of the cylinder.

In a vertical engine the circulation of water cannot be reversed in order to conform with the principle adopted with horizontal engines, because the air contained in the water or the vapour produced by the increase of temperature could not then escape. For these reasons the circulation of water in vertical cylinders is insufficient to prevent unequal expansion of metal due to excessively high temperatures produced by the high compression pressures employed in modern engines unless the consumption of water is increased.

Exhaust.—In vertical engines the exhaust being placed at the upper part of the cylinder, it necessarily follows that a greater length of pipe is required to connect up to the exhaust muffler, which is generally placed below ground. Unless water is injected into the interior of these pipes, which would greatly increase the consumption of water, this long length of exhaust pipe heats the air in the engine room and makes the engine itself less accessible.

Efficiency.—It is universally recognised, and particularly by the constructors of automobile engines, that the efficiency of engines diminishes with the number of cylinders, and that it is an error to think that by combining four cylinders, each rated at 10 H.P., for example, it is possible to obtain a 40 H.P. engine. At the most it would only be possible to obtain about 35 H.P., for the simple reason that the engine with one cylinder taken separately can be regulated experimentally from the point of view of timing of ignition, so as to obtain the maximum useful effect, whilst it is very difficult to ensure the ideal ignition point in four distinct cylinders using a distributor common to all. This is therefore one loss of efficiency, quite apart from the mechanical efficiency, properly so-called, which is itself dependent in great measure on the correct proportions of frictional surfaces and workmanship.

From the preceding remarks it will be recognised that anyone who proposes to commence the construction of industrial gas engines ought to give preference to the horizontal type, because it presents fewer difficulties.

The Güldner Gas Engine.—Güldner, who, at his works at Munich, has undertaken the construction of vertical engines, has set forth the following advantages in favour of vertical engines over the horizontal type in his excellent book. He claims that a higher mechanical efficiency can be obtained in a single-cylinder engine. The piston can be better kept gastight, the base casting can be much lighter, the foundation required for the reception of the engine is much less, while it is a much easier matter to arrange for future extensions of plant and for coupling two engines together. But Güldner also recognises the fact, which the author has already pointed out—namely, that

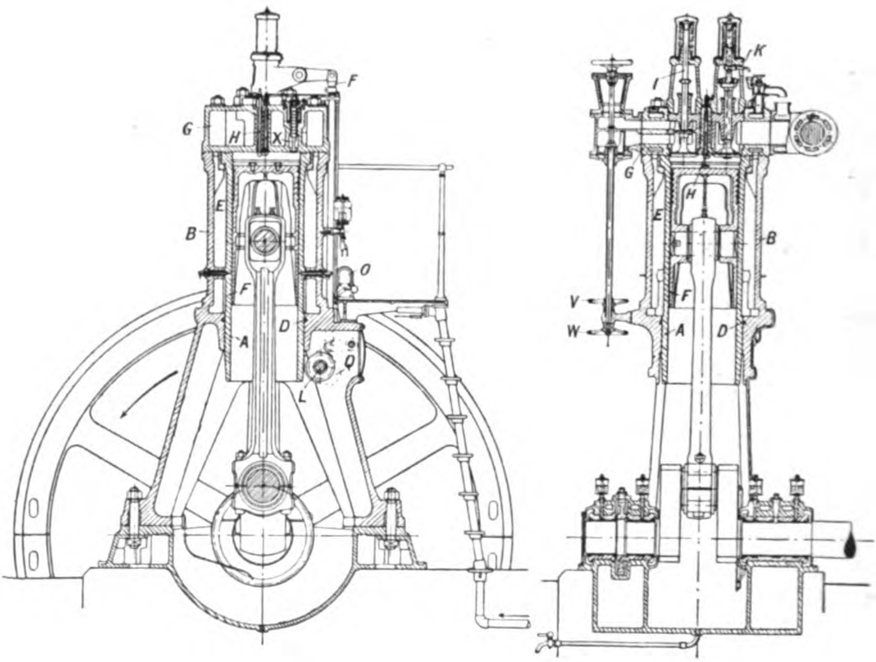


FIG. VI.—1. Sectional elevations of Güldner vertical Engine.

when imperfectly purified gas is used the horizontal type of engine is preferable.

The Güldner vertical engine with open frame must be considered the standard type of this method of construction. The inventor has designed his engine so that the details are of a simple character, and which, while being strongly built, permit of easy construction. The valves are arranged side by side on the top of the cylinder, the ignition gear being placed between them. The combustion chamber is thus arranged in a perfectly rational shape, and permits the best results to be obtained from a thermal point of view. For large engines the exhaust valve is arranged to be cooled by water under pressure. As indicated diagrammatically in Fig.VI.—1, the frame is in the form of an A, and the upper portion forms the cylinder jacket. It rests upon a cast iron bed-plate. A long piston fitted with at least six rings works within a separate liner. The water-cooled cylinder cover is flat. The valves are operated by means of a horizontal shaft placed

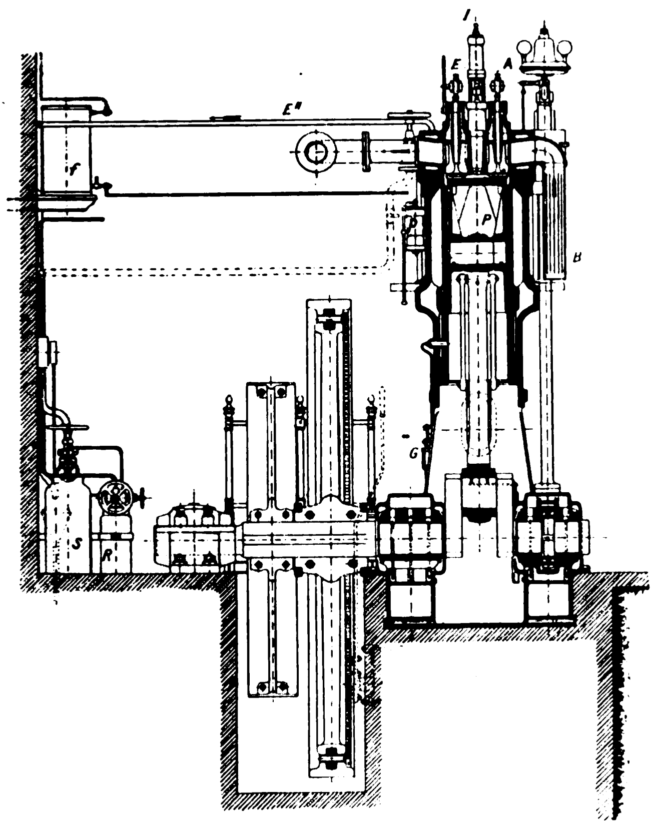


FIG. VI.—2. Sectional elevation of Diesel Engine.

about mid-height. The general appearance of the engine is as illustrated in Fig. IV.—23.

Diesel Engine.—Figs. VI.—2 and 3 represent the Diesel engine such as is constructed by the various licensees and particularly the Nürnberg Co. in their works at Augsburg, and by Messrs. Carels Frères at Ghent. This engine is classed as an internal combustion engine, which uses heavy petroleum, unrefined, and without sulphur, or, at any rate, in which the sulphur is less than 1 per cent. The manner of working is as follows :—

1st stroke.—Air is drawn into the cylinder.

2nd stroke.—The air is compressed by the backward motion of the piston.

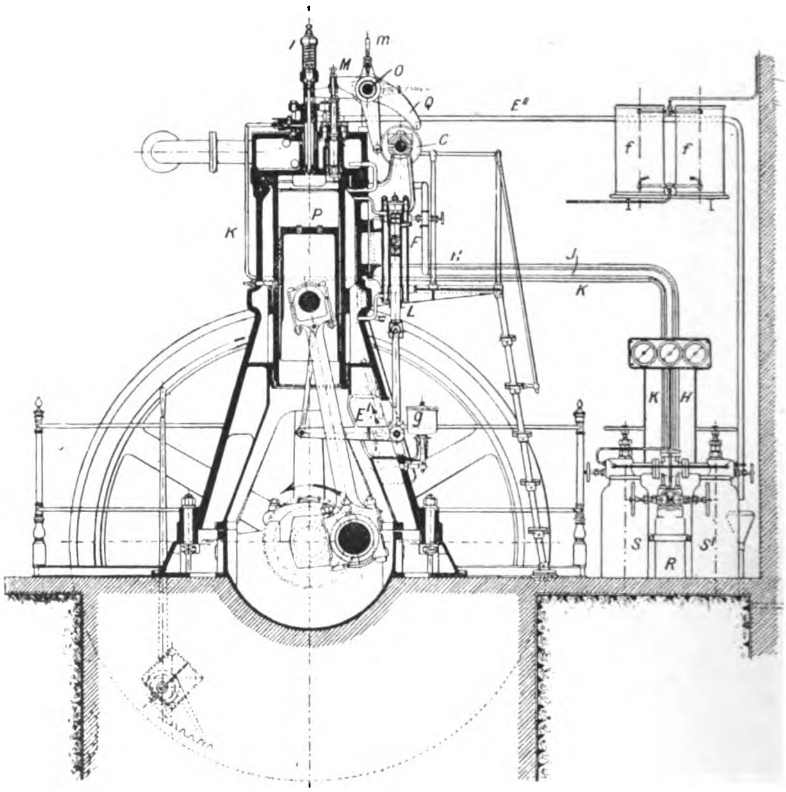


FIG. VI.—3. Transverse section (Diesel Engine).

3rd stroke.—Injection of the petroleum which immediately ignites by the heat of the compressed air.

4th stroke.—Expulsion of the products of combustion.

The valves are four in number and are arranged at the back of the cylinder, and include that used for starting by compressed air. The valves are operated by a half-speed shaft, placed at the upper portion of the cylinder, driven from the crank shaft by a vertical intermediate shaft with gear wheels.

Fig. VI.—4 shows a section of the cylinder cover. The arrangement for the injection of petroleum is represented in Fig. VI.—5. Air at

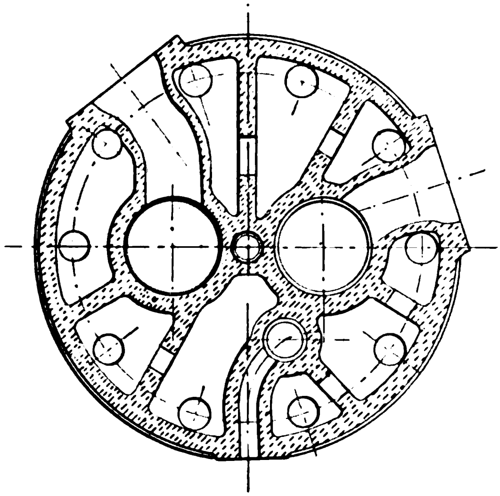


FIG. VI.—4. Section of Cylinder Cover (Diesel Engine).

high pressure is brought into an annular space concentric to the admission valve, and in the lower portion of this space the oil is delivered. The liquid upon the opening of the valve is forced through a series of perforated plates, the holes being arranged in fives and about $\frac{1}{16}$ inch in diameter. The employment of wire gauze for the purpose has not given equally good results. The exhaust valve is water-cooled.

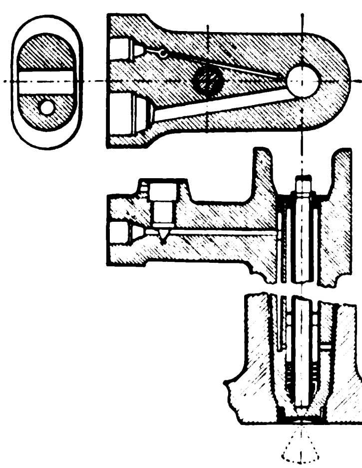


FIG. VI.—5. Section through Fuel Injection Valve Box (Diesel Engine).

The lubrication of the piston is obtained by means of oil under pressure from a series of holes arranged circumferentially in the lower part of the cylinder jacket. The lubricating oil valve is controlled in such a manner that the entry of oil takes place at the moment when the two upper rings of the piston arrive at the level of the lubricating orifices.

Engines with one, two, or three cylinders are fitted with one fuel pump. Four-cylinder engines have two. Each pump (as shown

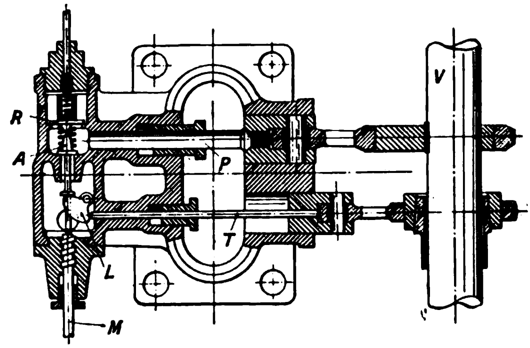


FIG. VI.—6. Section through Fuel Pump, showing automatic Governing Device (Diesel Engine).

in Fig. VI.—6) generally supplies several cylinders. It forces the fuel towards the distributing device placed at the junction of the passage,

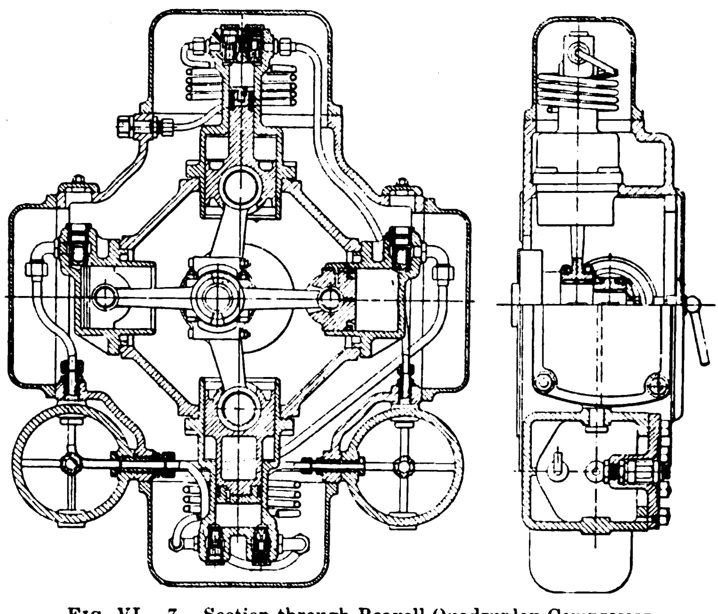


FIG. VI.—7. Section through Reavell Quadruplex Compressor.

each branch being fitted with a retaining valve and a screw which permits the amount of fuel delivered to the injection valve to be regulated.

The piston plunger (P) of the fuel pump is actuated by an eccentric fastened upon the vertical shaft (V). The suction valve (A) and retaining valve (R) are placed vertically one above the other. A second eccentric placed under the first, by means of a rod parallel to the axis of the plunger, gives an oscillating movement to the small lever (L) placed in the suction chamber of the pump body, and this lever (L), when at its highest position, keeps the suction valve open. The mechanism is arranged so that the valve remains open during the early portion of the backward movement of the piston, permitting the oil to flow back towards the suction chamber. The effective pumping portion of the stroke is determined by the instant when the suction valve is allowed to regain its seat. The governor decides whether to advance or retard the instant when this shall occur, in giving an angular displacement, by means of a vertical shaft, to the eccentric which works the small lever.

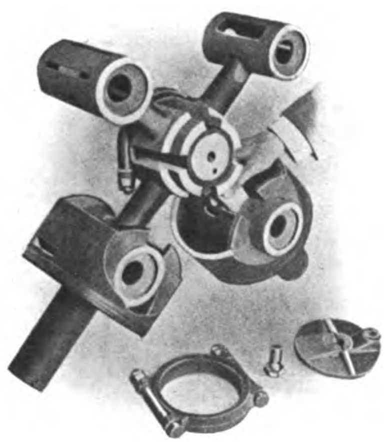


FIG. VI.—8. Pistons and Connecting Rods of Reavell Quadruplex Compressor.

In the lower casing of the fuel pump and in the axis of the valve a rod (M), fitted with a long double thread, is screwed. By means of a rod with which it is connected, furnished with two handles and with three stop notches, the screwed rod is able to receive three different adjustments, and by this means it may be fixed at three determined heights. In the lowest position it permits the valves to work freely; in the mean position, which determines the stoppage of the engine, it causes the suction valve to close, whilst in the upper position it at the same time raises the retaining valve, and permits the oil pipes to empty themselves by gravity. If the petroleum pump be not charged, an auxiliary piston worked by hand must be used, the return stroke being made by the action of the spring.

For compressing the air to high pressure, Messrs. Carels Frères employ three-stage Reavell compressors which enable them to easily obtain in continuous working, air at from 60 to 65 kgs. per square centimetre (850 to 925 lbs. per square inch) and over. This apparatus is called a quadruplex compressor. Two cylinders are utilised for simple compression, one for the second stage and the fourth for the third stage. Fig. VI.—7 shows clearly the method of construction of the machines. Tubular cooling arrangements are placed round the high-pressure cylinders to keep the air at low temperature between each degree of compression. By this means and by suitable proportioning of the diameter of the cylinder a high efficiency is obtained.

Reavell compressors deliver air at a pressure of 1,000 lbs. per square inch. They have a volumetric efficiency of more than 90 per cent., and an equally high mechanical efficiency. The firm build similar apparatus to compress air up to 3,000 lbs. per square inch. Fig. VI.—8 shows the details of pistons and connecting rods.

Otto-Deutz Engine (Diesel Type).—The Gasmotoren Fabrik Deutz, of Cologne, are also builders of Diesel engines, and the following figures give the results of a test made by M. Barthe in October, 1907, upon one of their 35 H.P. engines :—

Revolutions per minute . .	209·3
Mean B.H.P.	35·0 (35·4 Metric B.H.P.)
Consumption per B.H.P. hour .	0·41 lbs.
Consumption of cooling water per B.H.P. hour	2·65 gallons
Mean temperature of inlet water .	66·0 ° F.
Mean temperature of outlet water	160·0 ° F.
Mean temperature of exhaust gas	536·0 ° F.

The fuel cost, delivered at works, including 3·5 shillings Customs Duty, was 11*s*. 10*d*. per ton.

The calorific value was 18,200 B.Th.U. per lb. and the solid residuals, 19 per cent.

Gardner Engine.—Messrs. L. Gardner & Sons of Patricroft, Manchester, build several different types of vertical engines, some petroleum engines with open frame, as represented in Fig. VI.—9, and others with closed crank case, as illustrated in Figs. VI.—10 and 11. The number of cylinders vary from one to four. The speed of rotation is from 600 to 800 revolutions per minute, and they are suitable for consuming either petroleum, spirit, or illuminating or producer gas. The weight of the engines amounts to about 55 to 66 lbs. per B.H.P.

Rathbun Engine.—The Rathbun engines, built by the S. M. Jones Co. of Toledo, Ohio, are built with two, three, or six cylinders and present several interesting features. The admission and exhaust

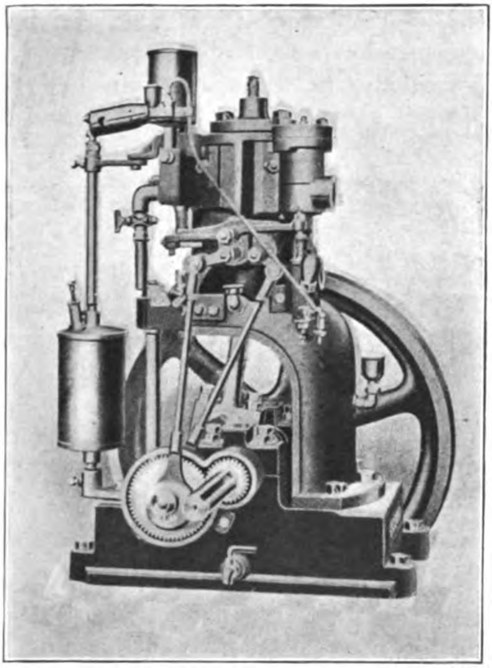


FIG. VI.—9. Open frame Gardner vertical Engine.

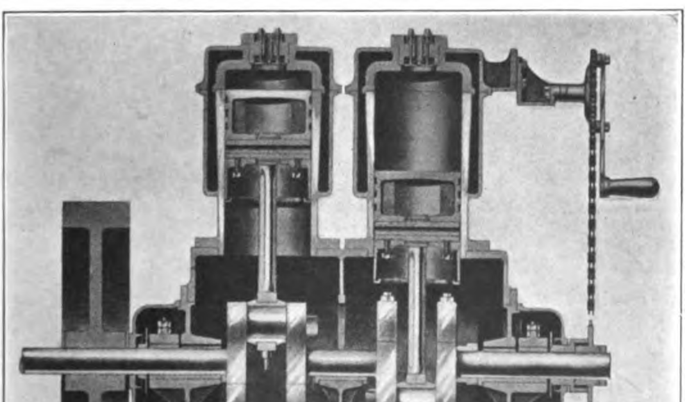


FIG. VI.—10. Section through Gardner (enclosed) vertical Engine.

valves are placed on the cylinder cover. The exhaust valves and their spindles are cooled by water circulation, as shown in Fig. VI.—12. Governing is by means of throttling, and adjustable ignition is provided. The half-speed shaft is placed at the base of the frame in a kind of box fitted with a cover which can be removed to obtain ready access to the cams, and to the ignition and governing mechanism. Ignition is of the make-and-break type with water-cooled plugs.

Westinghouse Engines.—Within recent years the former method of construction of this well-known engine has been abandoned in favour

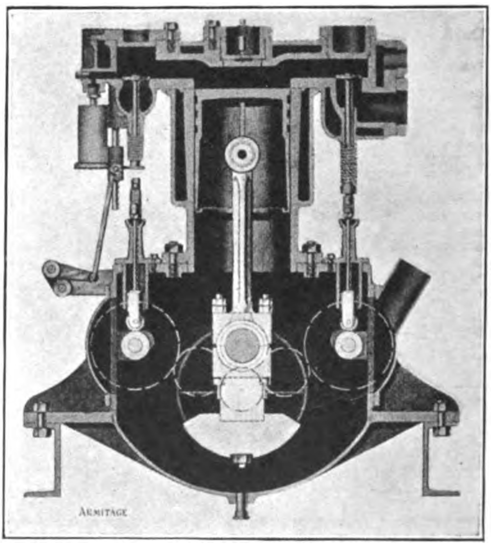


FIG. VI.—11. Transverse section, Gardner (enclosed) vertical Engine.

of a vertical tandem arrangement of single-acting cylinders so that the suction stroke of one cylinder is the explosion or working stroke of the other, the inertia of the moving parts being absorbed by compression in one of the tandem cylinders on the up-stroke and by a buffer cylinder on the down-stroke. At the Franco-British Exhibition in London, 1908, several engines of this type were installed for generating electric current and motive power, amongst them being one engine of 500 kw. capacity, direct coupled to a dynamo, and erected in the large Machinery Hall. Two other engines, of 130 kw. out-put, were erected in a building adjacent to the Canadian section; a third engine of smaller power, and a fourth of 100 kw. were placed in the adjoining building to the Australian section; each of these supplying the electric current for illuminating the various sections named.

The total ground area occupied by the 500 kw. engine (750 B.H.P.) as shown in Fig. VI.—13 was 24 feet by 12 feet 6 inches. The diameter of the lower pistons were 21 inches and of the upper pistons 28 inches, by 24 inch strokes; the normal speed was 200 revolutions per minute. The three pairs of cylinders are placed above the crank shaft, the buffer

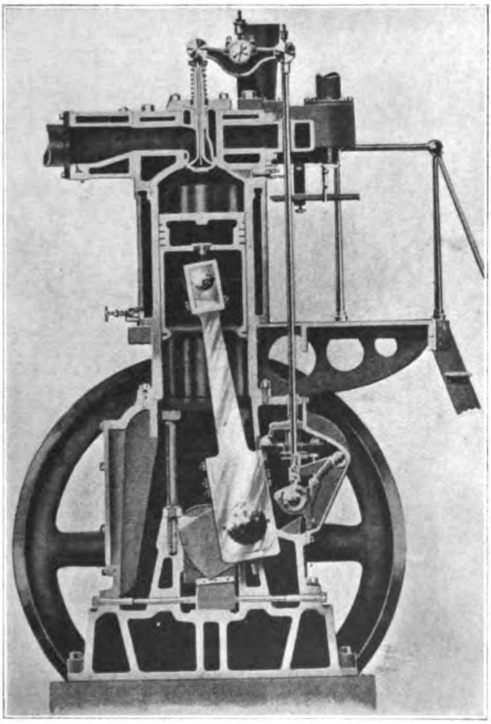


FIG. VI.—12. Rathbun vertical Engine.

cylinder being formed by the space between the open end of the upper and the cylinder cover of the lower. The frame is totally enclosed. Forced lubrication is adopted by means of duplicate valveless pumps. The oil is made to pass through wire gauze before passing to the pumps and the filters can be removed and cleaned whilst the engine is at work. None of the moving parts are water-cooled, as is often provided for in horizontal engines of the same power which have both their pistons and exhaust valves water-cooled. This is one of the advantages that result from the use of a number of small diameter cylinders, while a further important advantage is, that the engine works with great cyclic regularity without requiring fly-wheels of exceptional weight. Thus, the engines are peculiarly suitable for driving alternating current dynamos working in parallel.

The engine has been designed with great care in order to obtain simplicity and accessibility of all parts. By removing the upper covers and disconnecting the lower part of the connecting rod bearings, the connecting rods, pistons, and crossheads can be quickly

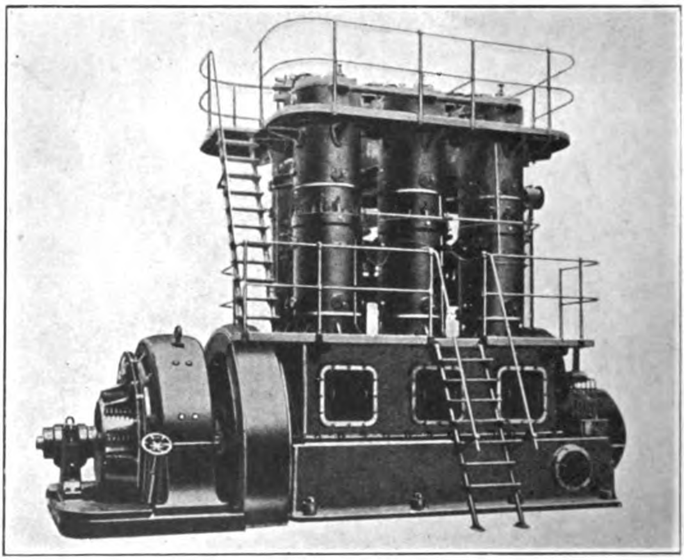


FIG. VI.—13. Six-cylinder, three-crank British Westinghouse vertical Engine.

removed. All the valves are operated by cams and the push rods work directly upon small bell crank levers pivoted on the cylinder covers. The engine is controlled by means of throttle governing.

Campbell Engines.—The Campbell Gas Engine Co., Ltd., of Halifax, England, have supplied over 54 vertical engines of from 60 to 360 B.H.P. up to May, 1909. The majority are in daily service in public and private electric generating stations and seven have been supplied as repeat orders, pointing to the fact that the engines originally supplied have been found to work satisfactory even under exacting conditions and in comparatively remote countries.

As special features the latest Campbell engines have their cylinders combined in groups of two, but each cylinder is distinct and independent with its own valve and ignition gear, so that in case of necessity the engine may run with one or more cylinders idle. The valve gear is arranged quite clear of the cylinder covers, enabling the latter to be removed or replaced with a minimum of trouble or time. The inlet and exhaust valves are operated from one cam shaft placed within the crank case and driven direct from the crank shaft by machine-cut gear wheels. The inlet valve is carried in a removable plug with a ground joint, the exhaust valve being water-cooled in the larger sizes.

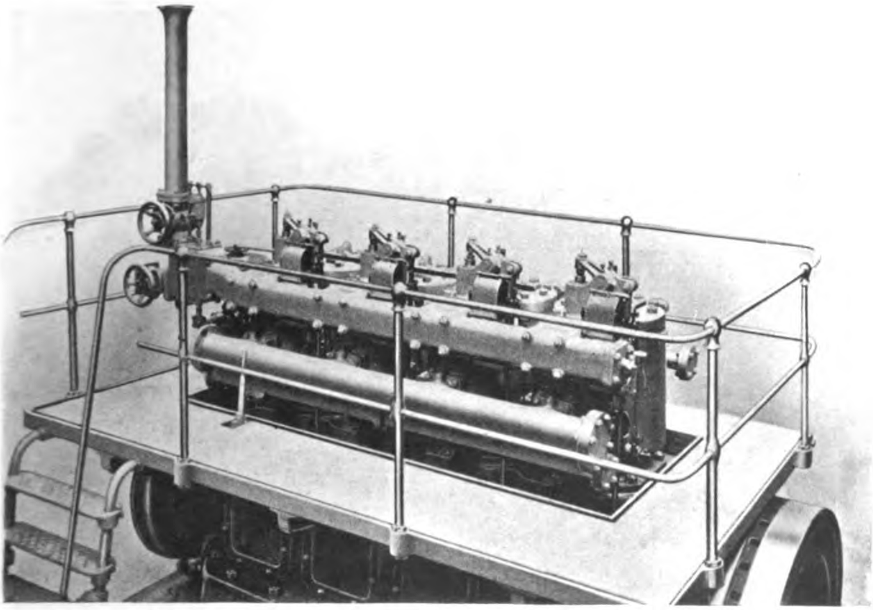


FIG. VI.—14. Cylinder-heads and valve gear, Campbell vertical Engine.

Governing is effected by variable admission of a constant mixture, the control valve being operated by a powerful governor. Low tension magnetos are fitted to each cylinder with adjustment for advancing or retarding the instant of ignition while the engine is in operation. Lubrication of all bearings in the crank chamber and of the crank-pin is effected by forced lubrication, the pressure being produced by one or more submerged valveless ram pumps driven by eccentrics on the crank shaft. The oil passes through a strainer to the pump, and a bye-pass, relief valve, and pressure gauge are provided. Water circulation is positively controlled by means of a centrifugal pump.

Fig. VI.—14 shows the ignition gear, inlet valves, and controlling valves of a 100 B.H.P. Campbell engine, with a platform all round the upper portion. Fig. VI.—15 is from a photograph of a four-cylinder engine rated at 125 B.H.P. on producer gas when running at 300 revolutions per minute.

Crossley Engines.—Fig. VI.—16 illustrates the four-cylinder vertical engine introduced by Crossley Brothers, Ltd., of Manchester, in the

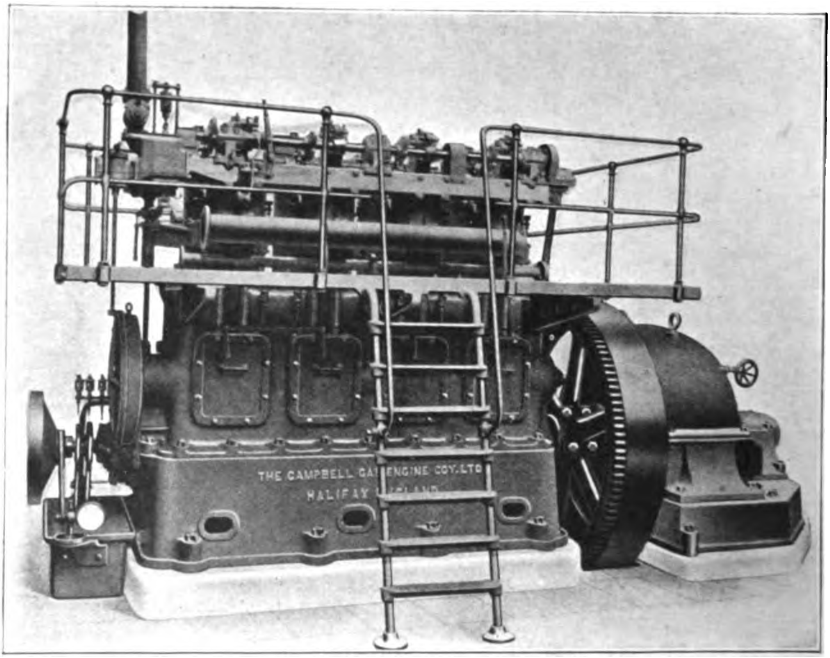


FIG. VI.—15. Four-cylinder, 125 B.H.P. Campbell vertical Engine, direct-coupled to dynamo.

early part of 1908. Forced lubrication is fitted and the oil is returned to the engine bed-plate in which are placed two valveless oil pumps which draw the oil through filters before its redistribution to the various bearings. The cam shaft is placed within the crank case, the inlet and exhaust valves, inverted in the cylinder cover, being worked by push rods and levers. From the section given in the right-hand bottom corner of Fig. VI.—17 it will be observed that the cylinder cover is provided with side apertures to register with the gas and air mains, the exhaust being led away above the covers (see Fig. VI.—16).

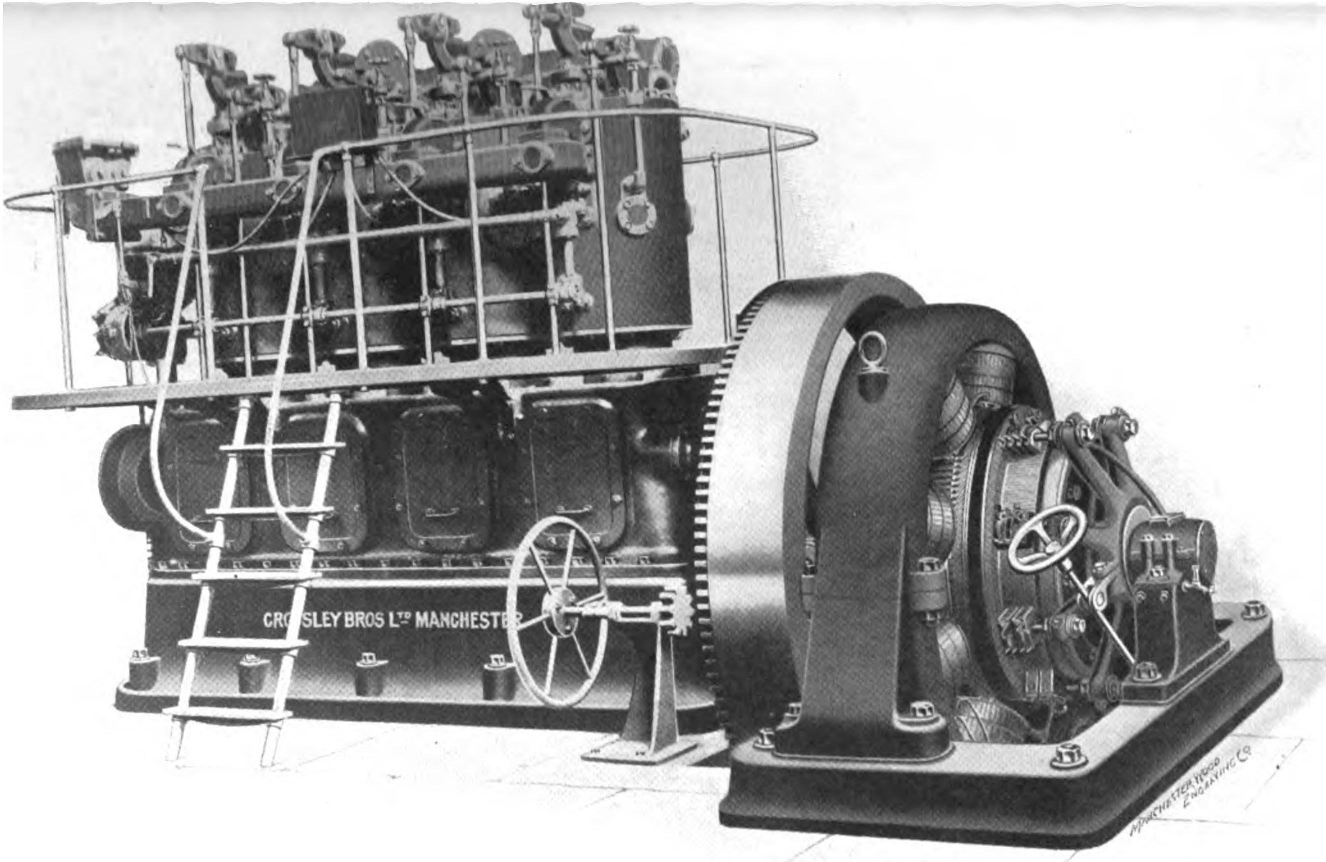


FIG. VI.—16. Four-cylinder, 180 B.H.P. Crossley vertical Engine, direct-coupled to dynamo.

The gas and air is conveyed to each cylinder in separate compartments of the supply pipe, as shown in Fig. VI.—17. In this way, should the charge admitted to one cylinder through derangement "back-fire," a minimum of disturbance occurs, even to that one cylinder. A gas adjusting valve is provided on all cylinders, so that the mixture can be suitably adjusted to obtain good results from each individually, or, if necessary, the gas charge to any cylinder can

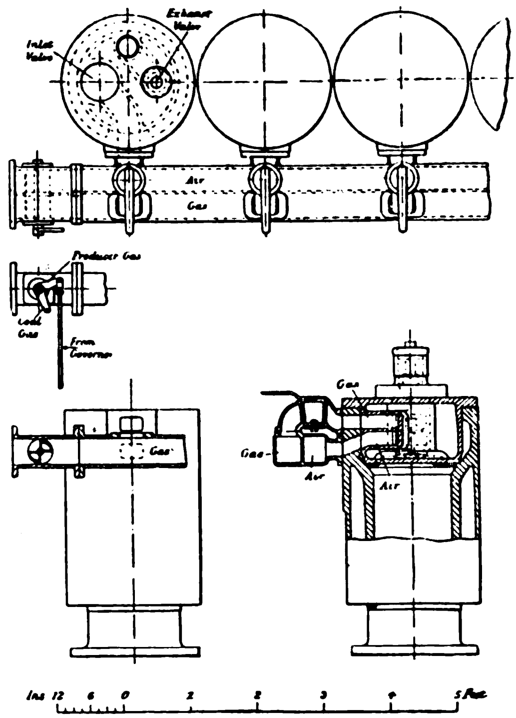


FIG. VI.—17. Mixture Regulation Device, Crossley vertical Engine.

be entirely shut off. The air can also be regulated, but such regulation affects all cylinders, being placed upon the air main. Governing is effected by altering more or less the position of two butterfly valves—one for gas and one for air—upon one spindle although in separate compartments. Thus the quantity of mixture admitted is varied according to the load.

A simple method has been devised by means of which the engine can be changed over from rich to poor gas or *vice versa* at a moment's notice and without stopping the engine. From the left-hand bottom diagram in Fig. VI.—17, it will be noticed that the throttle valve is of peculiar design, having four cutting edges. The air throttle has these four edges similar, but of the gas throttle two are similar to those of the air and are used when the engine is served with producer gas, while the other two are arranged with inclined surfaces of such a shape that they give suitable proportions of gas areas, to the air areas, when working with rich illuminating gas. It is only necessary to disconnect the governor rod from one arm of the bell crank lever and connect to the other arm to change from the one gas to the other.

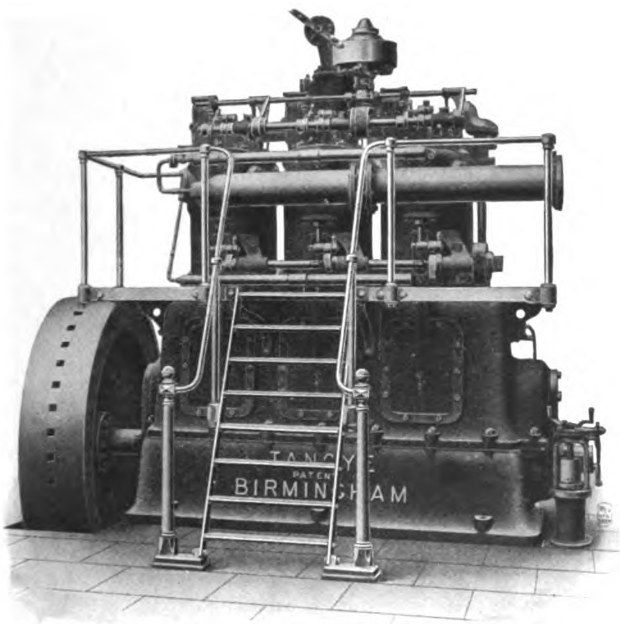


FIG. VI.—18. Three-cylinder, Tangye vertical Engine.

Tangye Engine.—Fig. VI.—18 illustrates the Tangye three-cylinder vertical engine, which is now being re-designed and improved in details. A number of these engines are in satisfactory application in sizes up to the maximum output of 180 or 200 B.H.P. The governing is by throttling the mixture. Splash lubrication is relied upon for the bearings and cylinder.

Attention has been directed to the inconveniences that apply to the vertical engine when compared to horizontal types. These inconveniences are, naturally, relative but are of sufficient importance in the majority of industrial installations to give the preference to the horizontal type. In a similar manner it is well known that the same arguments hold good in connection with the steam engine where the horizontal arrangement is generally preferred, vertical types only being used when the amount of space available is restricted.

Such considerations particularly apply in the application of the gas engine for marine work, a subject which is dealt with in the following chapter.

CHAPTER VII

MARINE GAS ENGINES

THE utilisation of gas engines for marine purposes has been the subject of very serious study by a number of makers who, in one way or another, have specialised in this direction, and who have successfully built internal combustion engines fed with heavy petroleum or even producer gas, which are working to-day with very good results. Mr. A. Vennell Coster, of Messrs. Crossley Brothers, Ltd., has published a series of interesting pamphlets upon this question, and has ardently advocated the attractive proposals to substitute gas engines for steam engines, both in warships and merchant vessels. However, in the actual state of progress of gas engines and gas producers, this kind of motive power is still subject to certain risks which appear to be incompatible with the reliability of work, which, at the present time, only the combination of steam engines and boilers appears to realise. Having regard to the exigencies of navigation it is none the less true that the practical solution of the problem must be accepted as the next for gas engines to overcome. The hypotheses considered by Mr. Coster, if somewhat bold, are not entirely premature, and the ideas that he has put forward in this respect deserve to be taken into consideration.

Mr. Coster has prepared a thermal balance-sheet of the three types of motive power, from the point of view of their application for marine service, and has drawn three diagrams, Figs. VII.—1, 2, and 3, which show the advantages of the gas engine from the point of view of heat utilisation in a striking manner. The figures given in these diagrams are deduced from best examples of current practice, and take into account, for both the boiler and producer, those fuels that are most suitable for each of them. In the diagram, Fig. VII.—1, relating to the marine gas engine, the efficiency of the producer is taken as 80 per cent., which is a conservative estimate in view of the fact that some producers have attained an efficiency of 90 per cent. of the total heat of the fuel. In the diagrams Figs. VII.—1 and 2, the marine boilers are internally fired and provided with tubes through which the flames return, and the efficiency of such has been taken as 70 per cent., which is somewhat over-rated because these boilers are well recognised as being uneconomical, more attention being given to the steaming qualities with regard to the production of large volumes of steam, and to restricted ground space than to economy. An efficiency of 80 per cent., sometimes obtained by stationary boilers fitted with economisers and other apparatus making for low fuel consumption, is not applicable in the case of marine boilers.

For the gas engine, the thermal efficiency per I.H.P. has been taken as being 33 per cent., and this is admissible in the case of well-constructed engines of the power under consideration.

As regards the steam turbine, Fig. VII.—2, both the thermal efficiency and the mechanical efficiency shown in the diagram have been rated very highly.

From trials made with two steamers identical with respect to form and water displacement, one driven by turbines and the other by reciprocating engines, at the same speed, with the auxiliary machines necessary, and working in the same limits of temperature, the thermal efficiency indicated for the turbine has amounted to 20 per cent. with a vacuum of 27 inches, the mechanical efficiency being 93 per cent. It should, however, be noted that a good mechanical efficiency cannot be maintained for as long a period in a turbine as with reciprocating engines, and, with regard to the latter, according to the diagram, Fig. VII.—3, the thermal efficiency is 17 per cent., and mechanical efficiency 92 per cent. These are the best results that have been specially obtained by the steamer *Iona* tested by a committee of the Institution of Mechanical Engineers.

By comparing the different diagrams it will be seen that the gas engine has a thermal efficiency double that of either the best reciprocating steam engine or the steam turbine.

The lowest consumption of coal per indicated H.P. in the case of turbines, has varied between 1·1 lb. and 1·5 lb. of coal. In power gas installations, this has amounted only to ·6 to ·8 lb., pointing to the fact that the coal bunkers, on this account, may be reduced by 50 per cent. from their present capacity, or, conversely, that the radius of action would be increased by 100 per cent. if the vessels were fitted with gas engines and producers.

Mr. Coster puts forward the following advantages in favour of the gas power system :—

1st. The consumption of fuel would be reduced by 50 per cent.

2nd. The stand-by loss would be reduced by more than 75 per cent.

3rd. Working pressure confined to the engine cylinders.

4th. No boiler or main steam pipes to burst or furnace crowns to collapse.

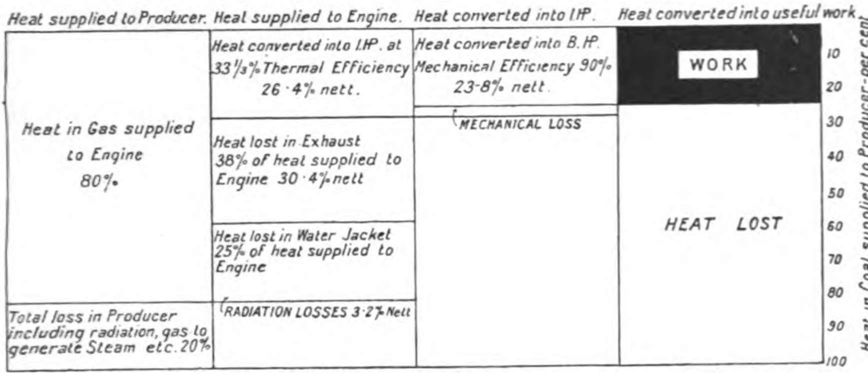


FIG. VII.—1. Thermal Balance-sheet of Marine Gas Engine.

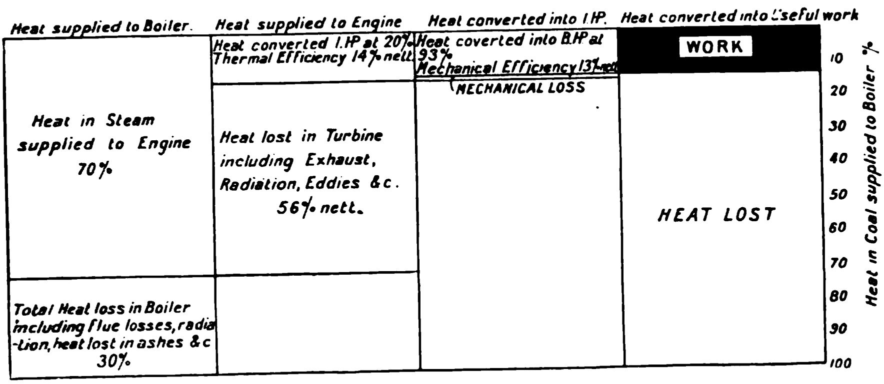


FIG. VII.—2. Thermal Balance-sheet of Steam Turbine and Marine Boilers.

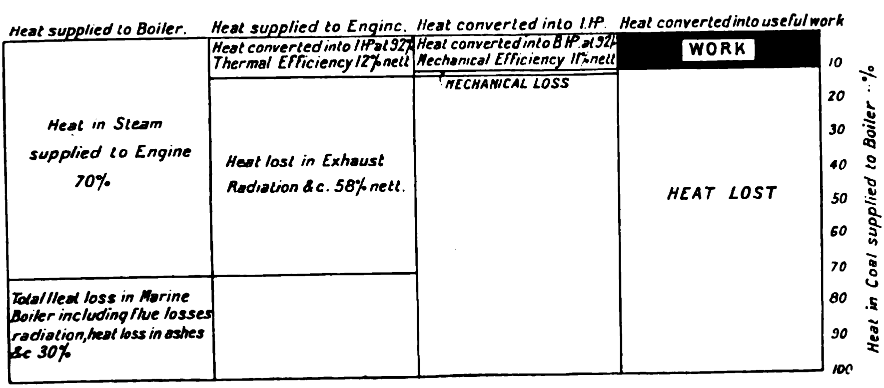


FIG. VII.—3. Thermal Balance-sheet of reciprocating Marine Steam Engine and Boilers.

5th. No priming in a heavy sea-way; no water-hammer in pipes and cylinders.

6th. No more difficulties with the firing of boilers in a beam sea. Gas producers may be charged only twice every twenty-four hours, and the rolling and pitching of the vessel is rather advantageous than otherwise in assisting the settlement of fuel down from the charging hoppers.

Mr. Coster, however, does not conceal the great difficulties presented by the problem in the present state of gas engine progress. He points out that these main difficulties are:—

1st. The construction of a gas producer able to gasify all grades of bituminous coal.

2nd. To find a simple method to cleanse the gas from tar and dust, either before the introduction of the fuel into the producer itself, or after the gas has left the generator on its way to the engine.

3rd. To ensure perfect control of the gas-propelled vessel, starting, stopping, reversing, and running at all speeds.

These difficulties are certainly not incapable of solution. The problem presents itself in different ways according to the particular service that the ship is called upon to perform.

In pleasure boats, motor-boats, and commercial vessels generally, for rapid transport with light loads, either vertical four-cycle motors are installed to work with spirit or refined petroleum, or two-cycle vertical petrol engines.

The use of light spirit involves risk of conflagration when the power of the engine is sufficiently large to require a large reserve of fuel. American engineers have developed the construction of two-cycle petrol motors for such small vessels, to a great extent, and supply them at an extremely low price.

For freight boats for river service, as well as for fishing and coasting vessels, a considerable number of refined oil engines are in use mainly of the vertical type, but sometimes horizontal.

The Griffin Engine.—The Griffin Engineering Co., Ltd., of Bath, have built an engine specially designed for marine work which will burn either refined petroleum or any kind of liquid hydrocarbon. The vaporiser is heated by the exhaust gases, and consists of a cylinder with rough walls, into which water and the liquid fuel are injected with a small quantity of air. The supplementary air necessary is admitted through an opening at the other end of the vaporiser. The temperature of the vaporiser never exceeds 400° F. In this way the decomposition and gasification of the fuel is avoided, and thus no tar is formed within the cylinder. Consequently, all the parts of the engine are maintained in good working condition even after long periods of continuous work. The residuals of the oil are automatically ejected from the vaporiser. Ignition is effected by means of a tube heated by a blow lamp, the tube being maintained at a cherry-red heat. It can be readily examined whilst the engine is at work, and the regulation can be instantly adjusted.

The engines are built up to 200 H.P., with two or four single-acting cylinders. The pistons of each pair are joined at the lower part by a rigid cross-head which is connected by a connecting rod to the motor shaft, the latter being fitted with a reversing clutch.

From a number of experiments with Griffin engines fed with various fuels, the average compression pressure is shown to be 60 lbs. per square inch. The initial explosion pressure is 240 lbs. per square inch, and the mean pressure 70 lbs. per square inch.

The Standard Engine.—The Standard Motor Construction Co., of Jersey City, U.S.A., have built several petroleum engines of 500 H.P. for ship-work. Several ferry boats are driven by 300 H.P. Standard engines, and recently a 500 H.P. double-acting engine has been installed which is set to work by compressed air and has an arrangement for reversing. For some time past gas engines and producers have also been employed.

The Otto-Deutz Engine.—The Gasmotoren Fabrik Deutz have built several vertical, reversible, producer gas engines. The first engine built upon this principle was of the following dimensions: Diameter of cylinder, 280 mm. (11 inches nearly); stroke, 300 mm. (12 inches nearly); revolutions per minute, 300; B.H.P., 150. It consists of four cylinders; two of these, when starting up with compressed air, work on the two-cycle principle, but in normal work all four cylinders are of the single-acting, four-cycle type. The mechanical details of each of the four cylinders are completely independent from one another. The frame is "A" shaped, as shown in Fig. VII.—4, and is carried upwards to form the water-jacket, within which the cylinder liner is introduced. The breech end holds the liner in position, and in each breech end the admission and exhaust valves and ignition apparatus are fitted. The admission valve on each cylinder, either of which can be worked by means of compressed air, is guided by a piston. Each group of two cylinders is erected on an iron base casting, the crank shaft being supported in bearings which form part of the bed plate. Lubrication of these main bearings is effected by means of a small force pump. The fly-wheel, 55 inches in diameter, is placed between the two base castings. The crank is unbalanced, the moving parts themselves counteracting each other to give an equal turning moment. The half-speed shaft which operates the valve gear of the four cylinders is placed at mid-height together with the reversing gear. It is supported on ball bearings, and is operated by an intermediate vertical shaft from which the governor is driven.

The movement is transmitted by means of two pairs of skew gear wheels, giving one-half the number of revolutions of that of the engine shaft. The half-speed shaft carries the cams for working the valves. A box, the lower part of which is formed by the main frame, contains the cams, levers, and reversing gear, and is shown in Figs. VII.—5, 6, and 7. In the upper part of these boxes are placed the valve rod guides. Each valve can be worked by one cam for forward or another cam for backward direction of rotation, these cams being placed side by side upon the shaft. To each of the two cylinders, which are arranged to work with compressed air for starting, two additional cams are fitted, corresponding to forward and backward direction, when the cylinders work as two-cycle engines.

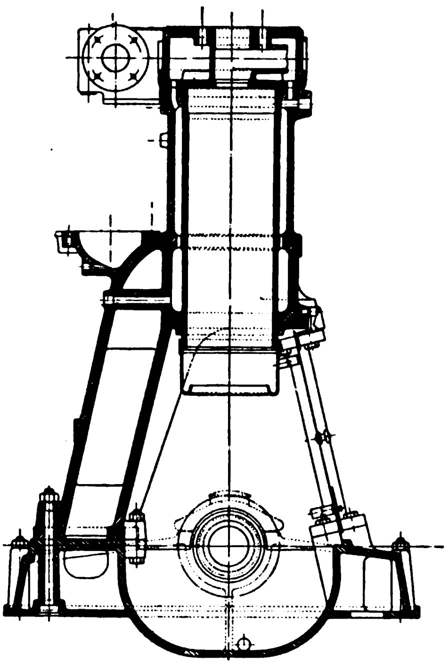


FIG. VII.—4. Section of Frame and Cylinder of 150 B.H.P. four-cylinder Gasmotoren Fabrik Deutz Marine Gas Engine.

Finally, between the admission and exhaust cams, two others are fitted, one for forward and one for backward movement, to control the ignition. To make the motor reversible the movement is transmitted indirectly to an intermediate piece, which determines the direction of rotation of the engine when it is displaced between the cam and valve rod guide. Every intermediate piece is operated by means of only two reversing levers. Governing is effected by the variable admission of mixture. For working with power gas, the air and gas pipes are fitted with a hand-throttle. For working with liquid fuel, the latter is conducted directly to a carburettor at constant pressure placed in the breech end. The quantity of air necessary is regulated by varying the stroke of the admission valve, as determined by the governor. Ignition is by electric magnetos.

For reversing, it is necessary to have a compressed air reservoir.

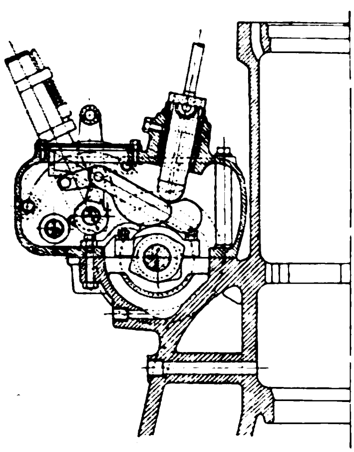


FIG. VII.—5.

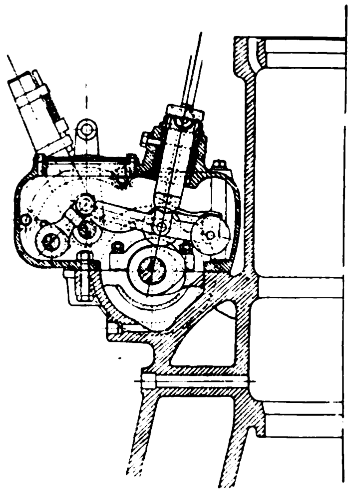


FIG. VII.—6.

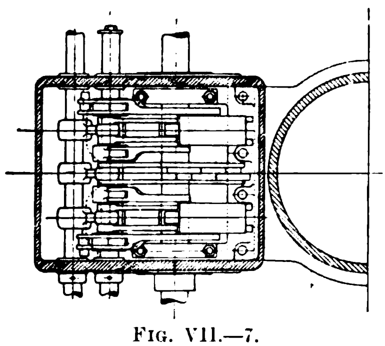


FIG. VII.—7.

Reversing Mechanism of Gasmotoren Fabrik Deutz Marine Gas Engine.

The latter is maintained under a constant pressure by means of a small compressing engine. A pressure of 90 lbs. is sufficient to permit the engine to be reversed. Reversing can be quickly obtained by introducing compressed air to produce back pressure.

For ships of heavy tonnage requiring units of large size, the constructors who advocate the use of producer gas are now engaged in making a series of trials. Some of the conditions that must be fulfilled, and to which attention has been directed in connection with Mr. Vennell Coster's statements on the subject, have not yet received an entirely satisfactory solution.

If the question of the use of internal combustion engines for merchants' ships has not yet made all the progress that one could wish for, it is to be attributed more to economical considerations than to technical difficulties. The solution of the question with regard to

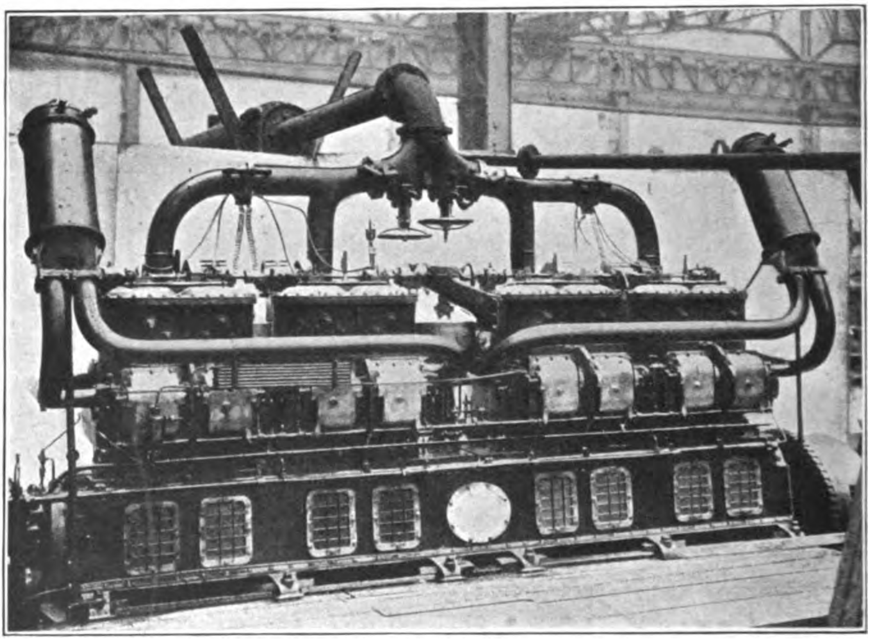


FIG. VII.—8. Koerting two-cycle, multi-cylinder Engine, for Submarine Vessels.

warships may be considered as having already been acquired, so far as concerns submarine vessels, although in this particular application the problem presented some special difficulties.

Koerting Engine.—The firm of Koerting Brothers have constructed the two-cycle engine shown in Fig. VII.—8, for service in submarine vessels. Its characteristic feature is the entire absence of valves. In this engine the admission and expulsion of the charge in the main cylinder as well as in the mixture pump, and control of the scavenging air, is effected by the principal piston working as a slide valve.

The cam shaft serves only the ignition and governing mechanism. Thus the motor is particularly reliable in working. The consumption of fuel is higher in these than in the Diesel engines, but their reliability is greater. The makers have received orders from several Governments.

The Thornycroft Engine.—Messrs. J. I. Thornycroft, of Chiswick, London, also build engines for twin-screw submarine boats. They have furnished, amongst others, several to the Royal Italian Navy.

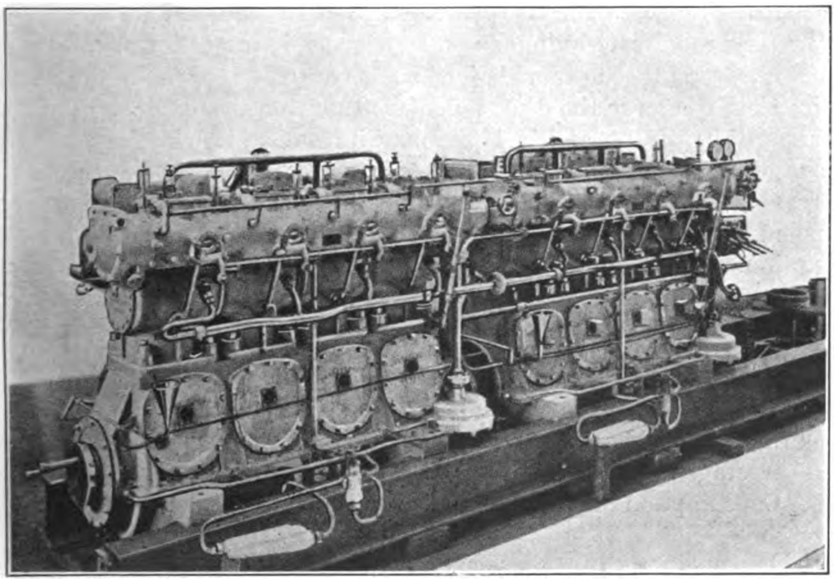


FIG. VII.—9. Thornycroft four-cycle, multi-cylinder Engine, for Submarine Vessels.

The engine consists of eight cylinders, in two groups of four, attached to the same crank shaft (Fig. VII.—9). Each cylinder is 12 inches diameter and 8 inches length of stroke. The crank case is completely closed. Lubrication of the bearings and those at the end of the crank case are provided with water circulation, and, in addition, the pistons and crank case are continually cooled by a blast of cold air from a fan.

To facilitate the removal and replacement of the pistons and connecting rods, the base is provided with doors. The cylinders can also be removed. The cooling water is circulated by a centrifugal pump, driven either by an electric motor or by a small oil engine using the same fuel as the main engine. The advantage of this method is that the engine can be rapidly cooled after stoppage to enable inspections and necessary adjustments to be made, and also that the amount of cooling water can be controlled to a nicety, according to the need of the engine or the peculiar nature of the fuel used.

The vaporiser is connected to the first portion of the exhaust pipe in such a manner that the temperature can be regulated to suit the brand of petroleum employed, the quality of the latter being very variable necessitating different treatment being given with respect to the quantity of heat required for its vaporisation.

With proper regulation of heat neither the engine nor the vaporiser require cleaning, and tarry deposits are not formed.

For reversing, the cam shaft is made to turn always in the same direction, whatever the direction of rotation of the engine. This is arranged by means of bevel pinions with solid jaw clutches. Compressed air is used to start the engine in either direction, the compressed air admission valves remaining closed whenever the ignition explodes the working charge. Low tension magneto is employed for ignition.

The consumption of petroleum is about 0·7 pint per B.H.P. per hour. The total weight per group of four cylinders is 7,700 lbs. The result of a three hours' test made by Messrs. Thornycroft on one of their eight-cylinder engines gave the following figures:—

Average power . . .	325·5 B.H.P.
Average speed . . .	582·5 revolutions per minute.
Consumption of petroleum .	·66 pint per B.H.P. hour.

These engines can be fed either with petroleum, as in the above trial, or by spirit. In the latter case the vaporiser is not required.

American engineers have principally employed gasoline (petrol) engines for this work. The United States naval authorities have six submarines, each driven by a single-acting, four-cycle 160 B.H.P. engine. Other submarines are under construction, and will be fitted with 250 B.H.P. Craig engines.

CHAPTER VIII

TWO-CYCLE ENGINES

Two distinct cycles are to-day favoured by manufacturers. The four-stroke cycle of Beau de Rochas (1862) and the Clerk two-stroke cycle. The latter was particularly studied and employed when the monopoly claimed by the *Gasmotoren Fabrik Deutz*, under the name of the Otto cycle, prevented makers from freely applying the four-stroke cycle to their engines. It was at this epoch that Clerk (1878), Ravel (1878), Wittig (1880), Koerting (1880), Benz (1884), produced their two-cycle engines.

From 1886, in which year the extent of the Otto patents was considerably curtailed by the decision of the German Law Courts, the large majority of makers began to use the four-stroke cycle, and since then only a few attempts have been made to design two-cycle engines. Amongst these, well in the front rank are the Koerting and Oechelhäuser engines, which have given some remarkable results when served with blast furnace gas. In America also the two-cycle engine, working with gasoline, is largely adopted for driving launches. In Chapter VII. mention has already been made of the two-cycle engines for submarine vessels built by *Koerting Brothers*.

Partisans of the double-acting four-cycle engine put forward the following as advantages of this type over two-cycle engines for large powers :—

1. A minimum of weight of engine, which so far has not been attained. This weight will be 200 lbs. per B.H.P. for single-cylinder engines, and in somewhat less proportion for the tandem engine. (Nevertheless, these low weights are not realised by the examples mentioned on p. 78.)

2. A mechanical efficiency of 90 per cent. instead of 65 to 70 per cent. for two-cycle engines. With regard to this point, it may be mentioned in passing that the *Siegener M.A.G.* guarantee an efficiency of 74 to 76 per cent. under normal load for their two-cycle engines, and 78 to 80 per cent. on overloads ; this efficiency being measured by the ratio between the indicated work in the air cylinder of a blowing engine and the indicated work in the engine cylinder. The work of the pumps on normal load never exceeds 6 per cent., and on overloads 7 per cent. of the total I.H.P. of the motor cylinder.

3. Elimination of cylinder ports which shorten the life of the cylinder itself and the piston rings.

M. Güldner, who has himself designed two-cycle engines, but now builds the vertical four-cycle type, is of the opinion that the future belongs to the two-cycle engine, and that only.

Professor Witz believes that four-cycle engines, double-acting, are bound to prevail, seeing that the mechanical efficiency of two-cycle engines is not usually more than 72 per cent.

Herr Reichenbach is of the same opinion, and states that the use of the four-stroke cycle, tandem, double-acting engines allows the combination of high thermal efficiency with an equally high mechanical efficiency. On one hand, there is no fear of wasted gas; the best mixture of air and gas is assured; no danger of explosion during the suction stroke need be feared; and finally, the compression can be raised sufficiently to assure the maximum utilisation of heat being obtained without risk. Besides these points, the internal resistances are reduced to a minimum, seeing that each compression is produced by the engine piston without any intervening rotary or reciprocating mechanism; therefore, the preparation of the gaseous mixture entails no frictional loss.

Professor Meyer, after having found that the negative work is 4 to 5 per cent. of the indicated power in four-cycle engines and from 10·5 to 11·5 per cent. in two-cycle engines, admits having thought, at one time, that the two-cycle engine will supplant its competitor, but he now points out the advantages of the four-cycle engine, particularly its simplicity of operation, which make it exceptionally fit to compete with the two-cycle type.

Herr Junge has compared the two types of engines particularly in relation to fuel consumption, initial capital outlay, space occupied, and ease of management. He states that the economy of fuel is a very important subject to enquire into when the relative merits of gas and steam engines are under discussion, but it does not enter into consideration when a decision is to be made in connection with two-cycle or four-cycle engines only, because, in the first place, no sufficiently precise tests have been made to enable just conclusions to be formulated as to the superiority of one over the other, and in the second place, because it is of very little consequence to know if one engine does consume a few hundreds of cubic feet more than another. Such gain would have no real value except when the gas thus saved could be stored and used. This cannot be arranged in the majority of large gas-power installations.

The initial capital cost is lower for a two-cycle engine of the

Koerting type than for a four-cycle engine, and the simplification of the pumping arrangements made from day to day increase this difference.

The space occupied by the Koerting is also less, and when this is an important consideration it should be given the preference.

The absence of exhaust valves in the Koerting engine is an undeniable advantage, but it is usually appreciated more by the attendant than by the proprietor.

Professor Diederichs has studied the question in great detail and has made a complete comparison between the four-cycle and two-cycle types. His conclusions are briefly summarised as follows :—

1. **Thermodynamic Actions in the Cylinder.**—At the beginning of compression in the two-cycle engines the temperature is lower and the pressure of the working charge greater. The denser mixture produces a greater specific power. In the two-cycle the loss of heat to the cooling water is greater during compression; this permits higher compression pressures and produces higher thermal efficiencies.

2. **Fluid Friction.**—The losses resulting from fluid friction are greater in a two-cycle than in a four-cycle engine. They are lesser or greater according to the system of pumps adopted ; the latter may be classed in the following order of merit :—

A. Separate pump with large reservoir.
B. Separate pump without reservoir.
C. One end of the working cylinder used as a pumping cylinder.
D. Use of the crank case as a pump.

3. **Mechanical Friction.**—The mechanical efficiency properly so called (making a distinction between mechanical and fluid losses) appears to be a little higher in two-cycle engines.

4. **Economic Considerations.**—The net cost of installation, the running expenses, ease of maintenance and reliability are the principal considerations. According to circumstances and the duty to be fulfilled, the advantages of two-cycle engines may outweigh those of the four-cycle, and *vice versâ*.

It is of interest to make a comparison between the two cycles with respect to the number of essential parts that are involved in construction, working and maintenance, and particularly those which are subjected to heavy strains and exposed to wear.

For instance, in a 2,000 H.P. engine, the number of such details for the various types of present day construction are as follows:—

	Two-cycle engine.		Four-cycle engine.
	Oechelhäuser.	Koerting.	
Number of cylinders	2	2	4
,, ,, pistons (engines and pumps)	6	6	4
,, ,, piston-rods and stuffing-boxes	4	10	8
,, ,, admission valves . . .	2	4	8
,, ,, exhaust-valves . . .	—	—	8

For the valve gear the four-cycle double-acting engine with two cylinders will require two side shafts, operated by four or eight gear wheels. These shafts usually carry sixteen eccentrics or cams for inlet and exhaust, and, in some cases, eight additional eccentrics or cams for the gas. In the Oechelhäuser engine, this complication is balanced by the four return connecting rods with their coupling rods and crossheads, and, in the Koerting engine by the set of distributing valves, slides, &c., for the supply of air and gas.

M. Reinhardt, a constructor of four-cycle engines, recognises the qualities of the two-cycle engine and even its superiority for operating blowing engines. His conclusion is that it is impossible to definitely decide the point at issue at the present time.

The author shares this opinion and thinks that it would be premature to positively accept one system to the exclusion of the other. He believes, moreover, that any such preponderance will never be manifest. The two cycles each have their special spheres, and the improvements that builders are constantly making in one or the other present an equal interest.

The characteristics of the two systems may be recognised by reference to the annotations to Figs. VIII.—1 and 2.

In the Oechelhäuser and Koerting two-cycle engines, the pumps attached to them introduce the explosion charge into the cylinder, delivering the air and gas separately under a pressure varying from 5·5 to 14 lbs. per square inch. In this way, not only is there no depression at the time of admission, but, by means of the separate control of the air and gas, a blast of air can be admitted previous to the introduction of the fresh charge to scavenge the products of combustion.

It is during the period between the end of the expansion and commencement of compression that exhausting, scavenging, and introduction of the new charge should take place. In spite of the

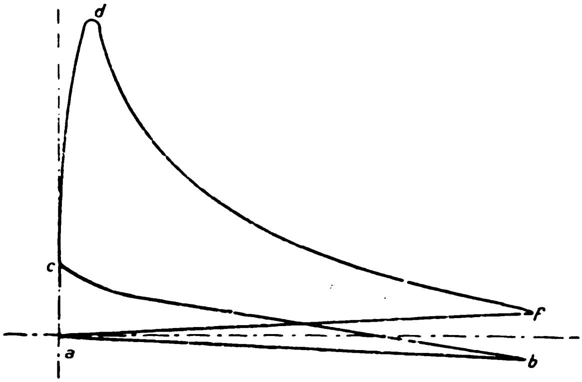


FIG. VIII.—1. Indicator diagram, illustrating four-stroke cycle.

1.	a. to b.	Induction of mixture.
2.	b. to c.	Compression.
3.	c. d. and f.	Ignition and expansion forming the only working stroke of the cylinder.
4.	f. to a.	Exhaust.

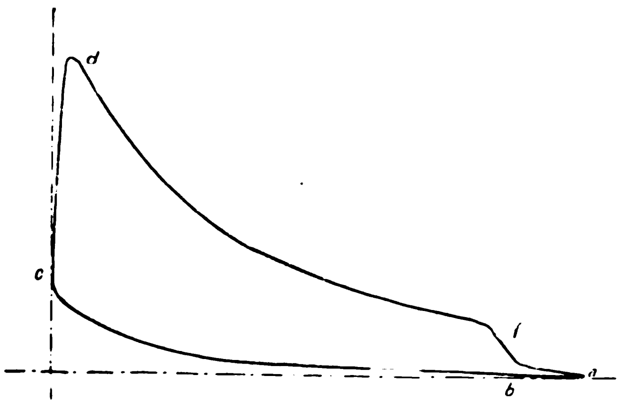


FIG. VIII.—2. Indicator diagram, illustrating two-stroke cycle.

1.	a. to b.	Induction ; b. to c. Compression.
2.	c. d. and f.	Ignition and expansion ; f. to a. Exhaust.

large admission ports provided, the short time allowed for these three operations necessitates the use of pressure to overcome the resistances to the admission of mixture. The pumps which accomplish this operation should be of nearly equal dimensions to that of the engine cylinder. This involves an appreciable amount of work and a reduction of the total mechanical efficiency to 75 or 80 per cent. under the most favourable conditions, and, further, tends to complicate the two-cycle engine.

The Oechelhäuser and Koerting systems have, however, some peculiar advantages, foremost amongst which may be mentioned the fact that the speed of these engines can be varied within a wide range, thus approaching the working conditions of steam engines.

The construction of two-cycle engines is more simple and easier than that of four-cycle tandem engines of the same power, for the machine is shorter, and only comprises one motor cylinder. The complicated and difficult operation of opening and closing large exhaust valves is also avoided, but on the other hand it becomes necessary to carefully study the operation and adjustment of the admission valves and pumps. In one word the construction of a two-cycle engine, and especially of the Koerting type, demands the utmost care, but when the makers' shops at Koertingsdorf are visited and inspection is made of the machines that are sent out, any anxiety that may be felt in this matter is entirely allayed. The workmanship, finish, and the scrupulous care bestowed upon the selection of material are traditional of the firm of Koerting Brothers.

This firm and their licensees have triumphantly responded to the arguments of their competitors by the number of installations already carried out or in progress. In 1907, 214 engines, representing about 182,000 H.P. were supplied, while the Oechelhäuser two-cycle engine was represented by 60 engines amounting altogether to about 50,000 H.P.

Two-cycle engines still find serious detractors, and up to the present time the preference of the majority of builders has been given to the four-cycle type because of the simplicity and high efficiency of these engines.

The following descriptions are given of the most interesting two-cycle engines:—

Koerting Engine.—This double-acting engine is represented in elevation and in horizontal and vertical section in Figs. VIII.—3 and 4. Its method of working is as follows:—

1. The ignition of the mixture and development of pressure takes place, after the introduction and compression of the charge, close to the back dead centre of the piston.

2. The expansion of the burnt mixture and transmission of power to the crank shaft takes place during the forward stroke of the piston.

3. When the piston has arrived at the front dead centre, the expul-

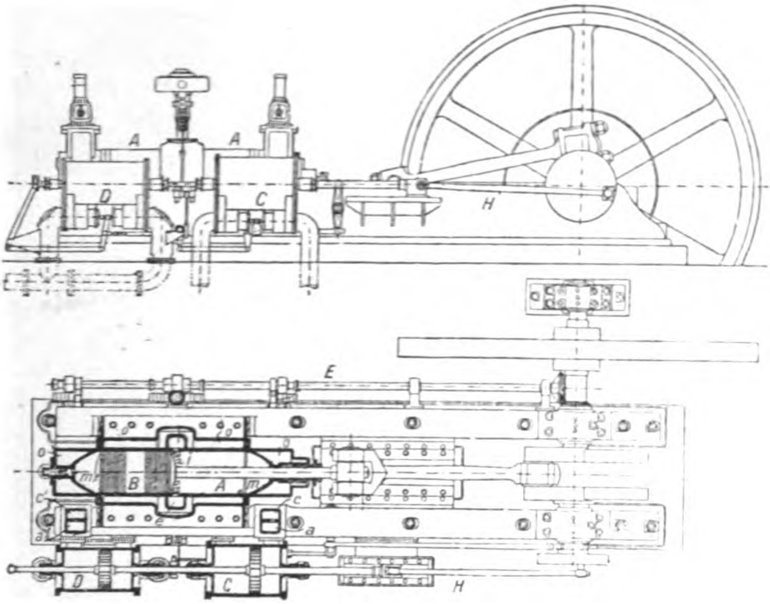


FIG. VIII.—3. Diagrammatic elevation and part sectional plan. Koerting two-cycle Engine.

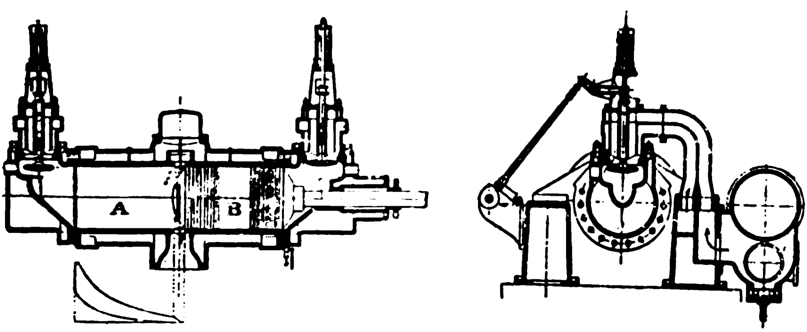


FIG. VIII.—4. Diagrammatic sections of Koerting two-cycle Engine.

sion of the products of combustion and then the admission of the new charge takes place.

4. During the return stroke the piston compresses the explosive mixture.

This work, which in four-cycle engines occupies four strokes of the piston, is effected in two strokes in the Koerting engines, the expulsion of the residual gases and the inlet of the new charge occurring during the very short time that the piston remains in the zone of the front dead centre.

The engine piston *B*, Fig. VIII.—4, does not draw in the mixture. The cylinder *A* is refilled by the pumps *D* and *C*, Fig. VIII.—3, the new charge itself expelling the burnt products and occupying the space thus made vacant. The introduction of fresh mixture is made in the following manner:—

1. It becomes diffused uniformly throughout the area of the cylinder and thus drives out the former mixture.
2. A cushion of air becomes interposed between the new charge and the hot residuals of the previous explosion, preventing contact of the two mixtures and premature ignition of the fresh charge.

The outlet ports in Fig. VIII.—4 for the products of combustion are situated in the middle of the cylinder in the form of annular orifices. The piston (of a length equal to the stroke, less the width of the exhaust ports—*i.e.*, one-tenth of the stroke) alternately opens and closes these openings in its backward and forward movement.

When the exhaust commences, the internal pressure is still from 30 to 45 lbs. per square inch, and consequently the burnt gas escapes at a velocity of from 1,000 to 1,300 feet per second, causing the pressure to fall to about 1·5 lbs. per square inch when the crank-pin is only 20° from the outer dead centre. The expulsion of the products of combustion and the introduction of the new charges, pushing out the last traces of the previous explosion, occurs in the period that elapses between the disclosing and the obstruction of the ports by the piston.

The admission valves—plates with spring shutters—are operated by the cams placed on the two sides of the cylinder. The gas and air are brought by the separate pumps *D* and *C* (Fig. VIII.—3) but only the mixture enters the cylinder. The pumps are driven from the main shaft, and the angle of lead of their crank is about 110° in advance of the engine crank. They at first force pure air, and afterwards a mixture of uniform composition of which the quantity only is varied to regulate the power developed by the engine. The pumps introduce the mixture at a pressure of about 3 to 4 lbs. per square inch. The pumping pistons are at the end of their suction stroke as soon as the engine piston has closed the exhaust ports, and then compression begins. The two pumping pistons are placed on the same rod and have the same movement; the composition of the mixture depends entirely upon the ratio between the areas of the two pumping pistons.

The supply pipes *Aa* (Fig. VIII.—3) for the air and *Cc* for the gas terminate at the admission valve. When the admission valve opens the fluid nearest to the valves will enter first into the engine cylinder; therefore, if by some means, air is made to pass into the gas supply pipe and thus push back the gas, air will then be present on both sides of the valve, as shown in Fig. VIII.—5, and upon the opening of the valve air only will pass through in advance of the gas until the latter reaches the valve.

The critics of two-cycle engines, however, say that a certain amount of gas is always mixed with the scavenging air because the two fluids do not remain distinct but become diffused. Analysis of the products of combustion made in different cases have in fact revealed the presence of combustible gas in the exhaust pipes. But this waste of gas is very

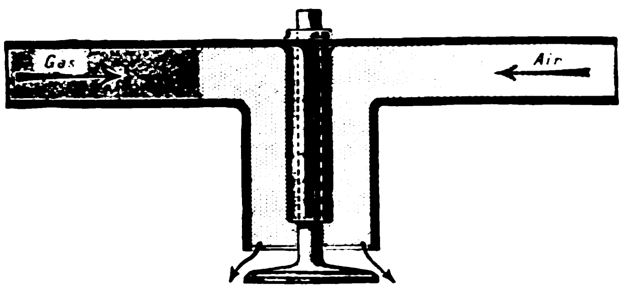


FIG. VIII.—5. Diagram showing Method of Supplying Air only at first opening of Inlet Valve (Koerting Engine).

small, because a portion of the scavenging air remains in the cylinder under normal loads.

The manner in which the pumps introduce and control the charge can be readily understood from Fig. VIII.—6. As already mentioned, being double-acting, they draw in and drive out at the same time, and are arranged to first introduce pure air and afterwards an intimate mixture of constant composition. The gas pump, therefore, has no work to do during a certain period, while the air pump is operating. Afterwards, the gas pump delivers in such manner that the composition desired reaches the engine instead of pure air.

The air pump works with full admission like a steam engine. In the gas pump the opening of the suction occurs only after the piston has completed 40 to 50 per cent. of its travel. During this time the delivery remains closed and the gas drawn in by the preceding stroke re-enters the suction pipe. The pump delivers during the second period of the piston's stroke, and the two pumps acting together from this moment, supply gas and air at equal velocity so that the composition of the mixture is uniformly maintained.

The engine inlet valve is not opened at the beginning of the delivery stroke of the air pump piston but only after the pump piston has nearly reached mid-stroke. During this interval the air accumulates in the supply pipe, and by its pressure forces back the gas, the gas pump not yet delivering. Upon the opening of the engine inlet valve, therefore, pure air alone enters until the gas pump commences its delivery. This produces the scavenging of the cylinder and the

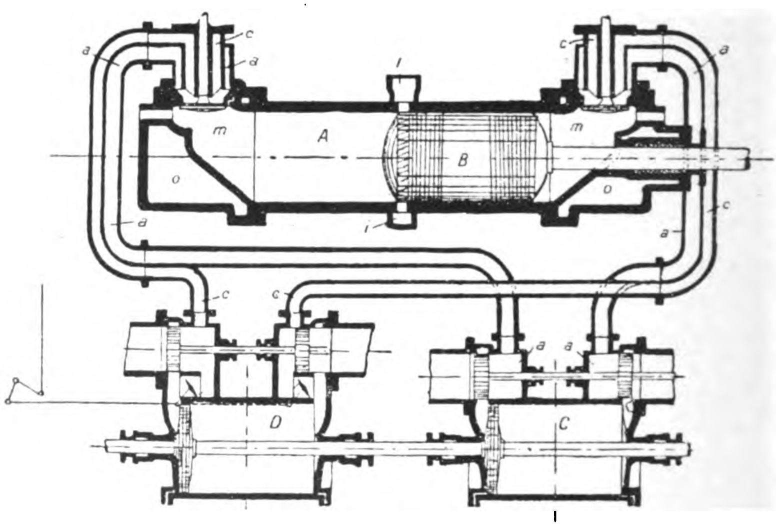


FIG. VIII.—6. Diagram showing Method of Governing (Koerting Engine).

formation of a cushion of air against the piston previous to the introduction of the uniformly constituted explosive charge.

The regulation of the engine is obtained in two ways:—

(1) By retarding the delivery from the gas pump, by actuating the side valve of the pump rod by a slotted link operated by two different eccentrics, and controlled by the action of the governor raising or lowering the link. This method necessitates the use of a very powerful governor.

(2) By a bye-pass between the delivery and suction branches of the gas pump by means of a butterfly throttle valve operated by the governor. When the throttle is partly open during the suction stroke and the first portion of the delivery stroke of the piston, a greater or lesser amount of gas is bye-passed from the delivery pipe to the suction

chamber. The column of gas flows back in the passage behind the engine inlet valve and is replaced by air. When the engine inlet valve opens so much more air and so much less gas enters the cylinder, according to the position of the throttle. This adjustment is effected by a slotted link as in the first case.

The butterfly throttle valves in some cases are now replaced by piston valves mounted concentrically to the admission and exhaust slide valves of the pump.

A reciprocating movement is imparted to the piston valves by the operating eccentrics and they can also rotate upon their rod as regulated by the governor. The two movements occur simultaneously and thus, according to the load upon the engine, permit the gas supply pipe to deliver more or less gas to the mixture fed to the engine cylinder.

Ignition of the mixture is obtained by means of two magnetos, with the spark advanced 18 or 20° before the dead centre. Compressed air is used for starting.

The normal speed of the Koerting two-cycle engines is from 80 to 140 revolutions per minute; this speed can be considerably reduced, and the engine works very well even at 15 or 20 revolutions per minute, which is a great advantage in certain circumstances. Owing to this great elasticity the engine can be started and even put under load, and, in consequence of its method of admission, it possesses a very large overload capacity. These qualities render it particularly serviceable for blowing engines and rolling mills.

As the Koerting engine is built by different licensees there are several variations from the original form in evidence. The author would particularly mention those made by the *Seigener M.A.G.*, and the *Gutehoffnungshütte of Oberhausen*, who have been referred to several times in this book.

The *Gutehoffnungshütte Co.* employ slide pistons for the air and gas pumps operated by rods and separate levers. Governing is effected by the turning movement of external piston valves under the control of the governor. They also receive a reciprocating movement parallel to the axis of rotation worked by a hand lever. The admission passages of the gas pump are then opened, more or less, according to the quality of the gas employed and the load of the engine.

Mather & Platt, Ltd., Manchester, build Koerting engines for manufacturing purposes and for generating electricity. At the outset, they constructed them upon the original lines, but have now made a number of modifications in the way of simplification and for the better utilisation of producer gas and coke oven gas.

The gas pump is double-acting and is placed on the same axis as the two air pumps which are single-acting (Fig. VIII.—7). The pump cylinders are separated from each other by the admission boxes, each of which is divided into two parts by a partition, and fitted with automatic valves.

The gas piston is hollow. Its two faces are fitted with automatic valves and it is provided with openings in its periphery. The admission ports are placed in the wall of the cylinder in such a

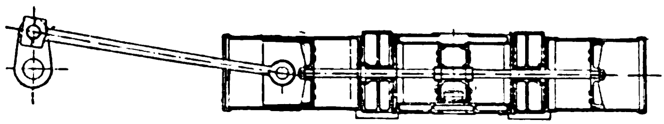


FIG. VIII.—7. Section through Gas and Air Pump, Mather & Platt Engine (Koerting type).

manner that the hollow piston places the gas supply pipe in communication with first one and then the other end of the pump cylinder. When the openings are masked by the piston, the gas must enter into the hollow portion by the ports and then pass through the automatic valves in the cylinder of the pumps at each suction stroke.

It is necessary to retain the gas charge until the air scavenging has been effected; therefore the position of the piston is given a certain

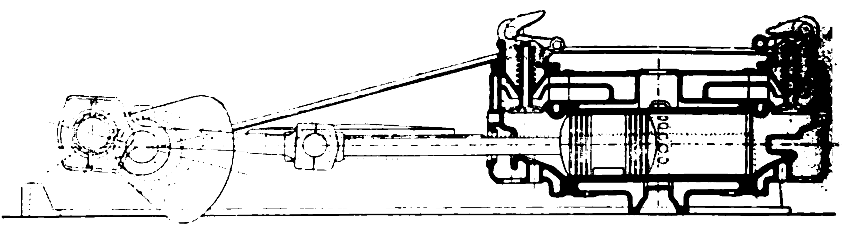


FIG. VIII.—8. Sectional elevation, Mather & Platt Engine (Koerting type).

amount of lead according to the calorific value of the gas used, in order to cover the openings and thus cut off the admission of gas. By the following movement the gas is compressed through the automatic valves into one or other of the valve boxes and from there expelled to the proper passage.

The different pistons are connected together by the same rod. The boxes forming the ends of the double-acting gas pump are placed respectively in communication with each of the engine inlet valves. These admission valves, as shown in Figs. VIII.—8 are mounted on a rod, passing through the valve boxes and surrounded by a coil spring which normally holds the valve to its seat. Above each valve rod is placed a pair of rolling levers. The lower one has a nose which rests on the end of the valve rod. As the admission valves should open only for just about one-fourth of a revolution of the engine crank, and remain closed during the other three-fourths, the contact surfaces of the levers are shaped so as to give a rapid movement to the

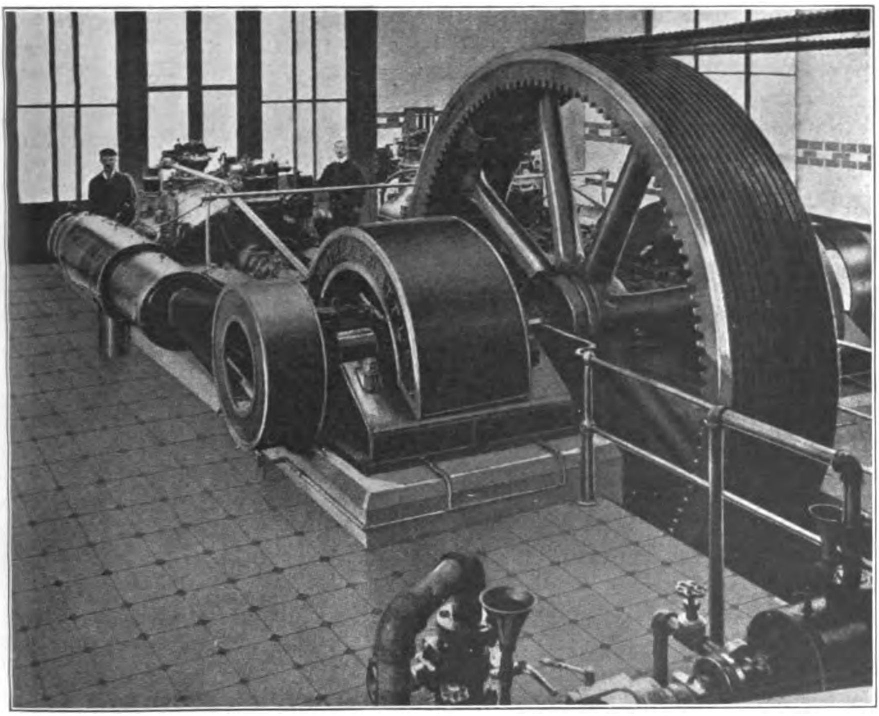


FIG. VIII.—9. 600 H.P. Mather & Platt Engine (Koerting type).

valve rods on opening and closing and a prolonged inaction for the remainder of the revolution.

The two pairs of levers are connected by a rod so as to work together in an opposite direction, one pair being closed when the other pair is opened. The levers are operated by connecting rods and simple eccentrics placed on the crank shaft and have means for adjustment. This patented arrangement dispenses with the usual side shaft with its cams and gearing.

Governing is effected by varying the quantity and not the quality of the mixture. Lubrication is controlled from a central distributer. Ignition is obtained from two magnetos placed at either end of the cylinder. Fig. VIII.—9 represents one of the most recent 600 H.P. engines by Mather & Platt, Ltd., distinguished by its extreme simplicity.

Oechelhäuser Engine.—The Oechelhäuser engine was one of the first designed for large outputs of power. It was put in practice early in 1898 in the form of a 600 H.P. engine, and has since been able to show its excellent qualities during an active life of nearly 11 years.

The characteristic feature of this engine is the employment of two pistons working in one cylinder, thus recalling the arrangements designed a number of years ago by Robson, and constructed by Scott

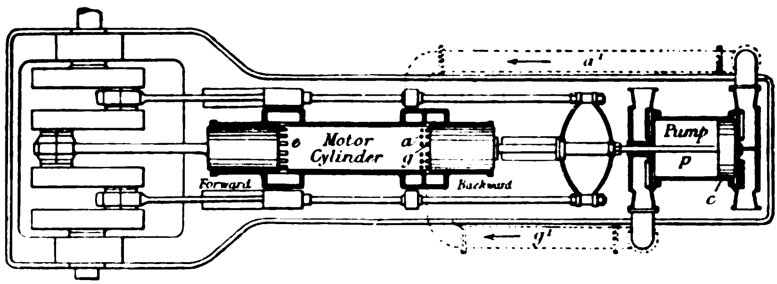


FIG. VIII.—10. Diagram of Oechelhäuser Engine.

Brothers, of Halifax (England). The Robson engine was of the four-cycle type with valve gear, whilst the Oechelhäuser engine is of the two-cycle principle with no valves whatever for the cylinder. This peculiarity is the more interesting as in all other large engines the valves are delicate organs, difficult to keep in good order in consequence of the high temperatures they have to withstand within the combustion chambers. The same remarks apply equally to stuffing-boxes and breech ends which are non-existent in the Oechelhäuser engine.

The diagram Fig. VIII.—10 shows the method of working of the two single-acting pistons. The front one is connected to the central throw of a triple crank shaft by a connecting rod and *pushes* the crank forward under the influence of the explosion in the ordinary manner, whilst the back piston is connected by cross bar and side rods to the two outer cranks placed at 180° and *pulls* the crank backward on power strokes.

The first advantage of this arrangement is that it produces a perfect balance of moving parts, and the equilibrium is particularly favourable on account of the neutralisation of working stresses in the crank-shaft bearings. Another advantage is that the combustion chamber is formed by the cylinder itself between the backs of the pistons when these have been brought together at the time of ignition. This explosion chamber is thus an enclosure presenting a minimum of surface cooled by the circulation of water and devoid of all passages and recesses which might be obstacles to the movement of the gas at the time of explosion.

These arrangements ought to have a manifest influence upon the thermal efficiency of the Oechelhäuser engine. They explain the favourable results obtained by an engine constructed by *Borsig* of Tegel. This engine of 500 H.P. has developed 1 I.H.P. hour with an expenditure of 6,350 B.Th.U., the thermal efficiency being 39·5 per cent., and the mechanical efficiency 83 per cent.

The valve action is obtained by the pistons themselves forming slides

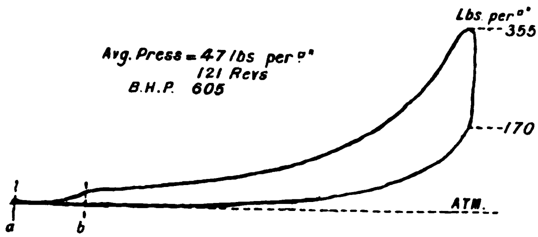


FIG. VIII.—11. Representative indicator diagram from Oechelhäuser Engine.

which alternately uncover and obstruct annular ports provided in the cylinder walls (Fig. VIII.—10), serving for the admission of air (*a*) and of gas (*g*), as well as for the expulsion of the products of combustion (*e*).

Before the pistons reach their dead centres on the outward strokes, the first piston uncovers the exhaust ports (*e*); then the back piston uncovers those for the admission of air (*a*), which, entering under pressure, scavenges the cylinder before the ports (*g*) admitting the gas are uncovered. The gas then mixes with the air to form the explosion charge. On the following stroke the pistons are brought back towards the middle of the cylinder, and, having first closed the inlet and exhaust orifices already referred to, they compress the charge previous to the commencement of another power stroke.

These operations are therefore conducted upon the two-cycle principles, (1) compression, (2) expansion. Admission and exhaust only occur during a fraction of the cycle, the air and gas being admitted under pressure.

The double-acting pump p (Fig. VIII.—10)—the piston of which is mounted upon a prolongation of (or independent of) the back motor piston—supplies both air and gas to the main cylinder. The front of the pump piston sends gas through the passage g' and air through the passage a'.

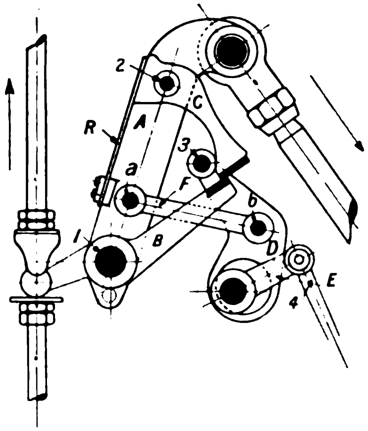


FIG. VIII.—12. Governor Trip Gear, Oechelhäuser Engine.

These passages always contain sufficient air and gas under pressure (the former at about 9 to 11 lbs. and the latter 6·5 to 8 lbs. per square inch) to fill the working volume of the cylinder, which corresponds to about 70 per cent. of the total volume. The difference in pressure between the two fluids is required by the exigencies of distribution.

The indicator diagram takes the form shown in Fig. VIII.—11, in which the exhaust, air blast and gas mixture admission takes place between the points a and b.

Governing is by variation of quantity of mixture. The compression is variable, but in order not to produce a vacuum the entire charge is introduced into the cylinder. The governor then regulates the opening of a return valve which permits a portion of the mixture admitted to be stored in an auxiliary receiver. The valve is operated by the medium of two levers A and B moving round a point (1) (Fig. VIII.—12), the movement of one being communicated to the other by the fulcrum block (C) which can oscillate round the point (2). Against this fulcrum block, at the point (3), bears the roller of the lever D mounted on an eccentric (4) forming a part of the lever E which is operated by the governor. The levers D and A are tied both together by the rod F fixed respectively at a and b.

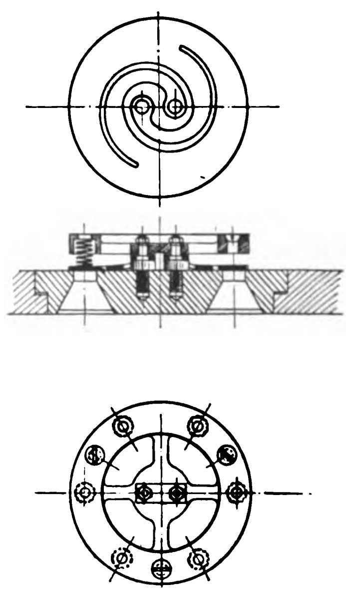


FIG. VIII.—13. Gas and Air Pump Valves, Borsig Co.'s "Oechelhäuser" Engine.

It will be seen that, according to the position of the eccentric (4), the roller (3) forces the fulcrum block *C* more or less to the right, and sooner or later actuates the trip gear at the end of the lever *B*. The sprag *C* is maintained by a spring *R*.

The Borsig Co. have fitted their new pumps with the type of valves used in their air compressors and steam engines. The automatic suction and delivery valve (Fig. VIII.—13) consists essentially

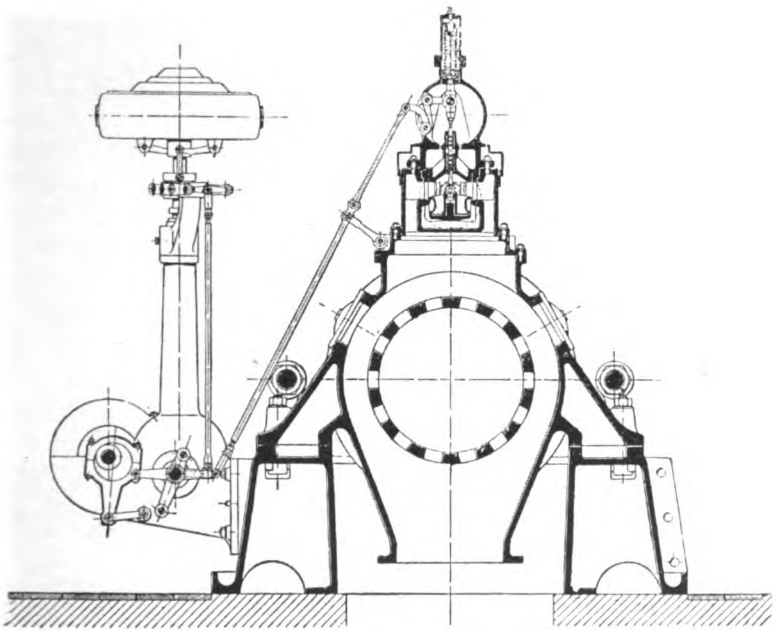


FIG. VIII.—14. Transverse section through Cylinder and Valve Box, Oechelhäuser Engine.

of a very thin steel sheet, about $\frac{1}{64}$ inch thick, weighing about $1\frac{1}{2}$ ozs., and stamped so that the surface presents two spirally radiating arms. This plate is attached to the centre by means of two screws. Above the plate forming the valve is a circular support against which bear several spiral springs. These springs are to load the valve and to press the disc against its seat. The circular support is fixed to the middle of the valve plate and thus the disc is forced to take a convex form.

With a view to being able to modify the areas of admission of gas and air, some slide valves are arranged round the admission slots. These slides are generally regulated by hand. However, the Borsig

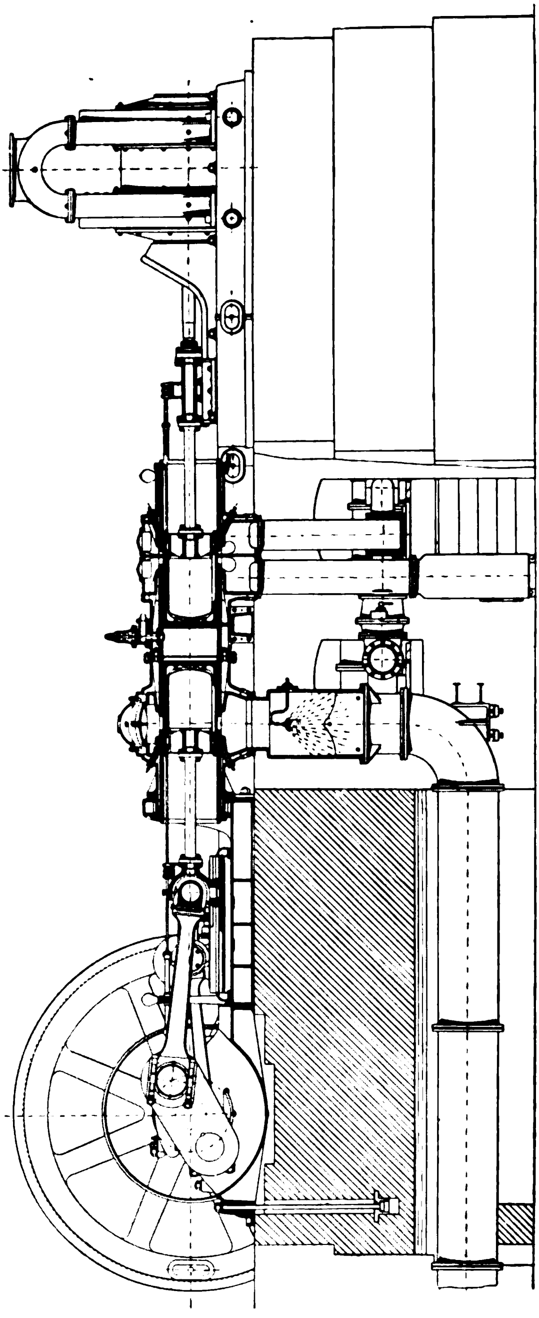


FIG. VIII.—15. Longitudinal section, Oechelhäuser Engine, with Blowing Cylinder.

Co. advocate that the gas slide valve should be controlled by the governor.

Fig. VIII.—14 shows a section through the valve chamber, Fig. VIII.—15 a longitudinal section, and Fig. VIII.—16 a plan. The two latter represent an engine with a blowing cylinder (on the right) for blast furnace work. The gas and air pump is shown on the plan in dotted lines.

The *Deutsche Kraftgas Gesellschaft* have built Oechelhäuser engines since 1899. Several other important firms build them under licence or on their own account having acquired the patents. It is in this way that about sixty of these engines, equivalent to a total of 50,000 H.P., have been installed in different countries, being used more particularly in metallurgical industries.

Two-Cycle Buckeye Engines.—With a view to obtaining higher mechanical efficiencies the Buckeye Engine Co. of Salem (Ohio, U.S.A.) have thought of utilising the pistons themselves and their crossheads as gas and air pumps. The engine works on the two-cycle principle, single-acting, with twin-cylinders. It is built for powers varying from 25 to 500 H.P. Above that power the Buckeye Co. construct double-acting two-cycle, and double-acting four-cycle engines.

The engine represented in Figs. VIII.—17 and 18 consists of two cylinders with one crank with two throws set at 180°, two gas and two air pumps. Fig. VIII.—17 is a longitudinal section of the engine. The piston (2) performs a double duty. It transmits the motive force to the crank and also compresses the mixture in the chamber (18) before its delivery in the explosion chamber.

The crosshead of the piston (6) is in the form of a plunger and acts as an air pump and compressor piston in the chamber (7). The exhaust ports are opened by the piston as usual in two-cycle engines. These ports are shown at (14) in Fig. VIII.—17. Each motor piston compresses its own explosive mixture and each crosshead plunger delivers the compressed air for scavenging to the combustion chamber of the other half of the engine. The cycle of operations is as follows:— When the piston uncovers the exhaust ports, the scavenging valve (11) is opened, and compressed air at about 8 lbs. per square inch is admitted to the combustion chamber from the air pump of the other cylinder. This air scavenges the burnt gases; the admission valve (10) is then opened, admitting a charge of gas and air from the compression chamber (18); this charge is also at a pressure of about 8 lbs. per square inch. The motor piston further compresses the charge which is fired by an electric igniter. Whilst the piston travels back to compress the charge in the cylinder, it draws a supply of fresh mixture

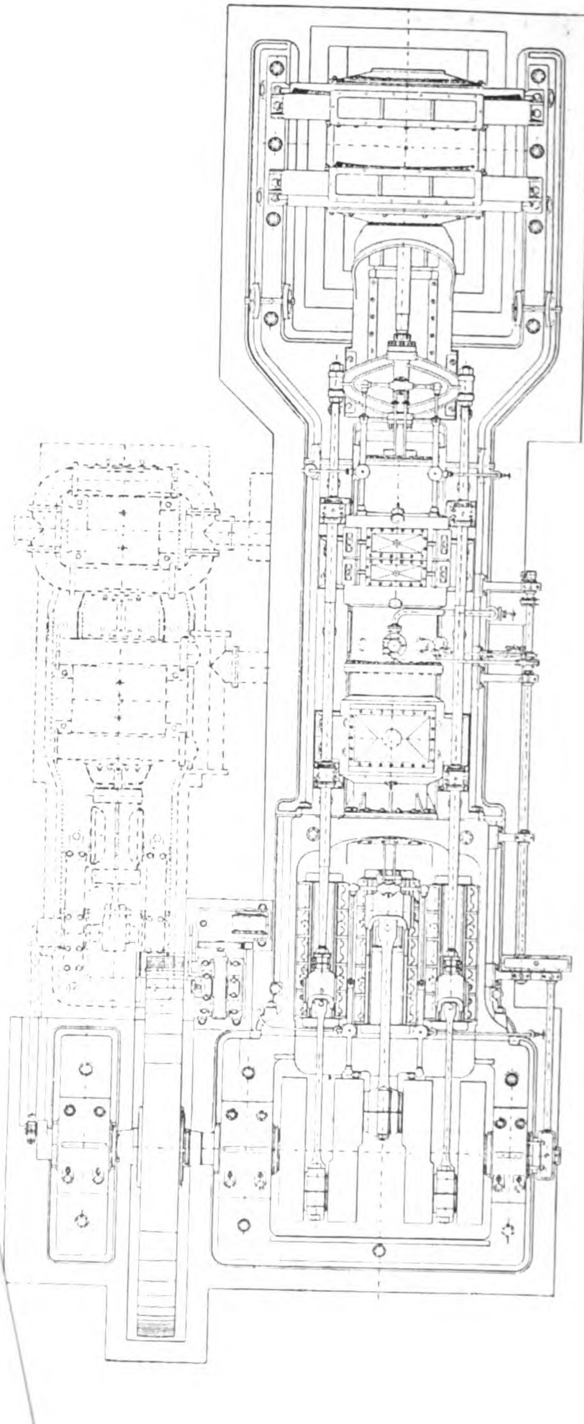


FIG. VIII.—16. Plan, Oechelhäuser Engine, with Blowing Cylinder.

into the chamber (3), and the crosshead plunger similarly draws in a charge of fresh air. The delivery of air from the chambers and passages (7, 20, and 19) is controlled by the valve (11), whilst the admission of air by the piston plunger (6) is regulated by a piston fuel valve (63) (Fig. VIII.—18), which takes the air from the chamber (68) in

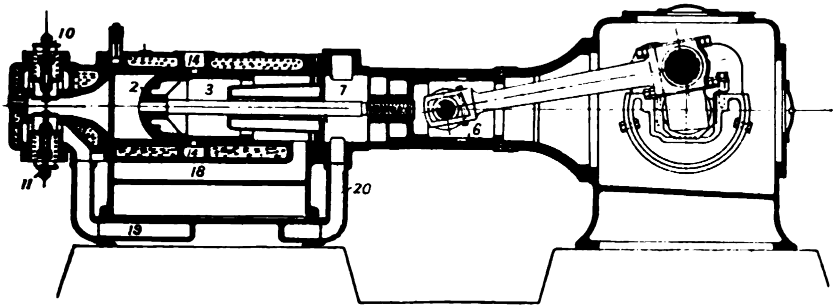


FIG. VIII.—17. Diagram of Buckeye Engine.

communication with the atmosphere by the base of the engine and sends it to the chamber (69) which is in communication with the piston plunger. The piston valve (62) takes in a mixture of air and gas through the chamber (67), which is in communication with the fuel supply, and then passes the mixture to the pump (3) (Fig. VIII.—17)

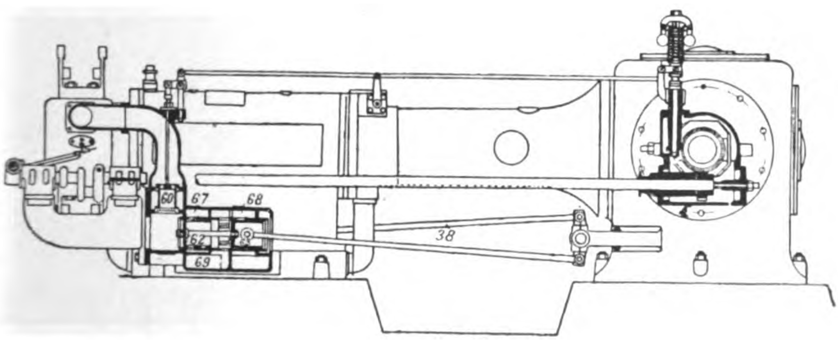


FIG. VIII.—18. Elevation, Buckeye Engine.

through a balanced throttle valve (60) (Fig. VIII.—18). The mixture pump then forces the charge back through the throttle valve (60) to the principal admission valve. This throttle valve (60) under the action of the governor thus regulates the quantity of mixture drawn in by the pump (3) as well as the quantity of mixture passing by the principal valve (10). The mixture and air scavenging valves are actuated by connecting rods worked by a side shaft. The rod (38) operates the corresponding valves for the other half of the engine. The admission and scavenging valves are actuated by rolling levers.

The centrifugal governor adjusts the position of the balanced valve (60) and thus controls the passage of the mixture. As the cylinder is filled with scavenging air at every stroke, the variation of the quantity of mixture introduced does not vary the compression, but modifies the richness of the mixture. The engine is provided with both high tension and low tension igniters. The valve gear can be adjusted to vary the compression and expansion in the motor cylinders. This changes the pressures of compression and of exhaust. Thus, for instance, with a valve closing at mid-stroke the gas will be expanded to one-and-two-thirds times its original volume if the free space were one-fourth of the piston displacement.

The piston can be similarly adjusted so that the mixture after admission in the motor cylinder is still 4½ to 5 lbs. above the atmosphere. This capacity is thus increased to the detriment of the thermal efficiency. Trials made with a 75 H.P. engine have shown a mechanical efficiency of more than 85 per cent., according to the makers.

CHAPTER IX

FOUR-CYCLE ENGINES

In the previous chapter the characteristic features of the few two-cycle engines in vogue have been dealt with, but, owing to the great number of four-cycle engines made, it is clearly impossible to mention all of these in a similar manner. As a matter of fact the various designs are adequately treated throughout the remaining chapters, and it only remains to give here a few details of those engines of the four-cycle type that possess features of special interest.

Dugald Clerk Supercompression Engine.—*Mr. Dugald Clerk* has endeavoured to produce supercompression by means of air or of products of combustion. The experiments of Mr. Petavel have confirmed the idea, advanced by engine makers, that the loss of heat by the walls is not increased proportionately to the pressure or the density of the gaseous mixture before explosion. With a mixture of 100 units of density, the loss of heat is only six times the loss corresponding to the unit of density.

The greater the diameter of engine cylinders, the greater is the difficulty of maintaining the temperature of the combustion chamber within proper limits during the explosion of the charge, and this leads to the conclusion that it would be advantageous to reduce the diameter of the cylinder and augment the pressures on the condition that the maximum temperature would not increase at the same time.

Mr. Dugald Clerk has modified an engine to permit the addition of air or cooled exhaust gas under pressure into the cylinder after the ordinary charge has been introduced during the suction stroke. Some tests made with the addition of cooled products of combustion gave an increase of efficiency of 31·5 to 34 per cent. per I.H.P. These results were so satisfactory that the National Gas Engine Co., Ltd., of which Mr. Dugald Clerk is a director, decided to build an engine of 300 H.P. to work with gas of about 120 B.Th.U. per cubic foot.

In this engine, Fig. IX.—1, the back of the cylinder is of the ordinary four-cycle type, but the front of it is made to act as an air pump. The frame is closed by a cover through which the piston rod passes. The air valve for the pump is placed above the cylinder, and is actuated by the cam shaft to admit air during the exhaust stroke of the engine; then the valve closes and the air is compressed in the clearance space between the piston and the front of the cylinder on the following stroke to about 15 lbs. per square inch.

The piston, at the end of the suction stroke, uncovers the ports of the clearance space and admits air under pressure into the cylinder, and this raises the pressure of the working charge by about 7 lbs. per square inch. The air under pressure remaining in the clearances of the pump is used for scavenging on the following exhaust stroke, so

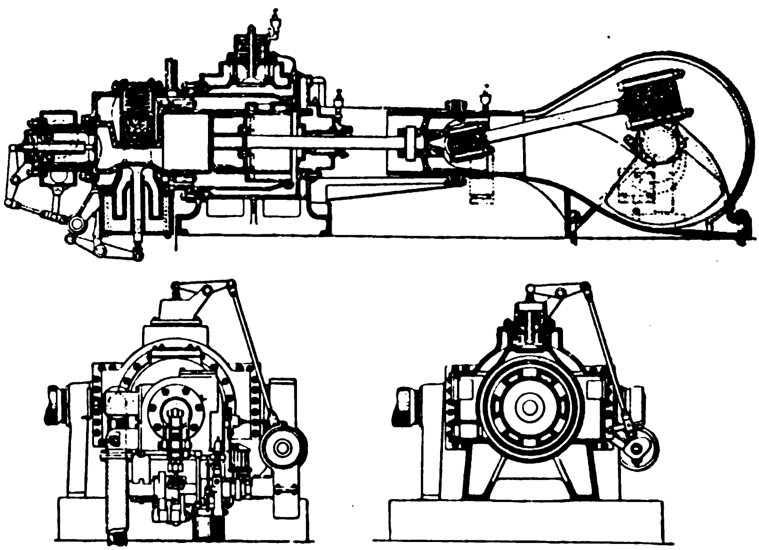


FIG. IX.—1. Clerk supercompression Engine.

that the engine has two advantages—supercompression and positive scavenging.

A mean effective pressure of more than 110 lbs. per square inch has been obtained, whilst the maximum temperature never exceeds 1200° C. (2192° F.). The working of the engine is very smooth and an explosion can be taken at every cycle without fear of overheated parts, and without requiring the piston to be water-cooled. The speed of the engine is 160 revolutions per minute, the diameter of the piston is 21 inches, the stroke is 36, which, with the mean pressure mentioned, gives about 270 I.H.P.

Mr. Dugald Clerk's theory is very attractive. It is evident that, in the present state of construction, it is necessary to tend to increase mean pressures in order to reduce, for a given power, the cost of building the engines. If, by the aid of Mr. Clerk's theory, this becomes practicable without proportionately increasing internal temperatures so difficult to deal with, a big step will have been made in the path of improvement of gas engines.

Sargent Engine. — The inventor of this engine (Fig. IX.—2) has endeavoured to obtain the expansion of the burnt gases down to atmospheric pressure and to vary the duration of admission in proportion to the load. The instant of ignition is further advanced as the mixture becomes weaker.

In the ordinary four-cycle engine, the piston at full load draws in a full charge of combustible mixture which, after compression and

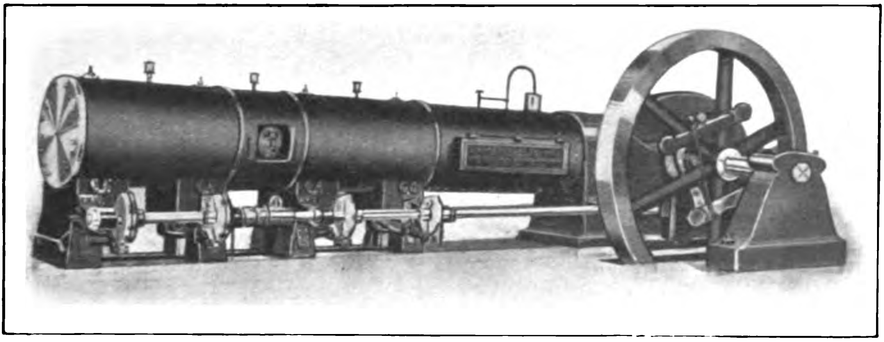


FIG. IX.—2. Sargent Gas Engine.

ignition, is expanded to its original volume and discharged at a pressure of 35 to 45 lbs. per square inch and a temperature varying between 750 and 900° F. In the Sargent engine, the admission of gas and air is cut off, at full load, between five-eighths or three-fourths of the admission stroke, according to the fuel employed. After compression and ignition the gases are expanded to the volume of the cylinder and discharged a little above atmospheric pressure with a corresponding temperature. The heavy line shown in the diagram (Fig. IX.—3) is that of the ordinary four-cycle gas engine at full load, while the thin prolonged line shows the increase of power obtained with the same fuel completely expanded in a Sargent engine.

The point of cut-off, which is constant for full load, is made to occur earlier by the governor when the load falls off, thus giving an elasticity which is impossible with ordinary internal combustion engines. Engines of this type below 100 H.P. require no cooling of pistons or rods. Fig. IX.—4 shows two diagrams taken with a light spring. In these the points of admission and cut-off will be noticed,

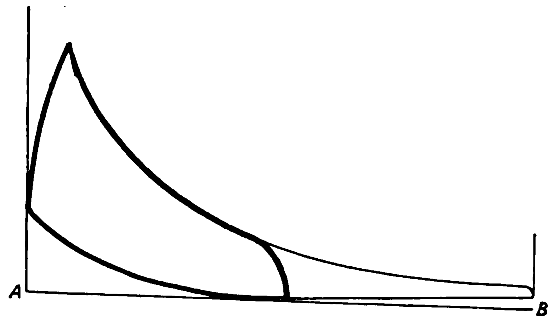


FIG. IX.—3. Comparative indicator diagrams of Sargent and ordinary Four-cycle Engines.

and it will also be observed that the pressure at the end of expansion is nearly that of the atmosphere.

The mechanism of the Sargent engine is very simple. It consists of one side shaft driven by the crank shaft, and carrying two cams for each explosion chamber, one for ignition and one to actuate the valves. Fig. IX.—5 shows a section through the valves and one of the explosion chambers. By unscrewing six nuts the valve boxes can be removed.

The gas is brought to the chamber *A* and air to the chamber *B*,

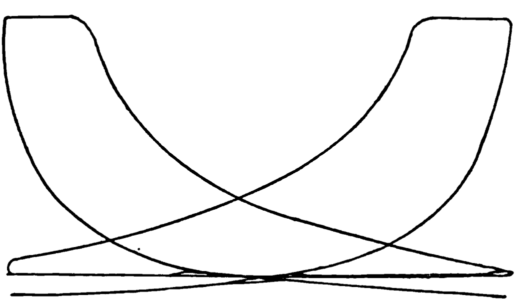


FIG. IX.—4. Representative indicator diagrams taken with "light" springs from Sargent Gas Engine.

entering the mixture chamber when the openings *F* of the piston valve correspond with the openings *E* and *D* of the gas and air chambers. The lift valve and the slide valve remain in the normal position during compression, ignition, and expansion. From the point L, the cam depresses the roller and raises the cylindrical slide which

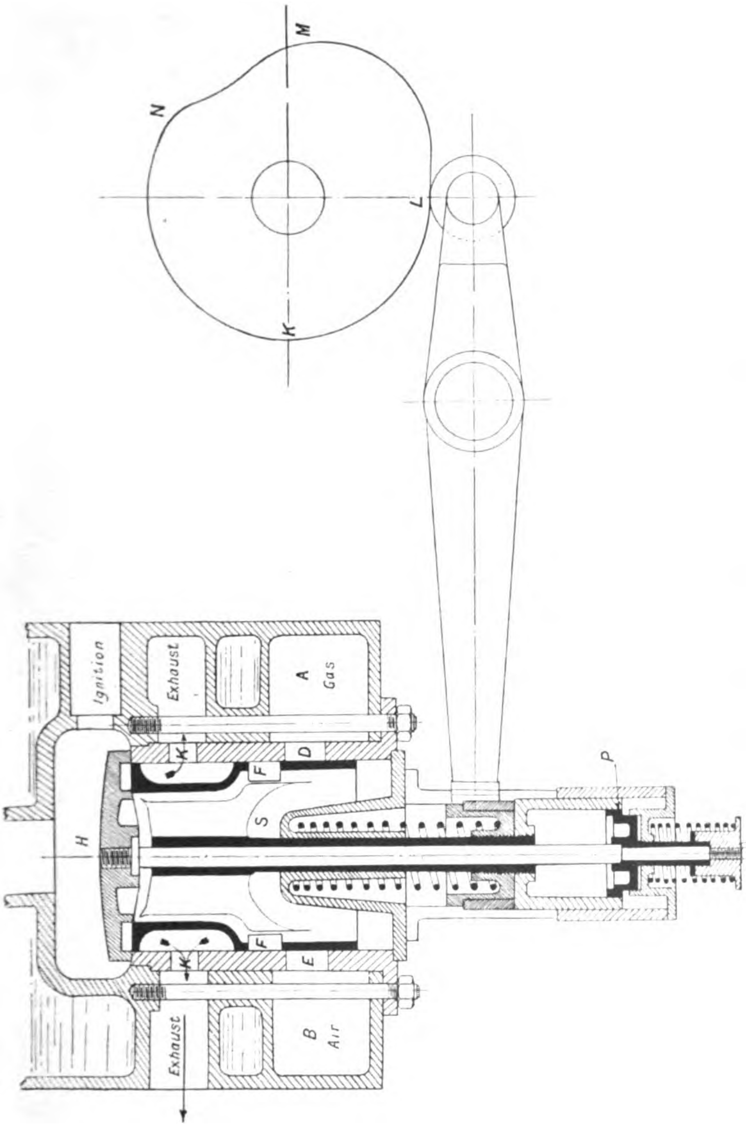


FIG. IX.—5. Section through Valve Chamber, Sargent Engine.

opens the valve, and the exhaust gas escapes by the openings K. At the points M and L of the cam the piston valve remains in the same position. From M to N it descends, and the ports F correspond with the openings *E* and *D* of the chambers *A* and *B*. When the piston valve descends towards this position, the air contained in the dash-pot *P* forces the valve *A* to lift and to admit a fresh charge, whilst the other end of the lever is raised and moves the piston valve to permit the admission. The lift valve thus controls admission and exhaust and remains sufficiently cool. The fly-wheel governor acts upon the piston valve *S* so as to regulate the mixture according to the quality of gas. The governor advances the valve shaft when the speed has a tendency to increase, and thus diminishes the mean pressure with the load. When the load decreases the mixture is cut off sooner, thus admitting a smaller quantity of constant composition in the cylinder. But, as the products of combustion are not decreased, the mixture becomes weaker and fires more slowly.

The following figures relate to a test made upon a Sargent engine fed with natural gas. The engine was 10 inches diameter of piston with a stroke of 20 inches; the normal speed was 200 revolutions per minute. The average load amounted to 50·64 H.P. The I.H.P. was 65·82; the mean pressure was 43·5 lbs. per square inch; mechanical efficiency 83 per cent.; consumption per H.P. hour was 10,800 B.Th.U.; and the B.H.P. thermal efficiency about 23 per cent. As will be seen, with the system of governing adopted the mean pressure is excessively low, involving the use of engines of large dimensions for a given output.

Trinkler Engine.—The Trinkler engine, like the Diesel, belongs to the class of heat engines in which the finely pulverised liquid fuel is injected into the combustion chamber containing air previously compressed by the piston. As built by Messrs. Koerting Brothers, the Trinkler engine is horizontal. Vaporisation of the fuel is obtained by a current of air at a higher initial pressure to that of the compression in the engine cylinder. The method of working is as follows:—The motor piston *A* (Fig. IX.—6) on its outward movement draws in pure air through the admission valve. On the reverse stroke the air is compressed to about 400 to 425 lbs. per square inch. The chamber *D* behind the injection piston *C* becomes filled with compressed air by the passage *E*, as also the passage *G* and the injector tube *F*. The fuel enters the vaporising valve chamber by the tube *K* and valve *I*. Towards the end of the compression stroke of the main piston *A*, the piston *C*, in consequence of the difference in the areas, is moved outwards closing the passage *E* and thus shutting off the air in *D* from the main charge. The piston *C*, however, is kept in motion by its rod which is operated by the lever *Q*. The movement of this lever, by means of a trip gear actuated by the cam shaft, ensures that the piston *C* is put in motion backwards at a suitable time. By means of a small hand wheel placed on the second motion shaft this movement can be adjusted so as to vary the instant of ignition.

When the lever is liberated by the trip gear the piston *C* moves

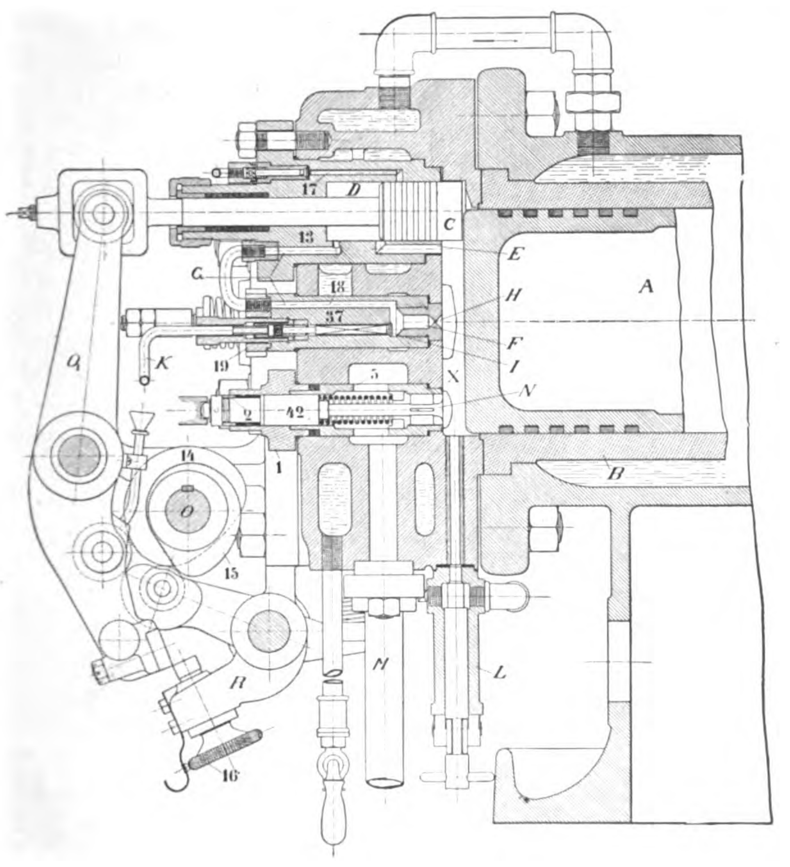


FIG. IX.—6.—The Trinkler (Koerting) Oil Engine.

under the difference of pressure. Air is forced from the chamber *D* through the admission tube *G* and injector *F* into the combustion chamber. As the opening of the injector is very small and the movement of the piston *C* is very rapid, the air is driven with great velocity, pulverising the fuel which it sweeps into the combustion chamber. The mixture ignites spontaneously in consequence of the high temperature of the compressed air. A little before the end of the power

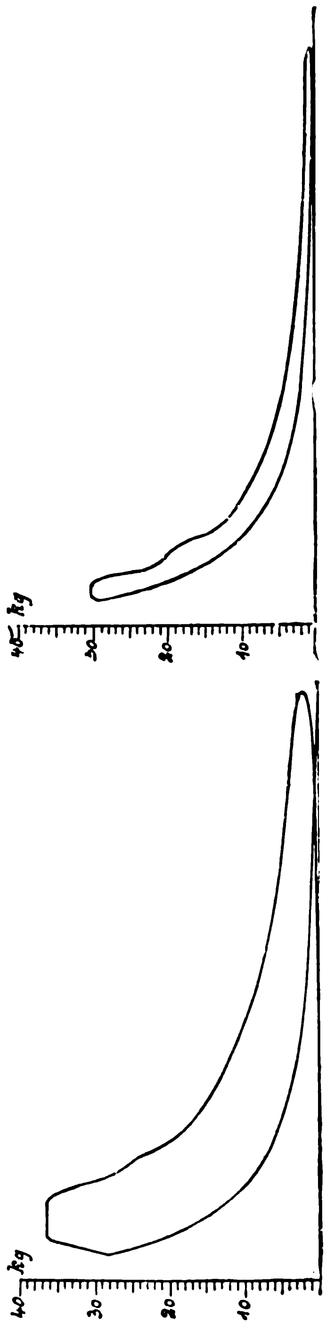


FIG. IX.—7. Full and half-load indicator diagrams, Trinkler Engine.

stroke the exhaust valve opens, and during the return movement the burnt gases are expelled. The engine is started by compressed air admitted to the cylinder by the valve *N*. The consumption of fuel is comparable to that of the Diesel engine.

From official trials made upon a 12 B.H.P. Trinkler engine it was found that for continuous work the consumption of petroleum was ·485 lb. per B.H.P. hour with Russian oil of 17,390 B.Th.U per lb., giving a thermal efficiency of 29·8 per cent.

At half load, the consumption was ·525 lb. per B.H.P. hour, which is but 7 per cent. more than at full load. Fig. IX.—7 represents the indicator diagrams taken from the engine. At full load the mean pressure was 120 lbs. per square inch, and the initial pressure about 540 lbs. per square inch. Running light, the initial pressure reached 430 lbs. per square inch.

Vogt Engine.—This engine works on the two-cycle double-acting principle, and has already been described by several authors, and particularly by Mr. Horace Allen. A certain quantity of water is interposed between the piston and the gas. The working pressure therefore acts on a liquid surface which transmits the power to the piston and crank shaft. The presence of the water obviates the use of cooling water or lubrication. Several important firms have undertaken the construction of this engine, but as far as the author is aware, no particulars have yet been published giving details of any tests.

However attractive the principle upon which the engine is designed may be, it is very necessary to know that great difficulties arise in practical work. It is particularly indispensable that the working parts shall remain gastight and that they shall resist the disturbances which must result from the alternating movements of the water at a certain speed, and also that any irregularity of ignition shall be avoided that might be occasioned by the special circumstances under which the explosive mixture is compressed.

CHAPTER X

THE WORKING OF GAS ENGINES

PRINCIPAL OPERATIONS

Ignition, compression, and cooling are essential details common to all gas engines. The general principles which govern these three operations appear now to be definitely established and only require elaboration in detail. After dealing with these matters, the author will refer to *lubrication, starting apparatus*, &c., *mean pressure, efficiency, utilisation of waste heat of exhaust*, and *silencing*.

Ignition of Gaseous Mixtures.—The study of the surest means of producing in a regular and normal manner the ignition of explosive mixtures, drawn in and compressed in gas, oil, and spirit engine cylinders, is one that has greatly occupied the minds of gas engine builders. Each of the systems hitherto adopted has found warm partisans and uncompromising detractors. These systems, for the most part, have been utilised in industrial work. They each present advantages and disadvantages, and it is illogical to expect from either a perfection which never can be entirely realised.

The phenomena relative to the influence of the walls, of pressure, and the speed of propagation of flame, play an important part in combustion.

Some tests on explosive mixtures of coal gas and air in closed cylinders were made in 1905 by Messrs. Leonard Bairstow and A. Alexander, of the National Physical Laboratory of Teddington, and were mentioned by Professor John Perry in a paper presented to the Royal Society.

A cast-iron cylinder, of 10 inches internal diameter and 18 inches high, was arranged to receive a mixture of gas and air, previously prepared, at different pressures. The reservoir was fitted with a disc with which to agitate the mixture from the outside, and the ignition arrangements were such as to permit the electric spark to be obtained in different positions.

The pressures were registered by means of a standard gauge, and indicator diagrams were taken on a continuous sheet of paper. The composition of the gas used was as follows :

Methane . . .	CH_4	27·8	per cent. by volume.
Hydrogen . . .	H_2	42·8	,, ,,
Hydrocarbon . . .	C_nH_n	5·4	,, ,,
Carbon Monoxide . .	CO	11·5	,, ,,
Oxygen	O	0·1	,, ,,
Nitrogen . . .	N	12·0	,, ,,

Summary of Tests.

Series I.	Initial pressure, lbs. per square inch absolute.	Actual maximum pressure, lbs. per square inch absolute.
Constant mixture. Initial pressure varied. Ratio of Gas to air by volume, 0·17.	44·8	348
	34·5	270
	24·7	180
	14·55	112
	9·71	68
	7·18	47

Series II.	Initial pressure, lbs. per square inch absolute.	Actual maximum pressure, lbs per square inch absolute.
Constant mixture. Initial pressure varied. Ratio of gas to air by volume, 0·104.	44·7	238
	34·6	185
	24·7	126
	14·4	74
	9·5	46
	7·06	33

Series III.	Ratio of gas to air by volume.	Actual maximum pressure, lbs. per square inch absolute.
Varied mixture. Initial pressure constant at 34·5 lbs. per square inch absolute.	0·176	272
	0·156	263
	0·137	245
	0·120	216
	0·102	193
	0·0845	97

Professor Bertram Hopkinson, of Cambridge, has published the results of his tests relative to the combustion of gaseous mixture in closed vessels in an article appearing in *Engineering* on June 15, 1906.

These results enable the student to form an approximate idea of what transpires in the first three-tenths of a second after the ignition of an explosive mixture. The temperatures at different distances from the point of ignition vary widely. The time necessary for the propagation of flame is about one-thirtieth of a second. Combustion and consequent expansion of the gas, remote from the original point of ignition, produces a compression of the products of the initial flame, and therefore the heat due to combustion at that point is increased by the supercompression.

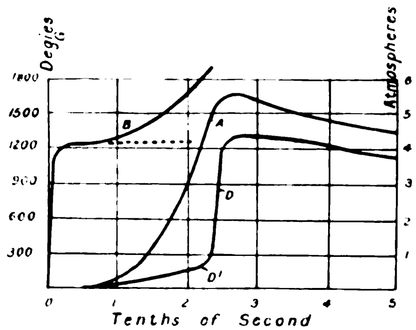


FIG. X.—1. Curve representing pressure-rise on combustion (Hopkinson's experiments).

The curve *A* (Fig. X.—1) represents the rise of the pressure due to the combustion of a mixture of town gas and air in the proportion of 1 to 9. The curve *B* shows the rise in temperature at a point *A* in the centre of the cylinder (Fig. X.—2). It will be seen that a steep curve is traced up to 1,200° C., and then a gradual rise coincident with the increase of pressure shown in curve *A*. The curve *D* shows the rise in temperature at the point *D*, at the end of the cylinder (Fig. X.—2). Professor Hopkinson remarks from these observations that, if a gaseous explosive mixture be ignited at discreet points, very great differences of temperature must always exist when the explosion is complete, and this quite apart from the cooling effect of the walls. Owing to the slowness of the propagation of flame, as compared with the velocity of the reaction at any point when the flame reaches it, the gas near a point of ignition will be burnt at low—and nearly at constant—pressure, and compressed after burning, whereas the last portion of gas to burn will be compressed before burning and not after. Thus the gas near the points

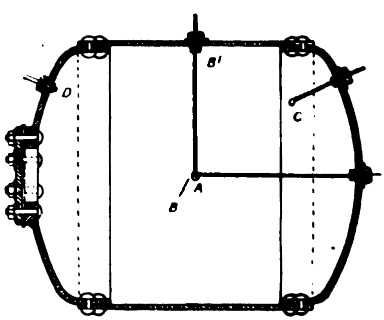


FIG. X.—2. Sectional drawing of Explosion Cylinder (Hopkinson's experiments).

of ignition will be much hotter than at a distance from such points. In the experiments described, the difference of temperature from this cause amounts to 500° C. (900° F.). When applying the results obtained from such experiments under static conditions in closed cylinders, it must be remembered that in practical gas engine work the "kinetic" condition of the gases will probably cause other variations of temperature.

Phenomena of Combustion in Gas Engines.—Mr. J. E. Junge, in discussing the phenomena of combustion in gas engines, states that, theoretically, ignition should take place sufficiently early to ensure that the whole of the mixture is in combustion when the piston arrives at the dead centre. At this moment, the maximum development of heat coincides with the minimum of cooling surface and in a space specially constructed to withstand high pressures and high temperatures.

When the mixture of gas and air contains just the quantity of oxygen necessary for complete combustion, the highest possible explosion temperatures are obtained. Above and below this ideal condition, the degree of inflammability becomes modified and the mixture burns in a manner more or less active, and consequently developes a higher or lower temperature. In practice, for gas engine mixtures more air is used than the quantity theoretically necessary, for the following reasons:—

(a) To reduce the temperatures to obviate premature firing which might be caused by the heat of high compression.

(b) To furnish the gas, even in weak mixtures, with sufficient oxygen to permit complete combustion.

(c) To reduce the losses from unburnt gas escaping through the exhaust, to a minimum.

Theoretically, the thermal efficiency of weak mixtures is at a maximum when the poorest mixture is compressed to the highest possible extent, but in practice, the upper limit is rigorously defined by the degree of inflammability of such mixtures, and falls short of the theoretical possibilities.

Gas engines burning poor gas generally work with a volume of air 30 to 40 per cent. in excess of the quantity theoretically necessary. With rich gas, a still greater excess is admitted when the average calorific value of the mixture in the engine is from 45 to 60 B.Th.U. per cubic foot.

The degree of inflammability varies considerably for different gases, as established by the researches made by Professor Eitner, in the laboratory of Carlsruhe Technical School, which gave the following figures:—

	Gas contained in the Total Mixture.	Corresponding degree of inflammability.
	Per cent.	Per cent.
Carbon monoxide	72	58·45
Hydrogen	67	56·95
Water Gas	67	54·35
Acetylene	51	48·95
Illuminating Gas	18	11·2
Acetylene	14	10·5

The quantity of air required to ensure inflammability is much less than that necessary to complete combustion. The mixtures employed in gas engines contain (beside the combustible elements and oxygen) nitrogen, carbon dioxide, and water vapour, and the degree of inflammability is thereby influenced. Conditions of pressure and temperature similarly intervene.

Tests recently made by Messrs. Le Châtelier and Boudouard have established that very weak mixtures of carbon monoxide and air become inflammable by a rapid heating. At normal temperature a portion of 16 per cent. of carbon monoxide forms the lower limit of the degree of inflammability, whilst at 400° C. (750° F.) this limit fell to 14·2 per cent.; at 490° C. (914° F.) it was 9·3 per cent.; at 600° C. (1112° F.) 7·4 per cent.

Carbon monoxide and hydrogen are the principal active constituents in all combustible gases.

The temperature at which hydrogen ignites is much lower than that necessary for the ignition of carbon monoxide. The rapidity of the propagation of flame at atmospheric pressure is about thirty times greater. Combination with the oxygen of the air is more rapid, and it requires a considerably greater excess of air to be present for safe working.

The ideal gas for the engine maker ought not to contain too much hydrogen in order that the engine may not be liable to premature explosions, but it should contain enough, however, to quicken the slower burning carbon monoxide and to accelerate the transmission of the flame throughout the volume of mixture.

Position of Igniters.—With combustion chambers of symmetrical and regular form, cylindrical or annular, the centre of the chamber corresponds to the ideal position of the igniter, inasmuch as the radial distance through which the flame should travel is not then too great. In the small and average sizes of single-acting gas engines it is generally an easy matter to place the igniter in the neighbourhood of this central position, and the high efficiency of certain engines must be attributed, to a large extent, to this arrangement. In the large double-acting engines the conditions are different. The distance that the flame has to pass through is very great and several igniters are necessary. It is almost impossible to get near the vicinity of the centre of the combustion chamber. Usually, therefore, it is placed near the inlet valve.

Some makers who use two igniters put one near the exhaust valve. This is not a suitable position for four-cycle engines, as in this vicinity it is likely to be surrounded by burnt products incompletely expelled

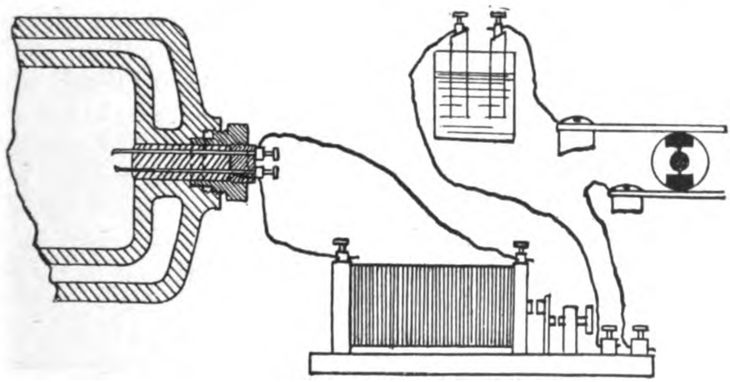


FIG. X.—3. Diagram of connections for electric ignition, high tension system.

and not with fresh mixture. It would be better to place the second igniter at a considerable distance from the exhaust valve and at a point about half-way between the inlet and exhaust valves.

Electric Ignition.—Lenoir adopted electric ignition in his engine, using a cell, induction coil, and sparking plug, and caused the ignition to occur about six-tenths of the stroke of the piston, that is, with a decided retardation.

The moment when ignition takes place is of great importance. Electric ignition is effected by means of a combination of three essential elements, comprising a battery, induction coil, and sparking plug (Fig. X.—3), or by means of a magneto producing the spark by mechanical "break" of the contacts through which the current is at the moment passing. The former arrangement was first used.

The battery may consist either of accumulators which require recharging from time to time, or of primary cells which need frequent renewals of materials and careful cleaning of detail parts. The induction coil is provided with a trembler, which easily becomes deranged, and which must be adjusted to a nicety. The sparking plug (Fig. X.—4) is a particularly delicate part and subject to many accidents.

The principal advantage of the system consists in the facility of adjustment, at will, of the instant of ignition. Its inconveniences are numerous. It demands scrupulous attention to its proper maintenance; its contacts are liable to become covered with oil and thus to prevent the production of the spark; the intensity of the spark depends upon the density of the explosive mixture, and falls when the compression is raised owing to the increased electrical resistance between the electrodes. The distance between the contacts is generally small and this gives a short spark, so that only a small number of particles of mixture are in immediate contiguity with it, and consequently, if the mixture be not very homogeneous, misfires occur.

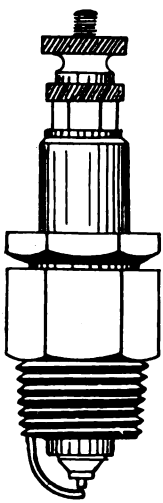


FIG. X.—4. Sparking Plug, high-tension current.

Everyone who has driven a motor-car, the engine of which is fitted with this system of high-tension ignition, knows the breakdowns that frequently result from its bad working.

(Dealing with the systems of ignition in chronological order, a description of the magneto will be found in later pages.)

Slide.—After the employment of electric sparking plugs, ignition was effected in the early types of gas engines by means of a travelling flame. This system was adopted in the engines of which the inlet of the mixture was effected by a slide valve. The latter is now completely abandoned, and requires no word of comment. The high compressions now universally employed could not be obtained with the old-fashioned slide valve.

Tube.—Tube ignition, which for many years has been exclusively employed by a great number of makers, consists of an arrangement comprising either a metallic or porcelain tube, the interior of which communicates with the explosion chamber, whilst it is maintained at a red heat by an external flame. The substance of the tube, the manner in which it is arranged, the distance at which it is placed from the explosion chamber, and the position of the incandescent zone, have a great influence upon the timing of ignition.

The author had a curious experience during one of his tests, in connection with the substance of the ignition tube. The engine was fitted with two adjacent tubes. The communication of each of them with the combustion chamber could be instantaneously intercepted at will. One tube was of porcelain, and the other was a metallic tube. With the Mathot explosion recorder the author obtained graphic records during work, first with one tube and then with the other, with the result shown in Fig. X.—5. It will be seen that the initial explosion pressure with the porcelain tube is less than 285 lbs. per square inch, whilst with the metallic tube it reaches 350 lbs. per square inch.

It must not be concluded from this experience that, as an absolute

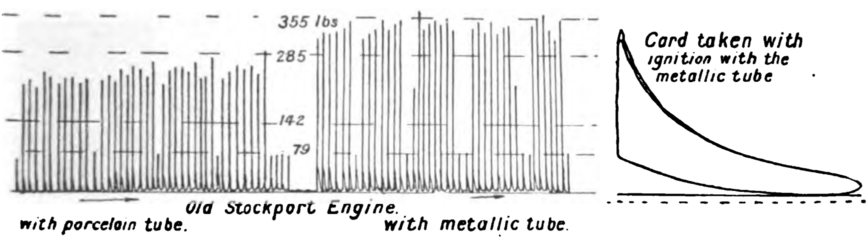


FIG. X.—5. Variation of explosion pressures with "tube" ignition.

fact, metallic tubes are preferable to porcelain tubes, but it is evident that, under the conditions peculiar to this test, the second tube gave better results than the first.

Figs. X.—6 and 7 show two arrangements generally adopted for ignition tubes.

The engine attendant is bound to adjust the heating of his tube in such a way that the gaseous mixture inside it reaches the incandescent zone at the precise moment when ignition and explosion should occur, namely, after the crank has passed the inner dead centre. This is not an easy thing to manage in practice, and very often the ignition device is adjusted in such a way that the combustion of the mixture occurs somewhat late, the attendant being prone to permit this, because early firing is then impossible and the working parts are protected from heavy and sudden shocks.

The difficulty of determining, with such an arrangement, the precise moment of ignition, has caused the makers for all gas engines above

20 to 25 H.P. to adopt an ignition valve which mechanically determines the instant of ignition. The tube is heated as much as possible, and the incandescent zone is brought down as near to the base as can be managed. The ignition valve is opened at the desired moment by the action of a cam fixed upon the half-speed shaft. In this way, the adjustment of ignition is made possible, and engines fitted with this arrangement and working with town gas, have usually given satisfaction as long as the compression of the explosive mixture does not exceed 100 to 110 lbs. per square inch.

There is good cause to look upon the timing valve as a detail which

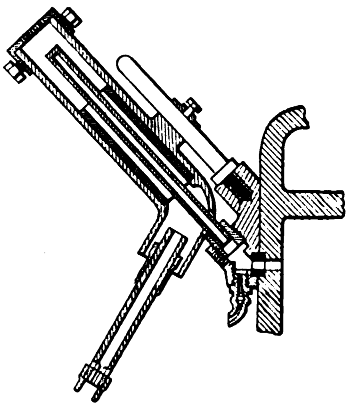


FIG. X.—6. Ignition device with metallic tube.

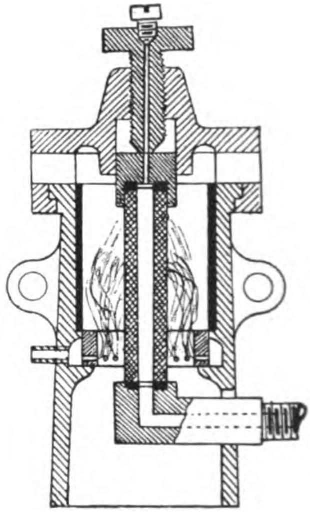


FIG. X.—7. Ignition device with porcelain tube.

deteriorates very rapidly. It is usually arranged in such a manner that it is impossible to cool its seat. It comes into actual contact with the flame at the moment when the highest temperature prevails, and therefore requires frequent renewal.

Ignition tubes have also been tried upon automobile engines in consequence of the combination of battery and sparking plug not having been found satisfactory. But, owing to the necessity of placing the burner, requiring to be fed with spirit, close to the pipes containing the working fuel of the engine, the risk involved was too great to run, and, for this reason, ignition tubes are deemed unsuitable for this class of engine. Electric ignition has since been greatly improved and is universally used.

The evolution of the gas engine during recent years, and particularly in connection with the greatly increased compression pressures and the weaker charges of mixture, has made the working of ignition tubes very doubtful. Experience has shown that their operation is satisfactory when the engine is fed with rich gas, or town gas, of about 600 B.Th.U. per cubic foot, but with producer gas of about 135 B.Th.U., or blast furnace gas of 100 B.Th.U., they are more or less irregular in action.

Magnetos.—The magneto electric machines have displaced the other

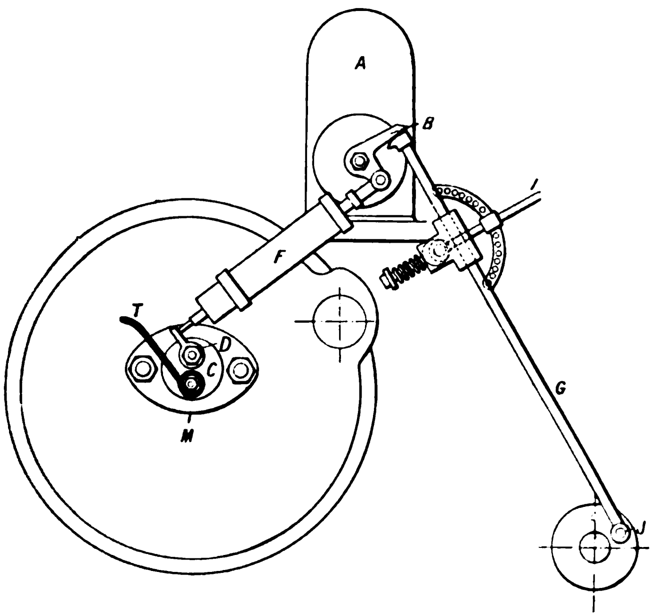


FIG. X.—8. Diagram of magneto low-tension ignition device (Winterthur).

forms of ignition devices previously mentioned. They are mechanically operated by a cam or eccentric fixed upon the half-speed shaft, connected with the armature in such a manner that the latter is slowly moved through a certain angle, and then, the levers slipping out of contact, the original position is quickly regained under the influence of a strong spring. This rapid motion creates a current which passes through the contacts arranged inside the cylinder, and by a mechanical movement the latter are separated suddenly, causing the circuit to "break," and, in doing so, to produce a spark. The contacts are usually made of steel, sometimes being tipped with nickel or platinum.

Fig. X.—8 shows the complete fittings for magneto electric ignition, as adopted by the makers of the Winterthur engine, the sparking plug with the contacts working in the combustion chamber being shown separately, in section, in Fig. X.—9.

The magneto *A* is made up of steel horse-shoe magnets between the iron pole pieces of which the armature oscillates. The armature spindle has a bell crank lever *B*, fixed by a nut, at one end. The cast-iron sparking plug is fixed to a movable flange bolted to the cylinder jacket, and carries two spindles, *D* and *M*. The spindle *D*, in gun-metal, is movable and is fitted with a hammer internally, and a small percussion lever and recoil spring externally. The spindle *M* is fixed and well insulated. It receives the current from the magneto *A* by means of an insulated copper wire *T*.

The spring *F* is made in two coils contained in a brass casing with a steel percussion pin. The operating gear of the magneto includes a spindle or actuating rod *G*, sliding through a guide provided with a safety spring and mounted on an eccentric pin. The range of movement of *G* can be varied by means of the lever *I*. The actuating rod is moved by the revolving pin *J* on the extremity of the cam shaft as shown.

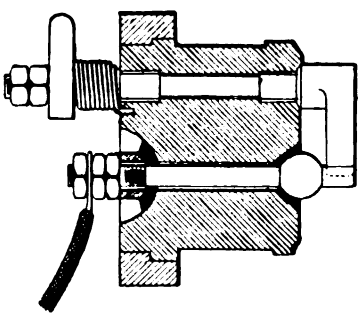


FIG. X.—9. Section through Sparking Plug (Winterthur).

The mechanical control of the sparking gear of a magneto entails some difficulties, the position of the apparatus having to be decided to permit ready access without interference with the valve gear and other details. M. Guillou has brought out a pneumatically operated sparking plug to obviate such difficulties, which allows the magneto to be placed at any convenient position above the half-speed shaft. The Dudbridge Ironworks, Ltd., of Stroud, have adopted the "White" apparatus shown in Fig. X.—10, which consists of a flexible tube filled with a series of cylindrical metallic pellets in contact with each other throughout the length of the tube, and ensuring thereby the transmission of movement from one end to the other. Other makers, notably the Bosch Co., obtain the "breaking" action by means of an electro-magnet. Figs. X.—11 and 12 show the "White" patent sight plug, usually supplied with the Dudbridge apparatus, by means of which the spark inside the cylinder is made visible to the operator.

By reason of their regularity of operation, magnetos are to be recommended. They work for many years without requiring to be remagnetised. They produce when released a very hot and fat spark,

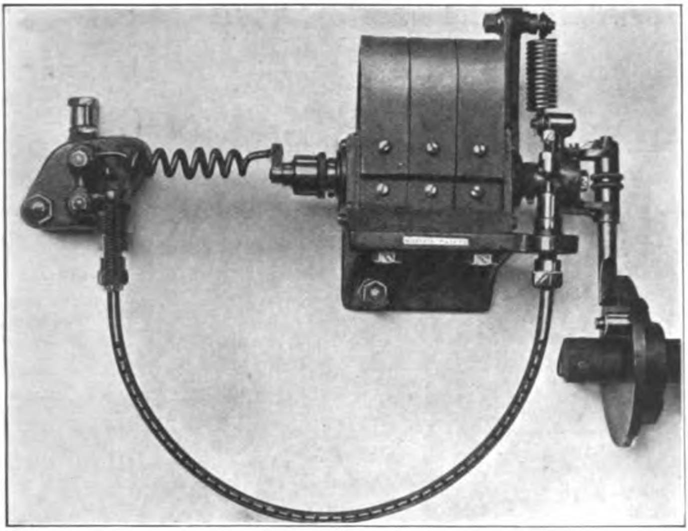


FIG. X.—10. Low-tension Magneto and Fittings (White's patent).

much more energetic than that produced by a battery and induction coil, and the gaseous mixture in the cylinder is fired with greater

FIG. X.—11.

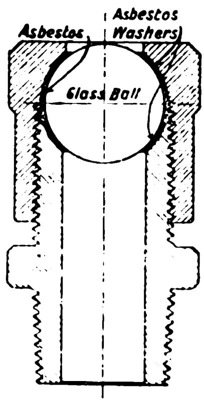


FIG. X.—12.

The "White" patent Sight Plug.

certainty. They answer well on engines working with high compression and served with poor gas. To maintain them in good condition it suffices to keep them from dampness and from splashes of oil. They require cleaning from time to time, while it is also necessary to protect them from excessive heat, which would cause them to lose their magnetism.

It is very important that the instant of the "break" which produces the spark should be capable of adjustment during the operation of the engine, and means to this end should always be provided. In the author's opinion some such provision is indispensable.

Most of the mechanism employed to operate the magneto and the sparking plugs comprises a system of levers operated by eccentrics or cams. Those furnished with means of adjustment are frequently

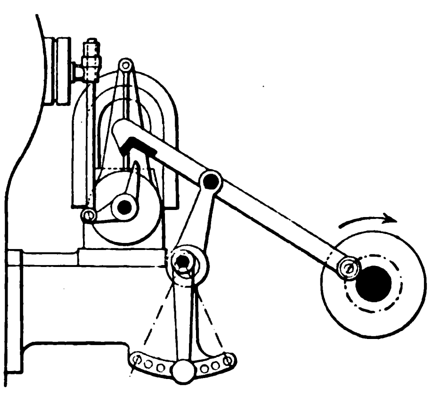


FIG. X.—13. Magneto mechanism, Gasmotoren Fabrik Deutz.

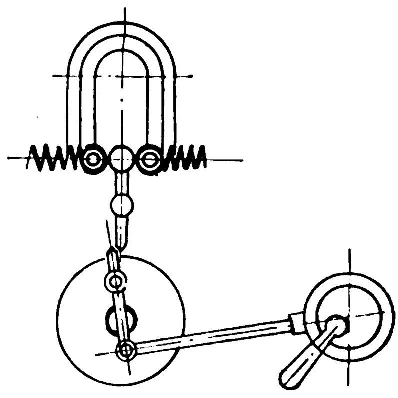


FIG. X.—14. Magneto mechanism.

defective in that they do not ensure an invariable length of stroke or oscillation of the armature. When ignition is retarded, the duration of contact between the gear and the armature is reduced, and thus a weaker current is produced which further accentuates the retardation. As a matter of fact, the contrary effect should be obtained, because when starting the engine to work with late ignition the spark should be more energetic, otherwise the weaker spark would cause the explosion to occur after the dead centre, and would tend to produce missfires. It is advisable, therefore, to adopt some method of adjustment which is not affected by the amplitude of movement, irrespective of the instant when ignition may be timed to occur.

The catch points of the armature and control lever have to withstand some considerable strains produced by the recoil springs, and require to be carefully tempered and provided with sharp edges. The Winterthur Co. provide a small lubricator to ensure con-

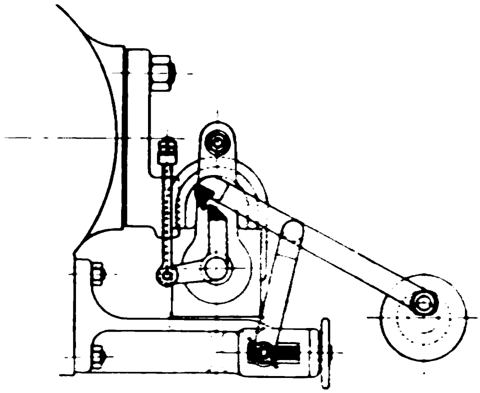


FIG. X.—15. Magneto mechanism, Tangye.

tinual lubrication of the catch point, an arrangement much to be commended.

The magneto armature is usually provided with springs to restore it to the neutral position after disengagement. These springs are either of the coil or laminated type, and are arranged and adjusted to counterbalance the armature and avoid undue wear of the bearings. For this reason magnetos fitted with a single spring are least suitable, for all the work is thrown upon the bearings of the armature spindle. They involve rapid wear and throw the armature out of centre, with risk of damage owing to the wiring becoming abraded by contact with the pole pieces of the magnets.

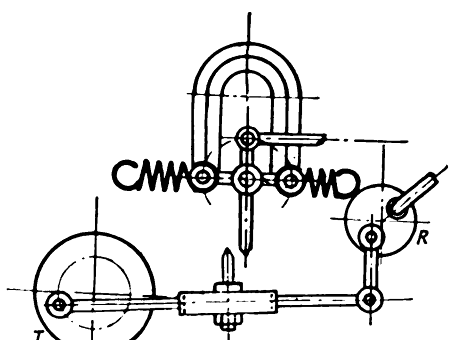


FIG. X.—16. Magneto mechanism, Gasmotoren Fabrik Deutz.

The ignition devices for magnetos are innumerable. They generally include, in the same mechanism, the movement that oscillates the armature, and that which, by the sharp recoil of the armature to its neutral position, causes the break between the contact pieces to produce the spark by the sudden cessation of the intensified current. The apparatus described below all work in this manner.

In Figs. X.—13 to 16 the mechanism consists of a connecting rod pivoted on a crank-pin projecting from a disc fixed at the end of the cam shaft. This connecting rod, either at its other extremity, as in Figs. X.—13 to 15, or in the centre, as in Fig. X.—16, carries a case-hardened point which describes an ellipse when in motion and momentarily engages with the contact fixed upon the magneto armature. The connecting rod has a movable support which can be raised

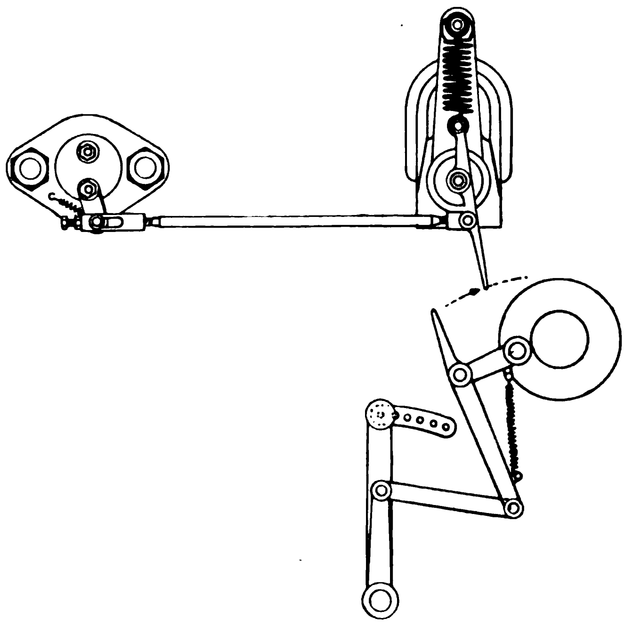


FIG. X.—17. Magneto mechanism, D. Stewart & Co.

or lowered by hand for adjustment of the instant of disengagement. In the arrangement shown in Fig. X.—17, an auxiliary connecting rod is connected to the cam shaft at one end, and to the oscillating catch point that engages with the magneto at the other. The amplitude of oscillation of this catch point is made variable by means of an arm and hand lever moving along a sector. The principal detail in Fig. X.—18 is a lever with one end in contact with a slip cam mounted on the half-speed shaft, whilst the other extremity is connected to a rod which operates both the contact points of the ignition plug and the magneto armature. The support of this lever is movable, and the instant of disengagement can be varied by means of the hand-

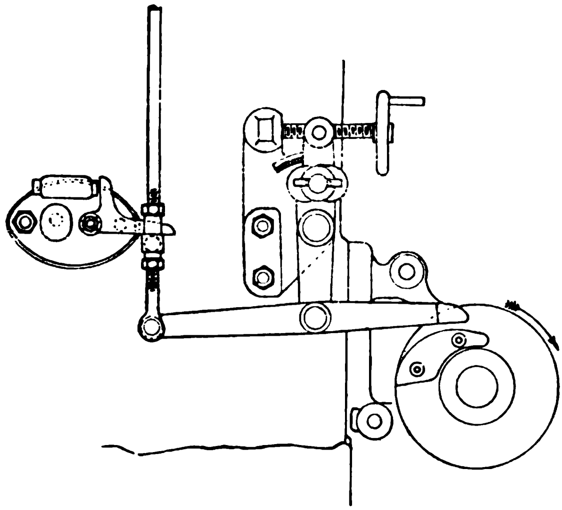


FIG. X.—18. Magneto mechanism, Tangye.

wheel and screw provided. The arrangements shown in Figs. X.—15 and 18 are employed by Tangyes Ltd., of Birmingham.

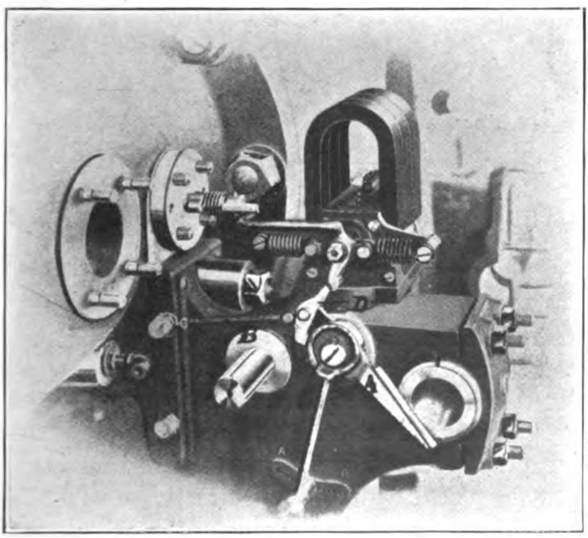


FIG. X.—19. Magneto mechanism, Olds Power Co.

Fig. X.—17, illustrates the method adopted by D. Stewart & Co., of Glasgow, and Landaal, of Appeldorn (Holland). Those shown in

Figs. X.—18 are used by many German makers, and particularly by the Gasmotoren Fabrik Deutz.

In order to obtain a compact arrangement with short and light transmission gear, the Olds Power Co., of Lancing (Michigan, U.S.A.), have adopted the arrangement represented in Fig. X.—19, under the advice of the author, the whole of the gear being mounted upon the cam shaft bearing bracket.

A is the operating lever, the free end of which is maintained in contact with a cam shaft, not shown in the photograph, by the coil spring *B*. The lever is mounted on an eccentric pivot which can be

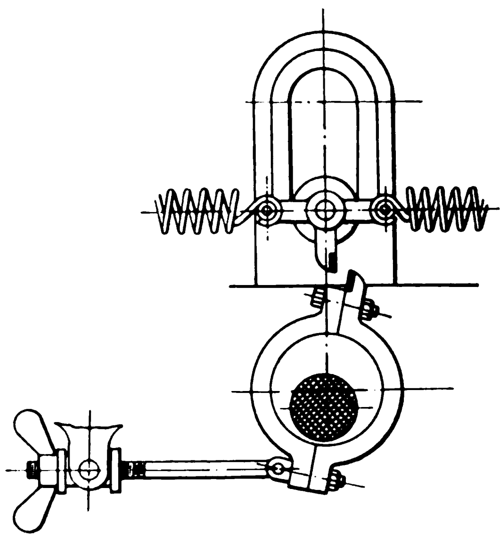


Fig. X.—20. Magneto mechanism, Kynoch.

moved by hand, by aid of the handle *C* on a notched dial. Near this pivot, and placed on the lever, is the contact piece *D* formed by a spring tumbler which becomes rigid when it is supported against the lever as long as it is in contact with the armature, and which fails to act when the engine is turned in the reverse direction. The armature carries a branch piece which acts directly upon the "break" mechanism of the plug shown in the side of the cylinder casing.

In all the arrangements described, the adjustment of the instant of disengagement at the same time influences, more or less, the duration or amplitude of the oscillation of the magneto armature, and increases both the relative space between the moving contacts and the movement of the armature. The gear shown in Fig. X.—17 has the advantage of reducing the amplitude very slightly when the disengagement is retarded. The arrangement adopted by Messrs. Kynoch, Ltd., of Birmingham (Fig. X.—20) has the same advantage, added to which is the fact that it can be placed anywhere along the cam shaft as well as at its extremity. It consists of an eccentric fixed upon the half-speed shaft, the strap carrying a catch point at one end and a jointed rod at the other, the latter being provided with a screw thread and socket which is pivoted on a fixed support. A wing nut permits the length of rod to be varied and thus displaces the eccentric strap, and

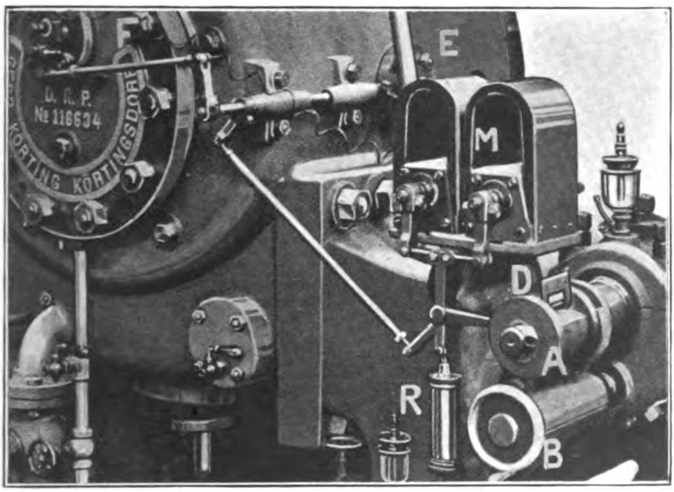


Fig. X.—21. Magneto mechanism, Koerting.

the latter, consequently, makes contact with the magneto sooner or later, as may be desired.

The method applied to some Koerting engines, shown in Fig. X.—21, ensures an absolutely uniform movement of the armature by the fact that adjustment is made on the cam shaft itself. This is carried out by means of a spiral groove along which a flange *A* moves longitudinally, carrying a tumbler *D* with a knife edge. This flange is connected to a screwed bush fitted with a hand-wheel *B*, by means of which the flange *A* can be moved longitudinally. The displacement thus obtained owing to the spiral groove, modifies the time of firing. The tumbler *D* is advanced or retarded accordingly. A coil spring in the casing *R*, by means of a connecting rod, brings the armature of the two magnetos back again to the neutral position. The two sparking plugs are fitted as shown at *E* and *F*, one in the side, and the other at the back of the combustion chamber. Each plug receives its current from one of the two magnetos.

Ignition by magneto is now almost universally adopted, and it appears to have superseded all former types of ignition. To guard against possible failures and to ensure more uniform combustion of explosive mixtures, duplicate ignition gear is advocated for engines developing more than from 80 to 100 H.P. in one cylinder. Messrs. Koerting Brothers have done so for the last eight or ten years, and other makers have also adopted the same plan. Generally, one of the

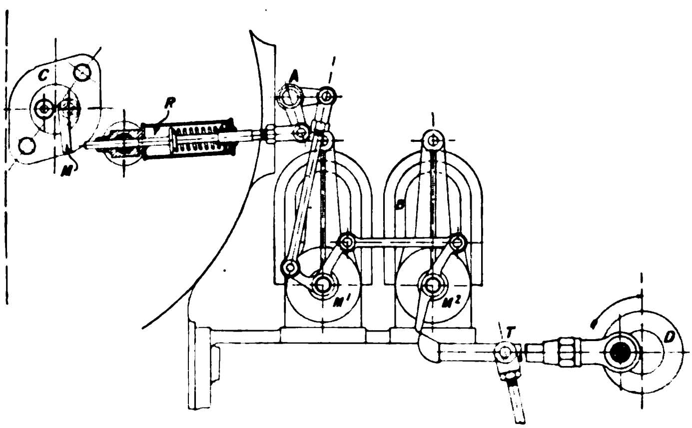


FIG. X.—22. Magneto mechanism, Koerting.

plugs is placed near the admission valve in the top of the cylinder, experience having proved, as has already been stated, that when placed near the exhaust valve at the bottom of the cylinder the igniters are unreliable.

Fig. X.—22 shows an arrangement which provides for two magnetos operated by one rod *T*, moved by a pin in the cam shaft *D*. The movement is imparted to the two magnetos M^1 and M^2 by a connecting rod *B*, and is transmitted by the spindle *A* to both sparking plugs. A single spring *R* controls the recoil. One sparking plug is placed at the end of the cylinder, as shown in the drawing, and the other at the side by prolongation of the spindle *A*.

Fig. X.—23 is a drawing of a recently designed duplex magneto ignition gear by the National Gas Engine Co., Ltd., of Ashton-under-

Lyne, two separate magnetos and independent ignition plugs being operated by the same mechanism. An eccentric pin on the end of the side shaft actuates a trigger *H* on the magneto through the connecting rod *J*, the latter being supported on a swinging guide at *K*. This guide can be raised or lowered by rotation round the eccentric pin *L*, and thus the timing of the release of the trigger *H* and, consequently, the instant of ignition can be varied through a considerable range. The trigger *M* on the second magneto is connected with the trigger *H* by the tension rod *N* passing through the tube *O*, and thus simultaneous operation of both magnetos is secured.

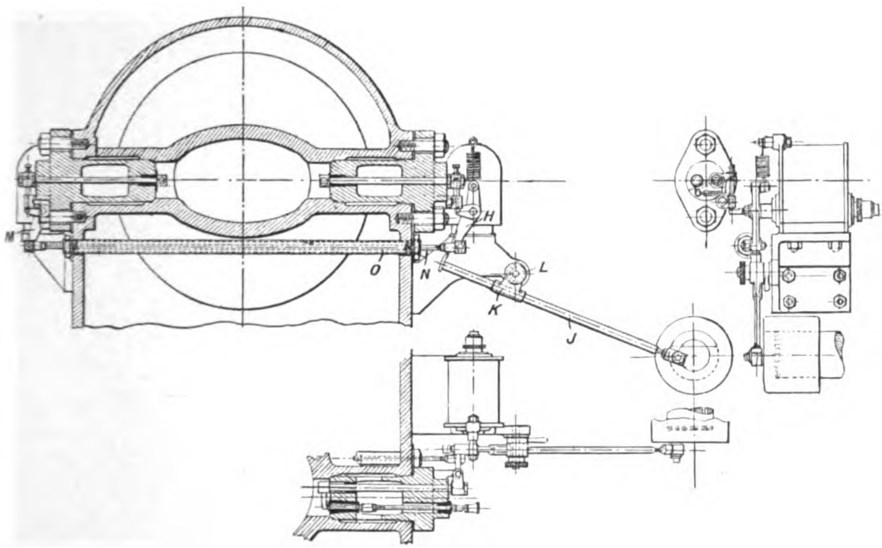


FIG. X.—23. Magneto mechanism, National.

All the working parts of ignition devices are subjected to sudden and repeated movements due to the magneto trip gear. They are consequently exposed to derangement and rapid wear. This fault can be made less pronounced by the use of pieces of light weight, presenting little inertia, and by arranging them in a manner which permits of ready access and manipulation, for inspection and proper maintenance, as in Fig. X.—19.

When the plug used has a movable hammer this constitutes a particularly delicate portion of the system. If the hammer be mechanically operated it is frequently impossible to place the plug in the most suitable position, as far as the combustion of explosive mixture is concerned. Moreover, the movable hammer is a difficult part to make properly, for the reason that some portions, much exposed to a very high temperature, have to remain in good condition with respect to insulation, and to be kept gastight and efficiently lubricated.

Tangyes Ltd., of Birmingham, have designed the fitting shown in Fig. X.—24 to maintain constant cleanliness of the contacts in respect of oil and soot. The entire arrangement is mounted on the flange of the ignition plug.

T is the rod which is gradually lifted by the ignition cam, and in so doing, causes the pawl *C* to engage in a tooth of a ratchet wheel fixed on the insulated electrode *I* of the ignition plug. When the

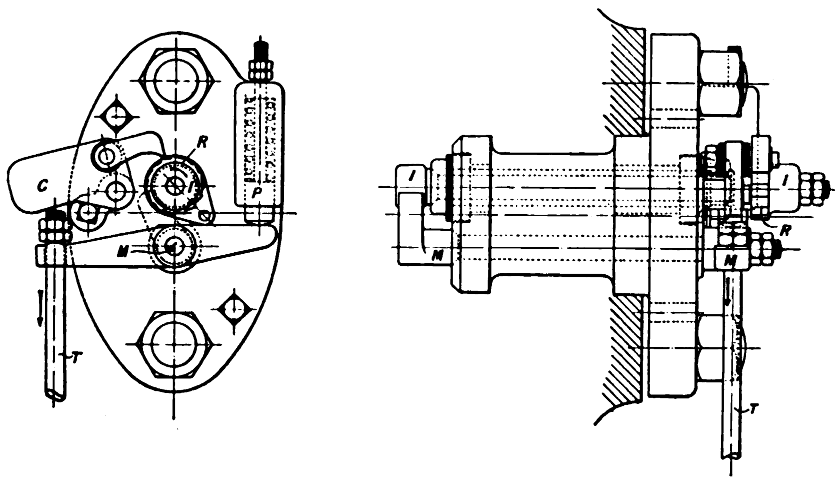


FIG. X.—24. Tangye self-cleaning Sparking points.

sudden return movement takes place, in the direction of the arrow, under the influence of the magneto spring, the separation of the contact points occurs by the movement of the pin *M*, which is afterwards brought back again to its normal position by the spring *P*. At the same time, the pawl *C* returns to its original position and engages with another tooth of the ratchet wheel in preparation for again slightly turning the insulated electrode *I*. Thus a fresh and clean contact is constantly maintained by the same mechanism that separates the contact points. Messrs. Tangyes supply a similar arrangement, hand operated, to readily permit the cleaning of the contact without necessitating stoppage of the engine.

In order to avoid troubles, which sometimes present themselves in plugs of the make-and-break type, an endeavour has been made to do away with all moving parts within the cylinder and to utilise high tension ignition, and it is now possible to ensure perfect insulation of these currents.

The ignition plug may be placed wherever desired, and the high tension spark which jumps between the points maintains these in proper order, doing away with the inconvenience of frequent removal of the plug to clean the contacts. The high tension current can be obtained from a source of electricity with a coil and condensers, which, at the desired moment, discharge a current of very high frequency in the ignition plug.

Sir Oliver Lodge, of Birmingham, the inventor of this system, attributes to it the advantage of being able to work in spite of oil, soot, or grease which may cover the plug, owing to the calorific intensity and

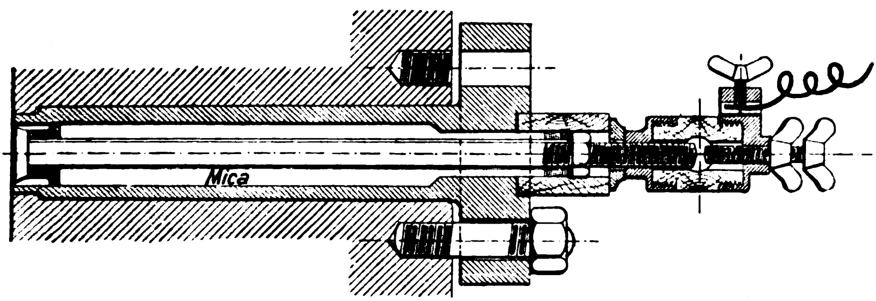


FIG. X.—25. High-tension Sparking Plug for Lodge ignition system.

the energy of the spark produced. The condenser is charged with electricity by means of an induction coil and the spark is obtained within the cylinder by means of an insulated plug.

Fig. X.—25 represents this type of ignition plug used upon some of the Oechelhäuser engines. It consists of a cylindrical external portion which is fixed on the engine, and a central rod insulated with mica. At one end this rod has a crown separated from the main body by about $\frac{1}{32}$ inch. At the other end, the rod is prolonged by a screw with an intervening space which can be adjusted so as to prevent all passage of the current across the insulator.

For a single-cylinder engine the equipment consists of an igniter, sparking plug, battery, and distributor. The igniter includes a large induction coil, a condenser, and a spark gap. The spark gap is placed in the upper portion of the coil and the spark can be easily seen through the glass cover of the apparatus (Fig. X.—26). This spark is a valuable indication, for it flashes in synchronism with that inside the cylinder. The battery includes an 8-volt accumulator, and, to avoid recharging, this can be placed in connection with the ordinary lighting mains in series with lamps. The distributor consists of two insulated brushes which bear upon a ring similarly insulated. This ring is provided with two metallic portions which place the two brushes in connection at each revolution. The brushes are mounted in such a way that by changing their position the ignition is advanced or retarded.

The igniter can be placed in any convenient position because no mechanical movements are needed to produce the spark. The external spark in the igniter is visible all the while the engine is at work.

For double-acting tandem engines, the Lodge system includes four

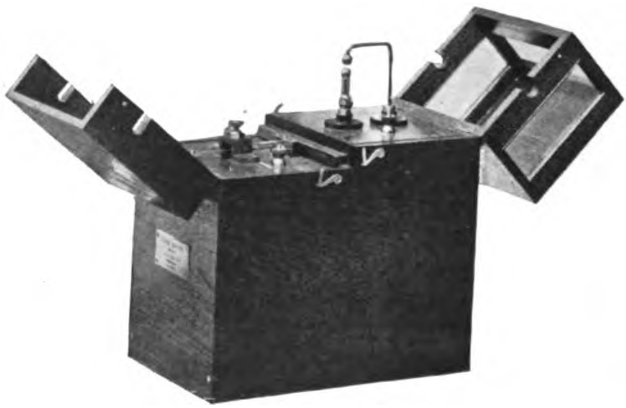


FIG. X.—26. Lodge high-tension ignition apparatus.

ignition plugs per cylinder. Each igniter operates two plugs simultaneously. The spark from each plug can be cut out by means of a two-way switch arranged on each igniter, and the ignition of any cylinder can be cut out by a switch.

The distributor consists of five brushes carried by a brush-holder which is moved in a guide concentric with the cam shaft. By displacing the brush-holder the ignition in all plugs is advanced or retarded. The centre brush is fixed, but all the others can be adjusted by a nut in order to allow the adjustment of ignition for each of the four cylinders.

One of the characteristics of this ignition system is that, when once adjusted, it does not become retarded after being at work for some time as is the case with low tension ignition, due to wear of moving parts.

Messrs. Richardson, Westgarth & Co., at Middlesborough, and Messrs. Ehrhardt & Sehmer, amongst other firms, have made numerous applications of the system.

Ignition by make-and-break continues to be in favour amongst all the Continental makers, although the Lodge high tension system has given very remarkable results wherever it has been tried. Nevertheless, ignition still remains a weak spot, and its influence upon steady operation is felt as well as on the power developed. Inventors should therefore be encouraged to do as Sir Oliver Lodge has done, and to seek improvements in a new direction.

Engines with very high compression of the Diesel, Banki, and Trinkler types, work without ignition devices, combustion being caused by the increase of temperature occasioned by compression only.

Compression.—With the increased size of gas engines, it has become necessary to deal with larger volumes of explosive mixtures but of poorer composition. At one time, some makers, in following the indications of theory, raised the compression pressures of their engines to over 200 lbs. per square inch. During the last four or five years, however, the majority of them have reduced this to a figure which better accords with the necessity to avoid premature ignitions while assuring the combustion of the very weak mixtures that must be employed for economy of fuel.

The highest compression pressure now employed varies between 140 to 170 lbs. per square inch, and there is a tendency shown to go below these figures just as much as the composition and treatment adopted for gas and air mixtures will allow.

Taught by the example of British makers who have long been averse to high compressions, the author deems it wise, in normal practical conditions, not to go beyond an average of 130 or 140 lbs. per square inch, if the engine is governed at constant compression.

It appears from a series of trials that the author has frequently made upon industrial engines in the United States and in England, that mixtures of 56 or 62 B.Th.U. per cubic foot, formed of 55 per cent. of air and of 45 per cent. of suction producer gas (of 125 to 145 B.Th.U. per cubic foot) burn perfectly with a compression of 100 to 115 lbs. per square inch, and give a mean pressure of from 70 to 80 lbs. per square inch and an explosion pressure of 340 to 360 lbs. per square inch. These mixtures, however, contained 8 to 10 per cent. of hydrogen, circumstances favourable to rapid propagation of flame. Independently of town gas, which, on account of its high price cannot be used in large engines, Mond gas (30 per cent. hydrogen), coke oven gas (50 to 55 per cent. hydrogen), and water gas (50 per cent. hydrogen) are suitable for the formation of weak mixtures. The presence of hydrogen assures rapid and complete combustion even with low compressions. For blast furnace gas only, low compression is unsuitable on account of its low content of hydrogen (2 per cent.), quite apart from its low calorific value and the large proportion of the inert gases,—carbon dioxide (CO_2), 12 per cent., and nitrogen (N) 55 per cent.

The average compression pressure already mentioned—140 lbs. per square inch—is therefore applicable—excluding blast furnace gas—to the majority of cases when producer gas is used. It lessens in a great measure the difficulties and distress occasioned by the fracture of cylinder heads due to high temperatures, and permits, at all events in single-acting types, the construction of engines of relatively high powers without having to provide water circulation for the back of the pistons and exhaust valves.

The compression pressure to adopt necessarily varies in accordance with the nature of the fuel employed. In industrial engines working with more or less heavy petroleum spirit, 60 to 70 lbs. per square inch is rarely exceeded. With the same fuel, automobile engines running at high speed can be given a compression pressure of 70 to 85 lbs. per square inch.

When using alcohol it is indispensable to carry the compression pressure up to 115 to 140 lbs. per square inch (or even 140 to 210 lbs. per square inch) with a view of overcoming the characteristic tendency of alcohol mixture to condense. This tendency should also be counteracted by very moderate cooling of the cylinder walls.

In indicator diagrams obtained from alcohol engines, those with a lower initial explosion pressure have a much greater area than those from benzine engines. This phenomenon appears to be due, partly at least, to the presence of water in the alcohol and to its transformation into steam. Without establishing an absolute comparison with the phenomena produced when water is injected in an oil (petroleum) engine, it can be said that a certain similarity exists between the two cases.

Water Injection.—From a purely theoretical point of view, it is evident that the addition of water produces no favourable effect upon the thermal efficiency of a cycle. All the theories propounded on the subject are based upon the hypothesis that fluids are working in a vessel entirely impervious to heat.

Given a volume of Gas Vo, at a temperature To, and at the pressure Po, if it be compressed to a volume V, the temperature will become T

and the pressure P. The curve will be adiabatic. After expansion to the original volume, the mass returns to the initial temperature and pressure.

Supposing that, before compression, within the enclosing vessel, a very small quantity of water is added as compared with the volume Vo of gas. This water will become heated to the temperature To. During compression it will absorb a relatively large quantity of latent heat of vaporisation. At the end of compression the mixture of gas and water vapour will occupy the volume V, but at a temperature T^1, and a pressure P^1, lower than T and P. The compression curve will more nearly approach an isothermal curve. After expansion, the initial conditions will be restored, the vapour becoming condensed.

In practice, the changes take place in a different way, the enclosing vessel not being impervious to heat. Take the two examples given above. In the first case, the gas will give out heat to the walls during compression; this heat will be in part lost to the circulating water; it cannot be entirely restored after expansion, so that the curve will not coincide with that of compression.

In the second case, the gas will give the greater part of the heat of compression to the water that it contains, its temperature will be less elevated, the losses to the walls will be diminished, and, during expansion, the vapour formed will restore by condensation the heat that it had absorbed. The expansion curve will be below the compression curve, but the difference of the ordinate between the two will be less than in the first case. The variations occur as if the permeability of the walls to heat had been diminished. It is therefore probable that the total loss will be less in the second case than in the first.

This reasoning applies also to power cycles, for in this case a considerably greater amount of water could be vaporised in the abstraction of heat produced by the combustion of gas. But, upon this basis, the walls give out heat to the gas during compression instead of becoming hotter, and, at the end of expansion, the vapour is no longer condensed, so that the heat of vaporisation is lost to the exhaust.

It remains to be seen whether this loss diminishes the inherent losses occurring in normal working, or, in other words, whether the efficiency of the engine is increased by water injection simply, compression being unchanged.

An engine with water injection that would maintain normal working conditions with regard to temperature and lubrication, and with re-heated walls, as for example by aid of the exhaust gases, would probably give an excellent efficiency.

At the present time there is a lack of sufficient data, based upon actual experiment with explosive mixture in the presence of water, to enable any precise rules to be formulated. However, from tests which have been carried out it is possible to deduce some interesting points which are of value in indicating new fields of research, and at the outset, examination can be made of the conditions under which water can be usefully employed in an explosive mixture containing a combustible such as rich or poor gas, mineral spirits, or heavy petroleum.

It appears, *à priori*, that water should enter the cylinder as a liquid and not as a vapour, so that the transformation, from the first state to the second, may have the effect of absorbing a comparatively large amount of heat, and so produce a pressure that will give out power,

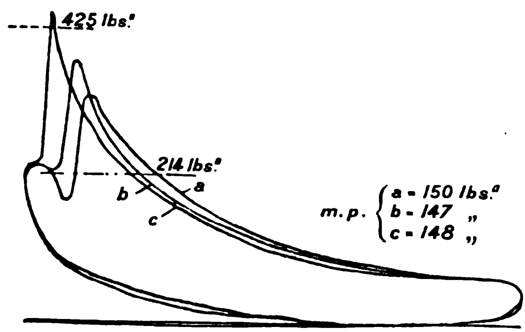


FIG. X.—27. Indicator diagram from Banki's Benzine Engine with water injection.

the power being the greater, the higher the temperature of the enclosing walls, and, in consequence, the water and vapour being able to borrow a greater number of heat units. Such conditions can be obtained either by means of increased compression pressure or by restricting the cooling of the combustion chamber.

If these conditions be not fulfilled, the introduction of water may cause the igniter points to be covered with small beads of water which would prevent the passage of the spark. On the contrary, if higher temperatures are established, it will be necessary to have recourse to water injection to prevent the various details becoming over-heated, and the occurrence of premature ignition.

Amongst the tests that have been carried out in connection with water injection, one of the most characteristic was that made by the author in 1899 on a 35 H.P. Banki engine at the works of *Ganz and Tarza* at Budapest.

The initial explosion pressure was about 400 to 430 lbs. per square inch. The casing maintained a uniform temperature due to the water injected with the benzine. The amount of water injected was from three to four times the weight of the benzine consumed. The mean pressure of the diagrams amounted to 146 to 150 lbs. per square inch, and their area was not sensibly affected by the amount of water injected. The latter simply retarded the combustion and diminished the explosion pressure as shown by the diagrams taken during the test and reproduced in Fig. X.—27.

In connection with this subject it is interesting to mention the tests made by *Crossley Brothers, Ltd.*, of Manchester, with their petroleum engines. The type adopted, shown in Fig. X.—28, consists of a cast-iron cylinder at the back of which a long cast-iron vaporising chamber

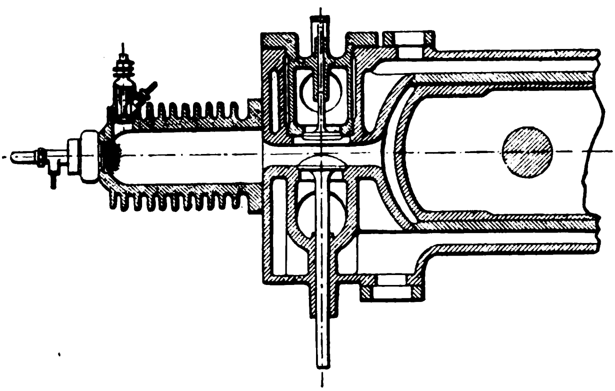


FIG. X.—28. Crossley Oil Engine Vaporiser with water injection device.

is attached, having external ribs to give stiffness and ensure cooling. This vaporising chamber is in communication with the cylinder by a narrow passage in which the inlet and exhaust valves are accommodated.

The petroleum is injected at the end of the vaporiser at the same time as a jet of water and a small volume of supplementary air. The resulting mixture is then compressed and fired by means of an ignition tube situated in the middle of the vaporiser. The makers claim that, owing to the water injection they are able to employ a higher compression pressure without fear of premature ignitions, and also that they are able to prevent the decomposition of the fuel which would have caused a deposit of carbon.

Messrs. Crossley have also applied water injection to gas engines, and have been able to reach compression pressures of 200 lbs. per square inch without causing pre-ignitions. Fig. X.—29 shows the method employed, and it will be seen that the mixture flowing past the inlet valve meets the water and sweeps it along, preventing it from being deposited on the cylinder walls.

Tangyes Ltd., of Birmingham, adopt water injection for their petroleum engines, and obtain increase of power by so doing. Fig. X.—30 shows the principle of admission that they employ.

It is therefore indisputable that water injection provides means

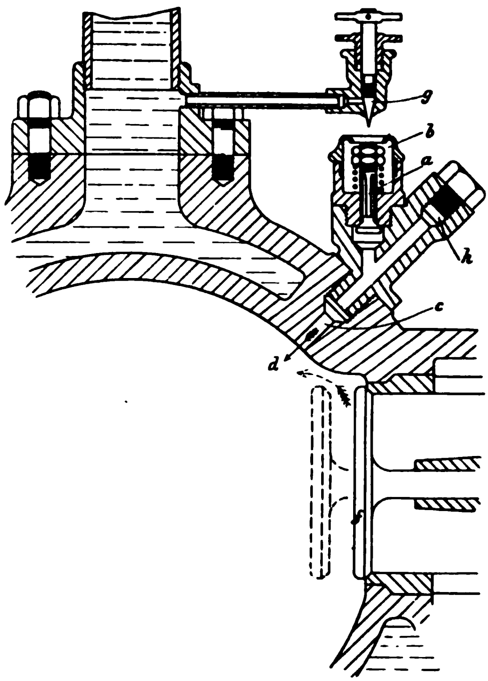


FIG. X.—29. Crossley Gas Engine cylinder with water injection device.

whereby higher compressions may be reached, resulting in reduced fuel consumption.

In some heavy oil engines, such as that made by the *Griffin Engineering Co.*, of Bath, for instance, a certain amount of water is injected into the vaporiser with the fuel, in the form of very fine spray, which is immediately converted into vapour combining with the fuel within the cylinder when the compressed mixture is ignited. The economical results obtained with this system of vaporisation is most marked when crude oil is used as fuel. The moderate temperature of the vaporiser (never more than 400° F.) produces fractional distillation of the heavy oils, the residuals are automatically rejected from the vaporiser whilst the lighter portions, set free by the distillation, pass into the cylinder, where they are completely burnt without the slightest deposit of tar.

The *Britannia Engineering Co.*, of Colchester, also use water injection in their crude oil engines. Their vaporiser *V* (Fig. X.—31) is a cast-iron box bolted to the breech end of the cylinder, and is fitted with deflecting plates. The oil and water are injected under pressure. The igniter *A* is a small piece of cast-iron enclosing a corrugated iron tube which is in communication with the combustion chamber at two points. This igniter, whilst the engine is at work,

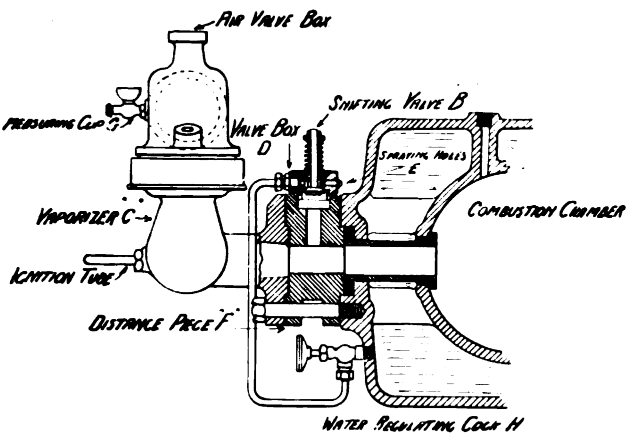


FIG. X.—30. Tangye Oil Engine with water injection device.

absorbs sufficient heat from each explosion to maintain a sufficient temperature to fire the rich vapours that are contained in it at the end of the compression stroke. The pumps, shown in the lower drawing of Fig. X.—31, are contained in one casting. The plungers are brought to their normal position by means of springs, and are depressed by bell-crank levers jointed to that which operates the gas or vapour valve. In this manner, neither oil nor water are injected, when the governor puts the gas or vapour valve out of gear. The stroke of the pump can be regulated by an adjusting nut placed at the top of each plunger, the quantity of oil and water admitted per cycle being thus controlled. The pumps are fitted with ball valves. With petroleum having a flash point of 250° F. it is necessary to renew the igniters after twelve hours' work, an operation requiring only a few minutes.

Cooling.

The use of water should be discussed from different points of view :—

1. Cooling apparatus.
2. Control and maintenance.
3. Effect of the circulation upon the working parts of gas engines.

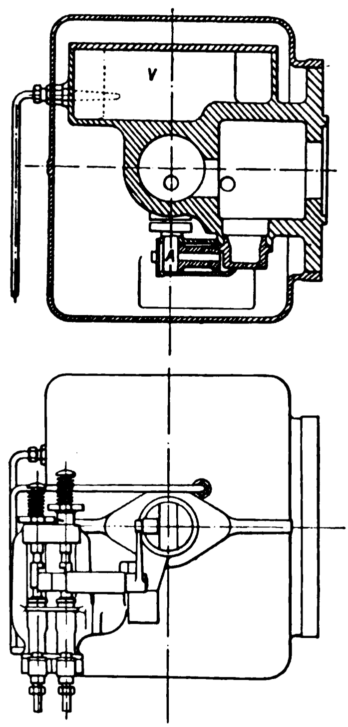


Fig. X.—31. Britannia Crude Oil Engine with water injection device.

1. Cooling Apparatus.—For a producer gas engine, single-acting, and of less than 200 h.p., the normal consumption of water per b.h.p. at full load is :—

5·5 gallons for cooling the engine, if allowed to run to waste.
3·3 „ „ washing the gas in the scrubber.
0·1 „ „ feeding the vaporiser.

making a total of 8·9 gallons or, in round numbers, 9 gallons per b.h.p. hour.

For a 50 h.p. engine, for instance, it will be necessary to arrange for 450 gallons per hour.

If, instead of running the cooling water to waste, some system of cooling is employed—either an arrangement of tanks on the thermo-syphon principle or by cooling towers—the consumption of water for the engine becomes negligible. It is sufficient to make up evaporation losses, and these scarcely amount to more than 1 per cent.

The scrubber water should be considered as useless in nearly all cases. However, if it be obligatory to economise as much as possible by adopting special treatment, it can be utilised over and over again.

Tangyes Ltd., of Birmingham, have brought out the purifying process illustrated in Fig. X.—32 for this purpose. The water, after leaving the scrubber (*S*), flows through the pipe (1) to a closed tank (*T*) where it becomes filtered by the settlement of the solid matters contained in it. The filtered water is then drawn through the pipe (2) by a small pump which forces it by the pipe (3) through the water-jacket of the engine (*C*), from which it is afterwards led by the pipe (4) to the combined exhaust silencer and deodoriser (*E*).

Owing to the heat of the exhaust box the water is purified from the gases with which it was saturated, the gases passing away to the atmosphere with the products of combustion from the engine itself.

The purified water is next conducted by the pipe (5) to the open

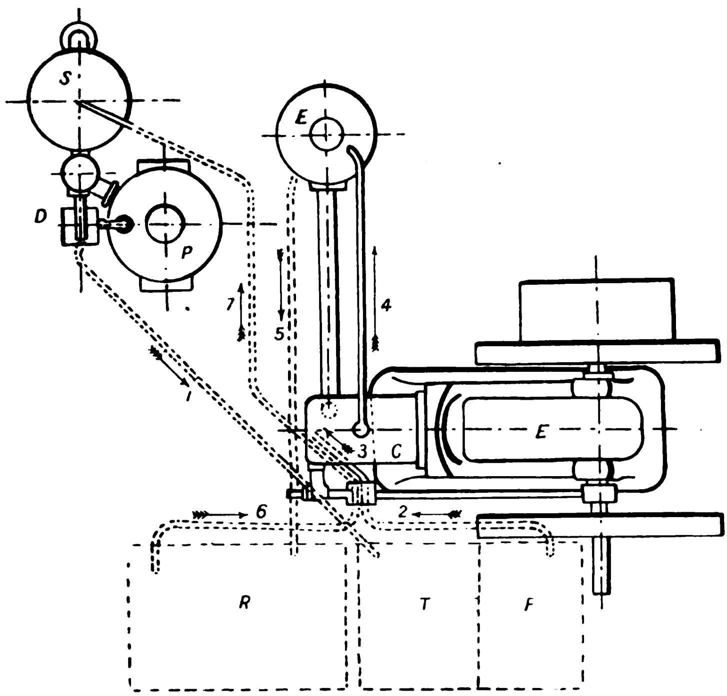


FIG. X.—32. Tangye Water Purifying Apparatus.

tank (*R*) from which it is pumped through the pipes (6) and (7) to the scrubber, in which it again meets and cools the gas and commences once more the "cycle" of purification. The water used in the vaporiser is necessarily irrecoverable, because it is transformed into vapour whilst passing through the mass of incandescent fuel, and combined with the latter to form hydrogen and carbon monoxide.

For the sake of comparison, it can be accepted that an improved non-condensing steam engine requires about 25 lbs. of steam, or 2·5 gallons of water per H.P. hour. For a 50 H.P. engine, 125 gallons per hour would be indispensable.

For a condensing engine, the consumption of steam required for working would be reduced to about 17½ lbs. of steam or, say, 1·75 gallons of water, and the condenser would absorb nearly 44 gallons per H.P. per hour, making a total of, say, 45 gallons per H.P. or 2,250 gallons per hour.

It is of course true that, as in a gas engine, a cooling apparatus can be utilised. The preceding figures enable the necessary arrangements to adopt in each case with regard to the consumption of water to be appreciated, but other considerations must be taken into account.

The cooling of gas engines with water running to waste is advisable only for small powers, and when the water available contains but a

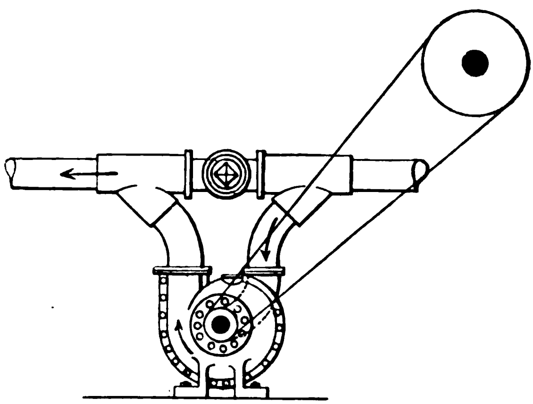


FIG. X.—33. Water circulation by Centrifugal Pump.

small amount of lime salts in solution. The latter, when rapidly becoming deposited in the jacket of the engine, blocks the proper circulation of water, and prevents the transmission of heat through the cylinder walls, owing to the formation of scale, and occasionally results in accidents and fracture of parts.

The quantity of water required is about 4·5 gallons per H.P. hour for "hit-and-miss" governed engines, and 5·5 gallons for those regulated by variable admission.

Cooling by tanks upon the thermo-syphon principle is not advisable for engines above 50 H.P., and when, owing to local considerations, it is necessary to place the tanks at some distance away from the engine, it is a good plan to insert a pump in the circuit to quicken the circulation, by means of a by-pass connection and a stop-valve on the main branch, as illustrated in Fig. X.—33.

To be really effective the cooling tanks should be placed in the open air, care being exercised during frosty weather to prevent the formation of ice in the pipes and valves. The contents of one tank is generally from 330 to 440 gallons; the number of tanks is decided by the total quantity of water demanded on the basis of 45 to 55 gallons per H.P.

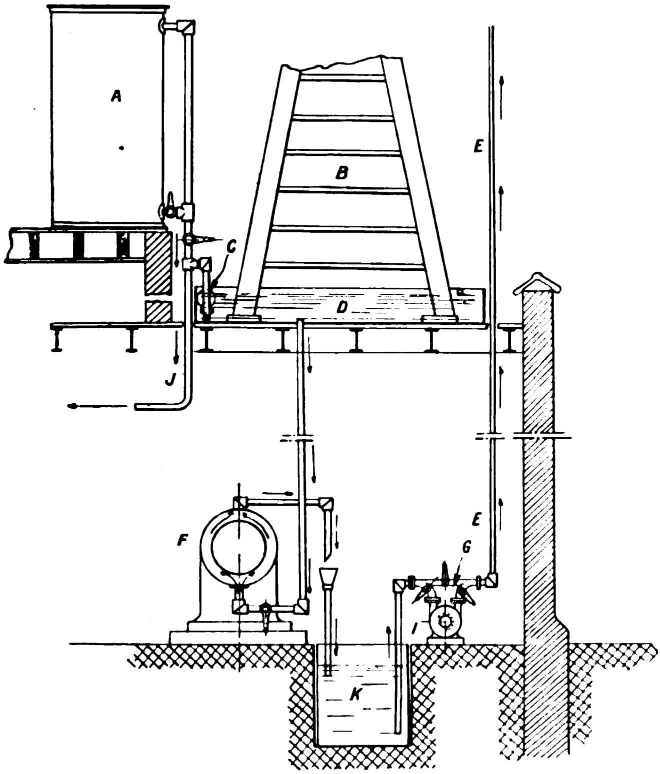


FIG. X.—34. Water circulation with cooling apparatus (visible outlet).

for "bit-and-miss," and 55 to 65 gallons for engines governed upon the variable admission system.

The most rational arrangement for 50 H.P. engines and over is to use coolers, and of these the coolers made with branches or twigs of birch wood, erected in the vicinity of the engine house, are least expensive. Two arrangements may be adopted, either by circulation with visible outlet, which permits the instant control of the temperature of the water, or by entirely closed circulation.

In the first arrangement, Fig. X.—34, the cooler (*B*) should be placed overhead and the water made to flow, under the influence of the difference between the levels of the tank (*D*) and the engine (*F*), round the cylinder jacket, and from this to flow freely into a funnel at the orifice of a pipe leading to an underground cistern (*K*), afterwards being lifted by a pump (*I*) and delivered to the cooler overhead. An objection to this method is that the pump is required to deal with hot water at 85° to 120° F., and therefore the pump must be erected in

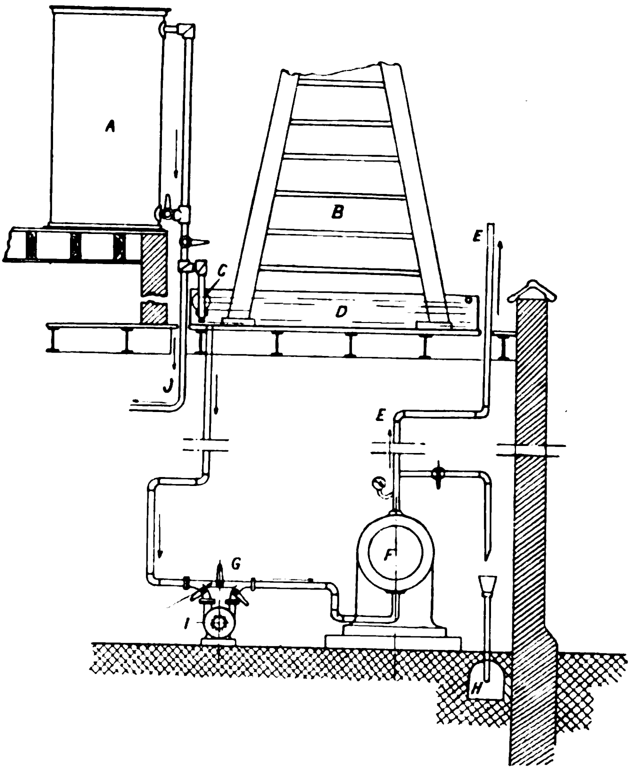


FIG. X. 35. Water circulation (closed system) with cooling apparatus.

such a manner that the temperature of the water will not prevent the water being raised. A reservoir (*A*) should be provided holding a sufficient quantity of water, which, with the amount in circulation, will permit the engine to continue to work for one or two hours with the water running to waste, in case the pump should fail. Two supply pipes should therefore be taken to the reservoir, the lower one being fitted with a cock which can be opened when it is necessary to work with the water going to waste.

With a closed system of circulation, the pump (I) is fed with cold water flowing from the cooler (D) and keeps the latter full by passing the water through the engine (F) as in Fig. X.—35. With this arrangement it is impossible to notice whether the circulation is working properly unless the flow of water from the top of the cooler can be observed. However, a gauge can be fitted on the outlet pipe (E) of the engine so as to show the pressure under which the water is being circulated. In order to obtain continuity of working in case of a breakdown of the pump, provision should be made to run the water to waste at (H), while it is also advisable to provide three stop-valves at (G) so as to isolate the pump (I) when under repair.

The arrangements mentioned above apply only to engines of small power. For larger installations birch twig coolers are scarcely suitable. There is always the possibility that obstructions caused by pieces of wood, shavings, and other foreign matter, will cause trouble, while the twigs themselves deteriorate comparatively rapidly. These risks are naturally greater in large installations. Preference should therefore be given to less cumbrous apparatus, which are not so liable to get out of order. Several firms make a speciality of such coolers, and amongst them are the following:—

Balcke & Co., Paris.
Blasburg & Co., Düsseldorf.
Klein, Schanzlin & Becker, Frankenthal.
Jarvis Brothers, Ltd., Middlesborough.
Munzinger Albert, Kaiserlautern.
Schwartz & Co., Dortmund.

These "open" coolers, or those provided with a chimney, with either automatic or mechanical ventilation, are designed so as to obtain as thorough cooling as possible, and to avoid the formation of vapour or water spray. They are a practically indispensable adjunct to large gas power installations when it is requisite to economise the water at disposal.

Cooling devices having for their objects (1) economy of water and (2) the prevention of scale within the cylinder jackets of gas engines, it becomes necessary to avoid using a common reservoir for the cooling of the engine and for the washing of the gas in the scrubber, for in the latter case fresh water would be constantly mixing with the old, and, consequently, the deposit of scale would be more abundant than if the same water were used over and over again for the engine.

It is a good plan to fix a regulating tank having a capacity of 100 or 200 gallons, at a height of 15 or 20 feet above the upper top of the scrubber, so that the water will flow from the sprinkler or distributing device within the scrubber in the form of rain. Insufficient pressure permits the water to escape in a solid jet on to the central portion of the coke bed only, with the result that the gas is badly washed.

This regulating tank should be placed above the level of the reservoir of the engine cooling system, in such a manner that, by means of a float, the quantity necessary to compensate for evaporation is fed automatically, and a constant level of water maintained in the cooler.

The water used for washing the gas and passing from the scrubber contains salts of ammonia and sulphuretted hydrogen in solution, which gives it an unpleasant smell. If the fuel used contains an appreciable amount of sulphur, the water will also contain sulphurous acid in solution, which, in the presence of air, will become oxidised and produce sulphuric acid. It is, therefore, necessary to avoid the use of zinc pipes or vessels which would soon be eaten away. Iron also rapidly oxidises if the sulphuric acid be present to an abnormal extent. To confine the smell within close quarters to the plant, the overflow from the seal box should be fitted with a syphon bend, and the top of the water should be covered with a film of oil which will prevent the smell from impregnating the atmosphere.

In installations outside towns where the overflow cannot be led into the sewers, the water must be exposed, with the result that the smell arising from it may give annoyance to neighbours. This possibility should not be lost sight of when contemplating the erection of a gas power installation.

The water for feeding the vaporiser is generally taken from the same supply pipe that feeds the scrubber. Care must be taken to arrange that the supply to the vaporiser shall not be influenced by any adjustment of the supply to the scrubber or vice versâ. The quantity required varies between 0·066 and 0·1 gallons per B.H.P. hour. But it must be remembered that this figure is but an approximation, variations being due to the peculiar features of the generators and vaporisers adopted.

As a result of about forty trials made with ten or twelve different types of generators, the average water consumption is at the rate of 0·7 to 0·9 lb. per lb. of coal burnt per hour.

In a number of producer plants the overflow from the vaporiser is diverted to the ash-pit of the generator, whilst others work with the supply of water strictly limited to that evaporated in the vaporiser. In the author's opinion the first method is preferable having regard to the preservation of the firebars, but it is difficult to adopt the plan with a rational system of feeding the fire with air and superheated vapour as in the Winterthur producer. The ash contained in certain kinds of coal becomes pasty, and causes very hard clinker to form when burnt without a sufficient amount of water vapour being present.

2. Control and Maintenance.—Upon the assumption that a properly equipped installation has been provided comprising everything that experience has shown to be necessary, how should the quantity of water be regulated to obtain best working results?

Engine.—Is there any advantage to be gained in using such a quantity of water for the engine that the temperature at the outlet of the jacket shall reach 130° to 140° F; or would it be better to permit sufficient water to pass through the jacket to bring the outlet temperature down to 75° or 85° F.?

Variations produced in consequence of water circulation have an influence in small engines. Mention is made elsewhere of the results of a B.H.P. test of a 10 H.P. English engine. With abundant water circulation the power developed by the engine was 10·65 H.P. With moderate cooling, other conditions remaining unchanged, it developed 11·25 H.P. (p. 472).

With large engines it appears that the more or less abundant supply of water has no influence upon the efficiency. "The amount of heat passing through the walls and the exhaust is practically constant." This fact has been demonstrated by different experimenters. For large engines, especially with high compressions, it is dangerous to work without an abundant water circulation, for otherwise, almost without exception, premature firing results.

Washers or Scrubbers.—The supply of water not being in sight there is no other means of taking note of its sufficiency or otherwise than by ascertaining, after several hours of work, if the temperature at the base of the scrubber has become higher. The external temperature of the lower ring of plates should not exceed 85° to 95° F., and the upper portion should remain quite cold. If otherwise, it is a sign that an insufficient amount of water is flowing through, and this must be attended to, or that chimneys or "cores" exist throughout the depth of coke which allow the gas to pass through too freely at such points.

Vaporiser.—It is impossible to give any general rule respecting the quantity of water to be admitted to the vaporiser, as so much depends upon the type of apparatus. Some producers are fitted with an automatic device which controls the supply in accordance with the work on the engine. There is, therefore, no other course to follow in such cases than to cause the water to flow into the apparatus in such a manner that it cannot escape unused from the overflow pipe.

When the overflow from the vaporiser is carried to the ash-pit, the inlet to the latter is fitted with a funnel and syphon, so that the amount dropping into the latter can be seen. In this case the supply to the vaporiser must be adjusted in such a way that the ash-pit is fed very slowly, drop by drop.

With a dish vaporiser placed on the upper part of the generator itself, attendants generally completely fill this with water before the plant is set to work. In such cases about two hours elapse before the apparatus attains sufficient heat to warm the mass of water in the vaporiser.

Very little water should be fed to such vaporisers at first, but they should be filled up gradually to the normal level as the generator gets hotter. For this purpose all vaporisers of this type should be fitted with water gauges. Faulty manipulation in respect of supplying too much water at starting rarely has any other convenience than the gas is slow in reaching its normal richness. It is a much more serious matter if the vaporiser be carelessly permitted to become dry and then to be suddenly filled as soon as this is found out. Such treatment would cause fracture of the vaporising dish, which is usually made of cast-iron.

When vaporisers with but a small volume of water are placed near the furnace itself, a high working temperature is rapidly attained and they should be filled with water soon after the plant is put in operation.

3. Effect of Water Circulation upon Working Parts.—Cooling should be efficient and absolutely assured, because the tensile stresses in cast-iron, set up by unequal expansion due to increase of temperature, become serious in the heavy pieces of metal used in the construction of gas engines. The breech ends, or explosion chambers, have often without apparent reason become fractured in most unexpected places. It is only by long experience, resulting from a number of failures, that makers have been able to decide how to make alterations to their designs so that the breech end castings may be better able to resist the strains set up. The use of other materials for their construction, such as steel, has been tried without success.

The principal factors operating for the preservation of the breech ends are their shape and the manner in which the cooling water circulates within them. For single-acting, four-cycle engines the type that appears to be most in favour at present is that in which the inlet and exhaust valves are arranged upon the same vertical axis, the two valves being placed in a passage or ante-chamber which envelopes all the parts with cooling water. To ensure equal expansion throughout, the chamber is arranged symmetrically in relation to the cylinder axis. The exhaust outlet is arranged upon an extension of the chamber and is surrounded with water as completely as possible.

An abundant circulation is thus provided round the seat and guide of the exhaust valve, and for engines of above 100 H.P. water is circulated within the interior of the valve itself. In Chapter XII. some further remarks appear in connection with the form of breech ends.

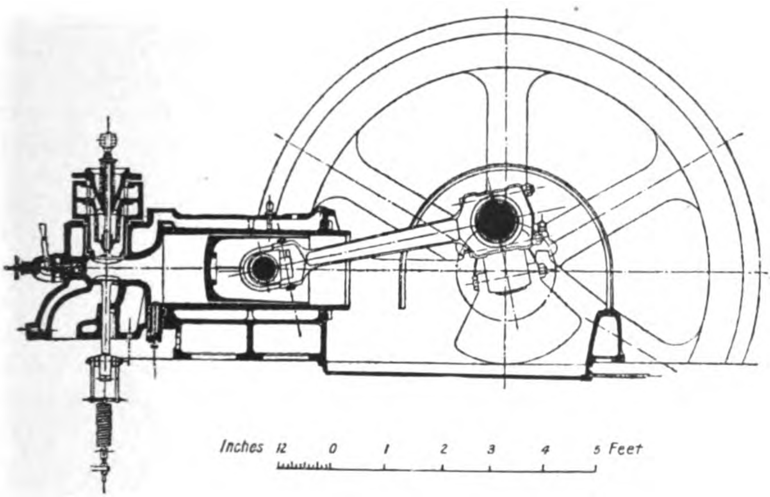


FIG. X.—36. Water circulation for small engine cylinders.

The water enters at the base of the combustion chamber and escapes from the top, or, sometimes, in motors of less than 75 H.P., it passes from the breech end into the cylinder jacket proper (Fig. X.—36).

It will be seen from the section that the jacket is cast with the base, the cylinder being an independent liner and free to expand, whilst the combustion chamber is connected to the cylinder by a flange and bolts. This arrangement ensures an extensive bearing surface for the engine, and does away with the objectionable overhanging cylinders, now completely abandoned in good modern construction.

To obtain high efficiency, the cylinder, at that part within which the expansion of gas occurs, should not be excessively cooled. In fact, expansion itself causes a fall of temperature, and to lessen it, it is necessary to keep the walls as warm as possible (as in the case of engines using saturated steam, by using a cylinder jacketted with live steam). For this reason, the water outlet from the cylinder jacket should be arranged towards the back and not towards the front as some makers wrongly cast it.

To overcome the excessive increase of temperature involved by high compression, a number of makers, amongst whom may be mentioned the *Machinenfabrik Augsburg Nürnberg*, have even provided a separate water circulation to the pistons of their single-acting engines ranging from 100 H.P. (Fig. X.—36). This is a complication that has been easily avoided by other makers up to even 150 H.P., by arranging the piston in such a manner as to obtain sufficient cooling, simply by free contact with the air during its backward and forward movements.

For gaining all the advantages of high compression without running the risk of "self-ignition," *Koerting Brothers*, in the interior of the combustion chamber of some of their engines, have arranged a hollow projection within which separate water circulation is made. This cooling taking place in the centre of the mixture damps down the excessive temperature, and has, it appears, given the best economical results with a compression pressure raised above the ordinary limits.

In double-acting engines, the cooling has to be carried out with great care, for, apart from the cylinder jacket and the cylinder covers, independent water circulation is required for the piston and its rods, the valve seats, exhaust valves, and stuffing-boxes. The general temperature is kept relatively low, and, to this end, the water in certain portions, such as the piston and its rods, is maintained at a pressure of at least 15 lbs. by means of a special pump.

Quantity of Water in Circulation.—From a number of tests made by the author upon double-acting, four-cycle engines, it seems that the quantity of circulating water required per B.H.P. per hour, for engines of 200 to 1,000 H.P., is as follows:—

Cylinders, covers, and stuffing-boxes .	5·5 to 6·5 gallons.
Pistons and their rods	2·0 to 2·5 ,,
Boxes, seats, and exhaust valves . .	1·0 to 1·75 ,,

Say, from 9 to 11 gallons for all purposes.

The water admitted at a mean temperature of 55 to 60° F. flows from the cylinder jackets at 75 to 95° F.; from pistons at 95 to 105° F.; and from the seats and valve boxes at 105 to 115° F.

An engine of 1,000 H.P., two-cylinder, double-acting, will require, according to the figures given in the above table, about 1,600 cubic

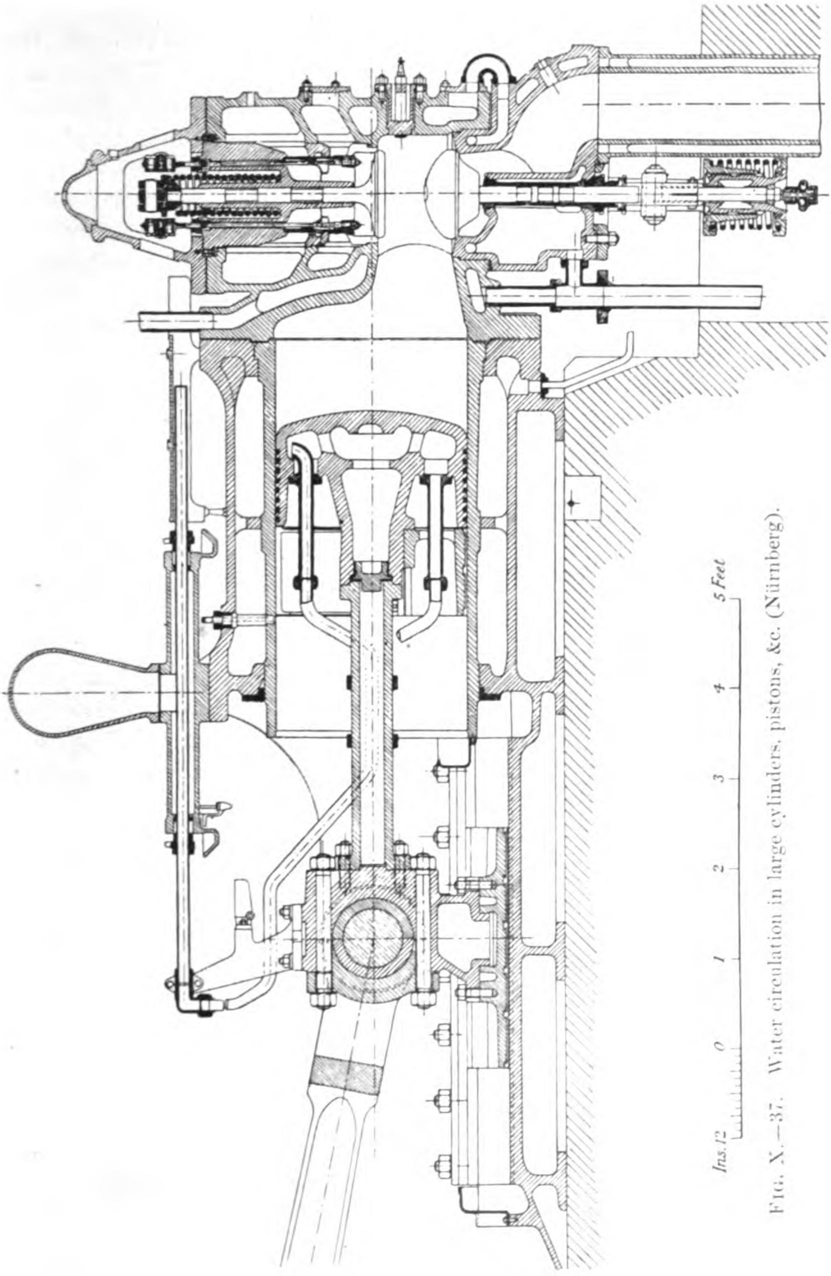


FIG. X.—37. Water circulation in large cylinders, pistons, &c. (Nürnberg).

feet (250 gallons) of cooling water per hour. As this is a large amount that cannot be arranged for continuously in every works, coolers are usually installed, which, as has been already mentioned, present the appreciable advantage of eliminating the causes of incrustation by using the same water constantly. Without presenting the same dangers to gas engines as in steam boilers, deposits of scale are a serious nuisance, by causing fracture of the parts within which they form. In casings which cannot be dismantled for cleaning, it is necessary to arrange large hand holes, closed with bolted cover plates, to permit free access to the interior for cleaning purposes.

Lubrication.—The great importance of lubrication and the necessity of using specially prepared oil for gas engine cylinders, are matters upon which all authors are in accord with manufacturers. In view of the conditions to be fulfilled, it is advisable to purchase oil from those oil merchants who have given close study to the question.

The following remarks have been abstracted from a circular published by the *Henry Wells Oil Co.*, of Manchester:—

The high cost for lubrication of gas engines that is mentioned as a weak point is due partly to the excess of oil consumed, and partly to the poor quality of oil used.

A number of makers of small gas engines have looked upon the quality of oil as a matter of indifference, thinking it possible to compensate lack of quality by a larger quantity. Other makers have become oil merchants and have insisted that their customers shall purchase their supplies direct from them. This has sometimes led to the sale of cheap oils to the engine builders, and the resale at a high price to the engine user.

The use of large engines has modified this state of things, and at the present time there is a marked tendency, as much on the part of users as well as of makers, to follow the advice of specialists who have studied the compositions of oils, their qualities and manner of use, from a scientific point of view.

At first sight it would appear that the flash point of oil is an important factor in the choice of a suitable lubricant for gas engine cylinders. As a matter of fact, the flash point is practically negligible. It is recognised that the most suitable oil for the purpose has a degree of viscosity similar to that of bearing oil, and such oils have a low flash point when compared with the cylinder oils used for steam engines.

In spite of the very high temperatures which obtain in a gas engine cylinder, the low flash oils used lubricate perfectly, due to the extreme care taken in the selection of the constituent elements. These oils may become vaporised under the high cylinder temperatures, but in doing so they should not leave a residue which, being burnt, would form a deposit of carbon.

During work, a very thin film of oil covers the cylinder walls; the temperature conditions of these walls and the rapid renewal of the oil film suffices to prevent combustion except in the case of oil of inferior quality.

The regular maintenance of this oil film is one of the essential elements of good lubrication, and the best oil is that which will remain in thin particles for the longest time and require to be less frequently renewed. Bad oils always necessitate a rapid feed, because

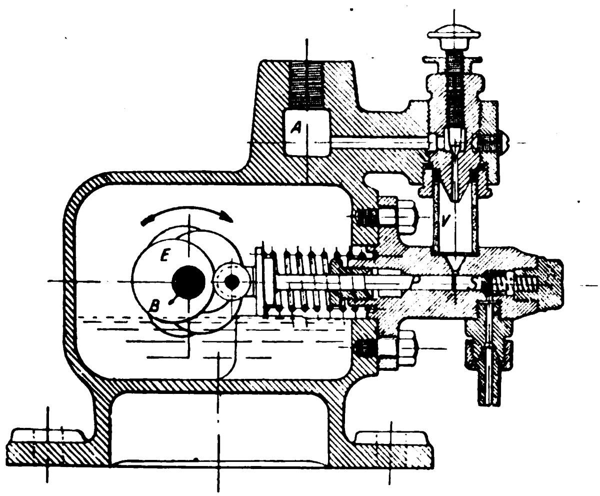


FIG. X.—38. Paxman Multiple-outlet Automatic Lubricator.

of the film being constantly evaporated, and any excess forms a deposit in the lower portion of the cylinder, when, if the oil is not immediately consumed, it combines with the tar, soot, and dust brought in by the gas and air, and gradually becomes dry and hard.

The question of cylinder oil is bound up with that of the oil used for the bearings and other working parts of the engine. At one time two different oils were used, but the almost universal practice now is to use but one. When two different oils are used it is impossible to use the bearing oil for the cylinder in case of need.

Cases have occurred when the bearing oil had been thrown upon the piston rods and gradually entered the stuffing-boxes and ends of cylinders. There it has been burnt, ruining the packing in the stuffing-boxes, spoiling the piston rods, and scoring the cylinders.

Oils should be practically free from acids, but it is not necessarily good for them to be exclusively hydro-carbon oils. The best gas engine oils are combinations of suitable elements. Different oils may be good, though the factors relative to viscosity, density, boiling point, and flash point may be very different. On the other hand, a very bad lubricating oil could imitate a good one in nearly all these factors.

One important point to notice in the selection of a suitable oil is that it does not form an emulsion with water. If water in any way whatever enters the cylinder, as sometimes happens, an oil which will

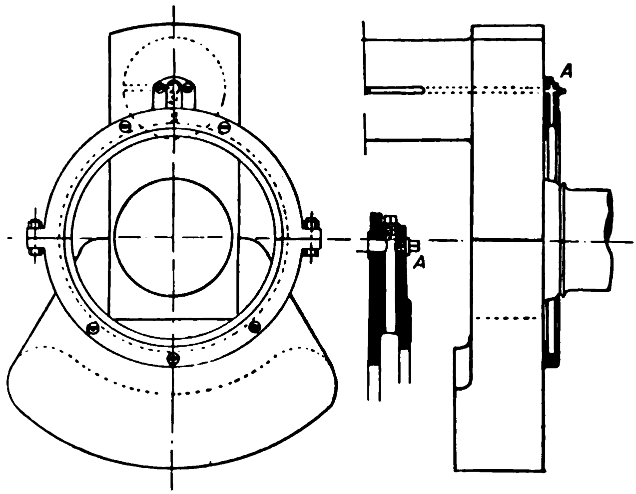


FIG. X.—39. Crank-pin Lubricator.

form an emulsion will form a compound which lends itself to rapid drying and burning.

Finally, cheap oils are neither economical nor efficient. A good oil costs more per gallon, but it gives greater efficiency and conduces to economy.

Lubricators.—There are a very great number of types of lubricators. The ratchet-wheel applied to lubricators is not to be recommended, for the amount delivered can only be varied within very wide limits. For example, if the pawl should take three teeth and a reduction of the quantity be desired, it could, of necessity, only be arranged to take two teeth instead. This corresponds to a reduction of 33 per cent., which may be too much. Similarly, on passing from two teeth to one tooth, the amount delivered will be reduced by 50 per cent.

Fig. X.—38 represents the section of a central automatic lubricator with multiple outlets, as made by W. Paxman, of Colchester.

The shaft *B* is mechanically operated by the engine. It carries a series of eccentrics *E*, actuating small pistons *P*, which force the oil through retaining valves *S*, with spring control, to the tubes. The oil escapes freely from a reservoir *A* by sight-tube *V*, which permits the amount delivered to be controlled, the adjustment being made by means of a needle-valve.

Some of the general rules which it is useful to observe in connection with the mechanical lubricators are as follows:—

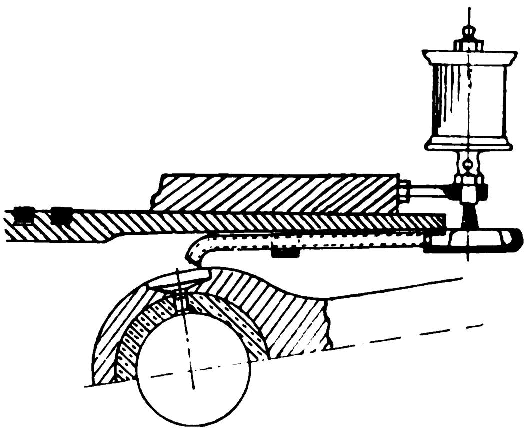


FIG. X.—40. Piston-pin Lubricator.

All rubbing surfaces should be automatically and continuously oiled with sight-feed apparatus.

The lubrication of the crank-pin or connecting-rod head should be ensured by a centrifugal ring, fixed to one of the crank webs, arrangements being made to permit the oil passages being cleaned, as shown at *A* in Fig. X.—39.

The piston pin should be lubricated by an independent apparatus with an adjustable and visible feed, as in Fig. X.—40.

The lubrication of the piston should be ensured by means of a pump, positively operated by means of a cam or eccentric, and not by a belt liable to slip. This pump should introduce the oil under pressure and be arranged in such a manner that the amount fed to the piston may be adjusted while the engine is working, and also operated by hand for lubricating the cylinder previous to starting up.

The reservoir containing the oil which feeds this pump should be fitted with a gauge glass so that its contents can be noted. The base should be fitted with a filter, to prevent all dust and foreign substance from entering the cylinder.

Towards its extremity, the tube opening into the cylinder should be fitted with a drip-sight-feed arrangement and a retaining valve, which can be easily dismantled during the working of the engine, as shown in Fig. X.—41. The amount of oil stored in these apparatus should be sufficient to ensure lubrication of the different parts for at least five hours at full load without needing replenishment.

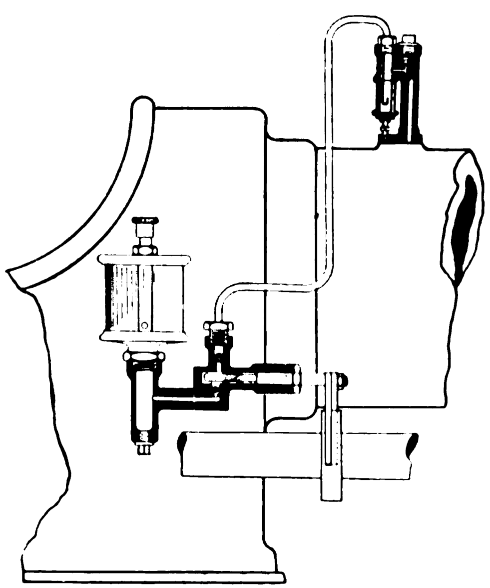


FIG. X.—41. Lubricating Oil Pump for Gas Engine cylinder.

For engines from 50 to 60 H.P. and upwards, the bearings of the crank shaft should be of the single ring lubricating type (Fig. X.—42). For engines of 100 H.P. and over, two rings should be employed in these bearings. The journals should be of the shell type and made of cast-steel lined with white metal. The oil reservoirs of the bearings should be fitted with oil level indicators with glass tubes and a drain cock. The admission valve spindle and more especially the spindle of the exhaust valve should be fitted with an efficient system of lubrication.

In motors above 400 to 500 H.P., lubrication should be ensured by a continuous flow of oil under pressure. This pressure should be sufficient to give positive lubrication to the large areas supporting the enormous weights, as in the case of the crank shafts of an engine from 2,000 to 3,000 H.P., which reach 20 inches diameter.

With regard to the lubrication of stuffing-boxes it is indispensable to use forced lubrication to ensure its penetration into the rings, as the "gastight" condition necessary depends to a great extent upon the free "play" which is guaranteed only by good lubrication.

The excess of oil in the cylinders is one cause of fouling by deposit and is the principal factor of premature firing. To reduce this inconvenience a blow-off cock is arranged on the cylinders as shown in Fig. X.—43. This valve is placed in the lowest portion of the cylinder. It is manipulated by hand whilst the engine is at work and gets rid of all excess of oil and foreign matters.

Double-acting engines, the exhaust valves of which are not arranged in the base, should be arranged with an automatic evacuation of oil by means of a blow-off valve as in single-acting engines. *Schuchtermann & Kremer* operate this valve mechanically.

With regard to oil consumption it has been ascertained from many

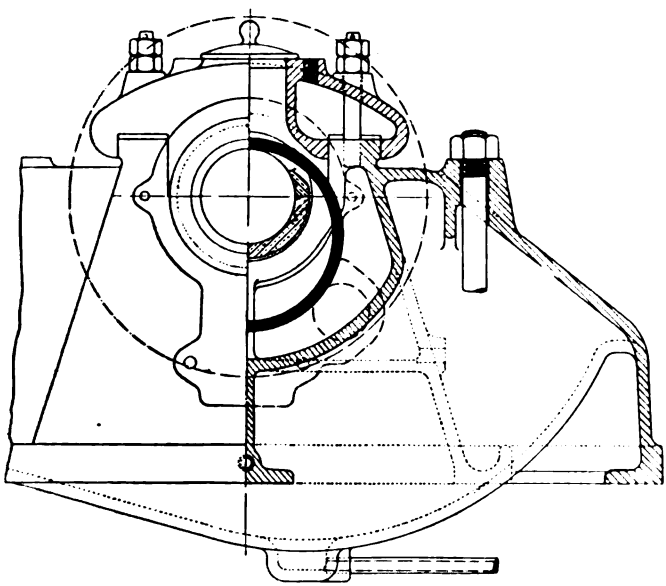


FIG. X.—42. Ring lubrication of main bearings.

installations that it does not exceed 0·035 to 0·043 ozs. per B.H.P. hour. The following figures have been taken in connection with a Nürnberg double-acting, twin tandem, 1,800 H.P. engine, during three months' work. The engine consumed for an average load of 1,500 B.H.P., one gramme of cylinder oil per H.P. hour, ·8 grammes of machine oil, and ·7 grammes of grease.

In connection with large gas engines, each detail should be fitted with a suitable system of lubrication. For cylinders, stuffing-boxes and exhaust-valve spindles, it is advisable to use forced lubrication by means of a pump, with sight-feeds separately adjusted for each part. For the main bearings of the engine shaft, cross-head, guides, and connecting-rod brasses, a central distributor with adjustable sight-feeds is generally installed.

For the cam shaft bearings, eccentrics, and the various joints, grease lubrication should be used. It is a good plan to collect the excess of oil in the reservoir placed below the engine, in order that it may be used a second time after being filtered.

Starting.—A half-compression cam is used for small engines, being kept in gear during the turning of the engine shaft—either by pulling the fly-wheel round by its arms or rim or by means of a crank-handle —until the first power strokes have set the engine to work. With an

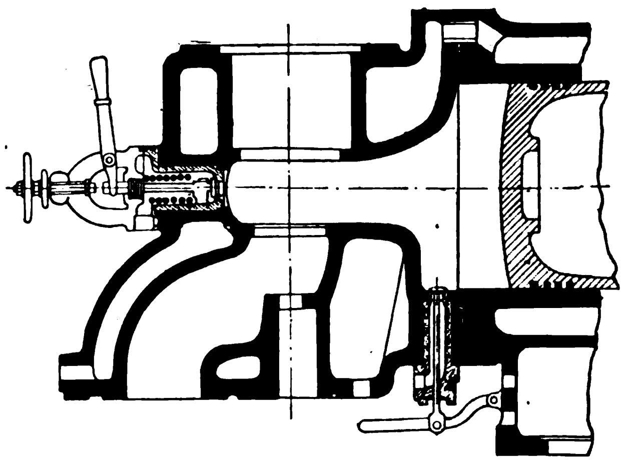


FIG. X.—43. Blow-off cock for evacuation of excess lubricating oil.

engine having a normal compression pressure of about 170 lbs. per square inch, the starting compression pressure will be about 100 lbs. if the exhaust-valve remains open for one-third, and about 50 lbs. if held open for two-thirds the length of the compression stroke. Beyond this limit, the force of the initial explosion will be insufficient to overcome the resistances and the inertia of the engine during the three succeeding strokes.

Explosion "self-starters" are only applicable to engines of low power. A charge of mixture, previously introduced while the engine is at rest, is exploded and creates a sudden and violent blow which may lead to accidents owing to the inertia of the moving parts. Moreover, their action with producer gas is somewhat uncertain.

However, some makers still use starting pumps for forcing in an explosive mixture behind the piston. The mixture can be either air and gas, or air and petrol or benzine.

The system now in general use is to utilise compressed air of about 150 to 200 lbs. per square inch, stored in a reservoir either by means of a compressor or by the engine itself.

When starting the engine, the crank is placed in the position corresponding to the outward expansion stroke by the use of a barring lever or toothed gearing.

Single-acting engines are supplied with compressed air by manually operating the valve shown at the back of the combustion chamber in Fig. X.—43. Repeated charges of compressed air are given

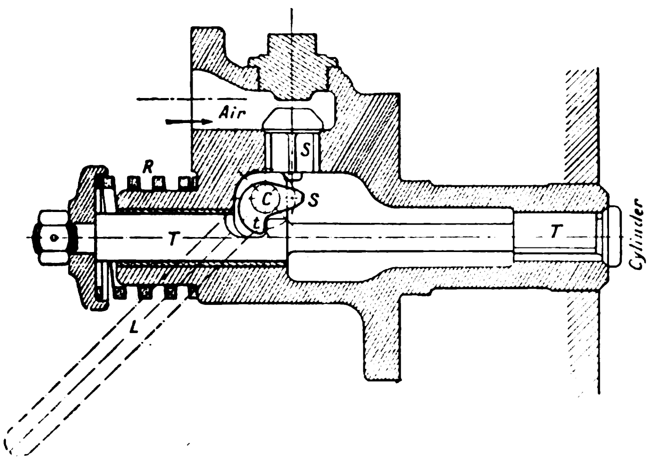


FIG. X.—44. Tangye compressed air Starting Valve. Section.

at each succeeding expansion stroke until the engine has gained sufficient speed to draw in and explode a working charge by its own mechanism.

The compressed air starting valve shown in Fig. X.—44 is used by *Tangyes Ltd.* It consists of a box containing a set of valves operated by the two solid bosses *t* and *s* on the cam *C*, which is moved by the external hand-lever *L*. The spindle-valve *T*, opens into the cylinder and is kept on its seat by the spring *R*. The spindle *T* has a notch in its "guide" portion, in which the boss *t* of the cam *C* engages, so that, by a movement of the lever *L*, the spindle is made to open the valve *T* communicating with the cylinder, whilst at the same time, the boss *s*, of the cam *C*, opens the back pressure valve *S* and thus admits the compressed air. The reversed movement of *L* closes the valve *T* and the back-pressure valve *S*, the latter shutting off the compressed air.

Fig. X.—45 shows the three different positions of the lever for starting, compressing, and closure, each position being secured by means of a pin.

The capacity of the air reservoir should be sufficient to permit at least four successive starting charges to be withdrawn without the pressure falling below 60 lbs. per square inch. It should of course be absolutely airtight.

Fig. X.—46 shows a complete compressed air starting set as arranged by the *Gasmotoren Fabrik Deutz*. This firm, for their large engines, make use of the ordinary inlet valves, thus avoiding any special

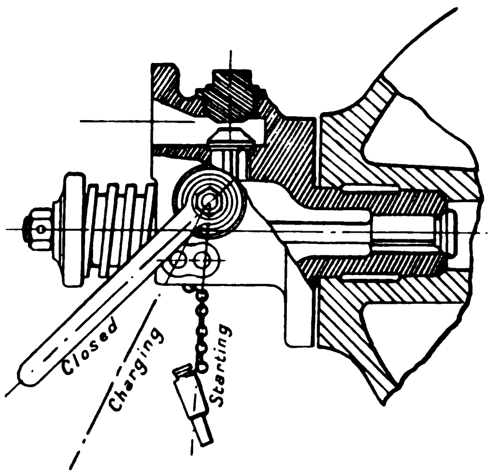


FIG. X.—45. Tangye compressed air Starting Valve.

starting valve for the compressed air. Until the engine is well under way, these valves work on the two-cycle principle.

The largest gas engines are usually fitted with an electric turning gear, so that when necessary, in addition to the ordinary use for starting, valve adjustments can thus be easily made. The turning gear should be complete with an automatic arrangement for throwing the gearing out of operation when the engine takes its own charges. For large engines the compressed air is carried up to 300 lbs. per square inch. The admission to the cylinder is made by gearing operated by the cam shaft and should be quite automatic—that is to say, the admission of air should cease as soon as the first explosion has been produced.

Mean Pressure.—The makers of single-acting engines more especially designed for use with producer gas have adopted the commendable attitude of calculating the power of their engines upon the basis of a mean pressure of 65 to 70 lbs. per square inch of area. In adopting this rule they do not sell their engines above the power limit, and they look upon this as being analogous to the conditions applying to steam engines, in that a certain margin is available for carrying

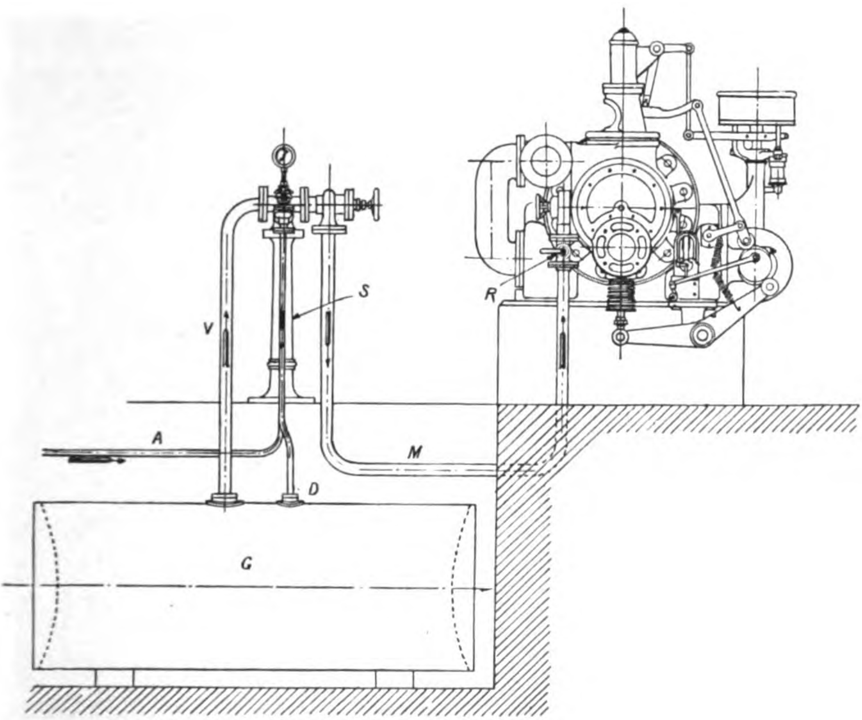


FIG. X.—46. Gasmotoren Fabrik Deutz compressed air starting equipment.

an overload of 10 or 15 or more per cent. above the normal rated output. This margin is taken as being sufficient to compensate for the variations in the richness of the gas or temporary derangement of adjustments.

In practice, with a well constructed and properly adjusted engine, the mean pressure easily exceeds 70 lbs. per square inch. The diagrams, Figs. X.—47 and 48, were taken from British engines, and are interesting on account of the mean pressure represented. That in Fig. X.—47 is 90 lbs. per square inch with pressure producer gas, and that in Fig. X.—48, with town gas, is 85 lbs. per square inch. The author has often remarked that British built gas engines, although working with comparatively low compression pressures, are able to realise higher mean pressures than German engines with higher com-

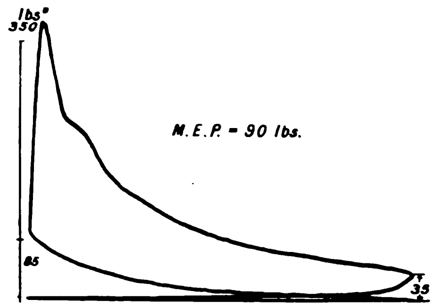


FIG. X.—47. Indicator diagram from Engine served with pressure producer gas.

pression pressures. He attributes this fact more particularly to the shape of the combustion chamber, and in confirmation of this opinion the diagrams in Figs. X.—49 and 50 are given, showing equally high mean pressures. These were obtained from a double-acting Otto-Deutz engine, that is to say, from an engine having a dissimilar com-

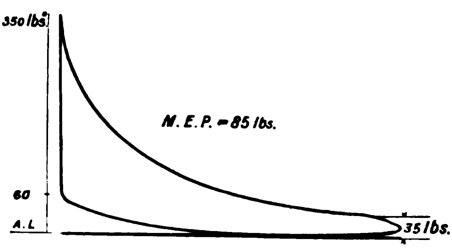


FIG. X.—48. Indicator diagram from Engine served with town gas.

bustion chamber end to that usually adopted in single-acting engines, and by comparison with Figs. X.—47 and 48 the respective compression pressures may be clearly differentiated.

The cylinder dimensions of double-acting engines are calculated upon the basis of 80 lbs. per square inch mean pressure with coke oven gas, and 70 lbs. with blast furnace and producer gas. The linear piston speed also tends to increase, and, except perhaps in the very large engines when the number of revolutions is limited by considerations of inertia, it now frequently reaches 850 feet and more per minute.

Consumption and Efficiency.—Modern gas engines have high mechanical efficiencies owing to the proportional reduction of their

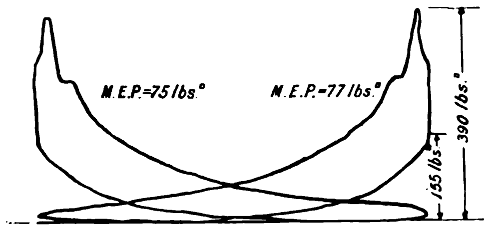


Fig. X.—49. Indicator diagram from double-acting Deutz Engine.

weight and to the care exercised in their manufacture. It is admitted that double-acting, four-cycle engines are able to attain 90 per cent. mechanical efficiency, while two-cycle engines can scarcely reach 80 per cent. This difference, being due to the work absorbed by the air and gas pumps, does not, however, detract from the high value of

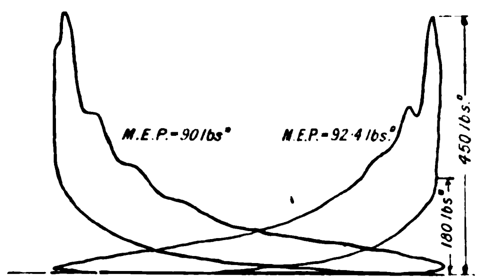


Fig. X.—50. Indicator diagram from double-acting Deutz Engine.

those splendid engines of which the Koerting and Oechelhäuser are the standard types. Double-acting, four-cycle engines attain a thermal efficiency of 28 to 30 and even 31 per cent. per B.H.P., that is to say, they develop 1 B.H.P. per hour with a heat consumption of less than 8,700 B.Th.U.

The volumetric consumption of the different power gases employed corresponding to these high efficiencies, expressed in relation to their average calorific value, is as follows :—

	Cubic Feet.	B.Th.U per Cubic Foot.
1. Coke oven gas	19·5	450
2. Mond producer gas	62·0	140
3. Anthracite producer gas . . .	65·0	135
4. Blast furnace producer gas . . .	77·0	107

The relative proportions of the constituents in each power gas is shown in Fig. X.—51, the symbols being as follows :—Saturated

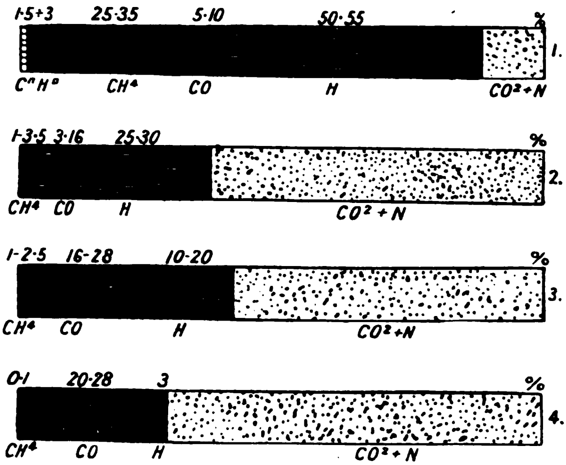


FIG. X.—51. Relative proportion of constituent gases in (1) coke oven gas. (2) Mond gas, (3) anthracite gas, (4) blast furnace gas.

Hydro-carbons, CnHn ; Methane, CH_4 ; Carbon-monoxide, CO ; Hydrogen, H ; Carbon-dioxide, CO_2 ; and Nitrogen, N.

In connection with the fuel consumption of gas engines a few results selected from about 400 tests carried out by the author, either as a consultant or as an expert appointed by the legal tribunals, may be referred to.

In 1906 a test was mutually arranged between the engineers of *Ehrhardt & Sehmer*, and those of the *Konigliche-Berginspection*, at Heinitz Saarbruck, upon a four-cylinder, double-acting 600 H.P. engine. After four months of constant work, without any previous cleaning, this engine was put under test with coke oven gas of about 450 to 470

B.Th.U. per cubic foot. It showed a consumption of 8,000 B.Th.U. per B.H.P. hour. The mechanical efficiency under the load carried was 83 per cent. The engine was new, and was tested with a three-phase dynamo mounted on the engine shaft. The principal details were as follows:—

Diameter of cylinder . . .	24·5	inches (620 mm.)
Stroke of piston	29·5	„ (750 mm.)
Diameter of piston rods. . .	6·8	„ (170 mm.)
B.H.P. average	520·0	
Revolutions per minute. . .	150·0	

It will be observed that the figures show that the thermal efficiency per B.H.P. was nearly 31 per cent., or, if based on the I.H.P., 37·5 per cent.

Similar trials are unfortunately rare because it is not always possible to arrange for a gas holder of sufficient capacity to measure exactly the gas consumed.

The following figures relate to tests made on a series of smaller installations, and which are interesting on account of the different fuels used.

SUMMARY OF TEST RESULTS.—SINGLE-ACTING NÜRNBERG ENGINES.

—	1. Royal Foundry of Würtemberg at Wasserflingen.	2. Imperial Post Office at Hamburg.	3. Municipal Electric Station at Greisswald.	4. Kurtz & Zanders, Wendelstein, near Nürnberg.
Fuel	Anthracite	Coke	Lighting Gas	Anthracite
Test load B.H.P. .	107·4	110	152·8	66
Normal load B.H.P. .	100	100	150	60
Fuel consumption per B.H.P.	0·775 lbs.	0·925 lbs.	15·8 cub. ft.	0·760 lbs.
B.Th.U. per B.H.P. per hour fed to Producer.	10,850	10,800	——	10,900
B.Th.U. per B.H.P. per hour excluding Producer.	8,700	8,150	8,850	8,730
Thermal efficiency of Producer per cent.	80	75	——	80
Mechanical efficiency of Engine per cent.	80	80	78	80
B.Th.U. per I.H.P. per hour.	6,950	6,500	6,900	6,940
Thermal efficiency per I.H.P.	36·3	38·3	36·6	36·3

A large number of further tests will be found in Chapter XVI.

Exhaust.—The thermal efficiency of gas engines being about 30 per cent., there is, therefore, a loss of 70 per cent. of heat by radiation,

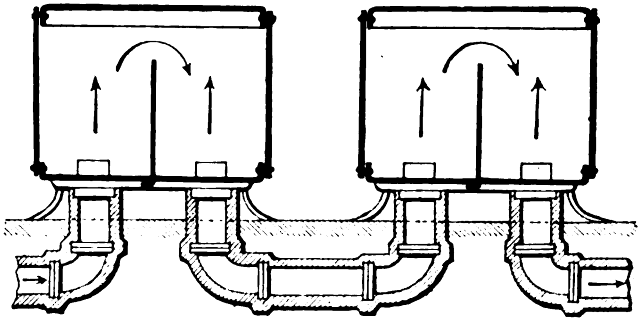
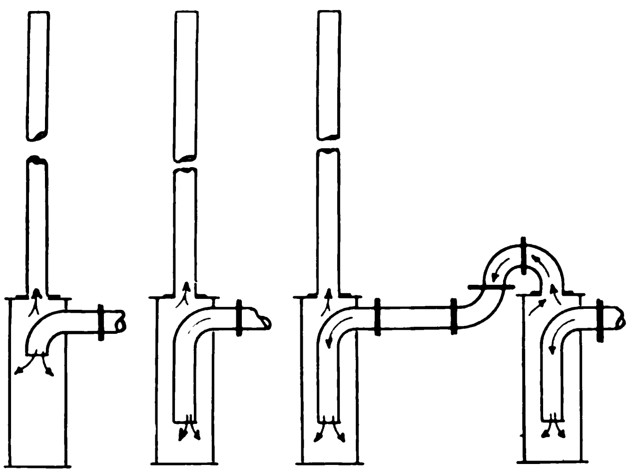


Fig. X.—52. Utilisation of waste heat in exhaust gases.

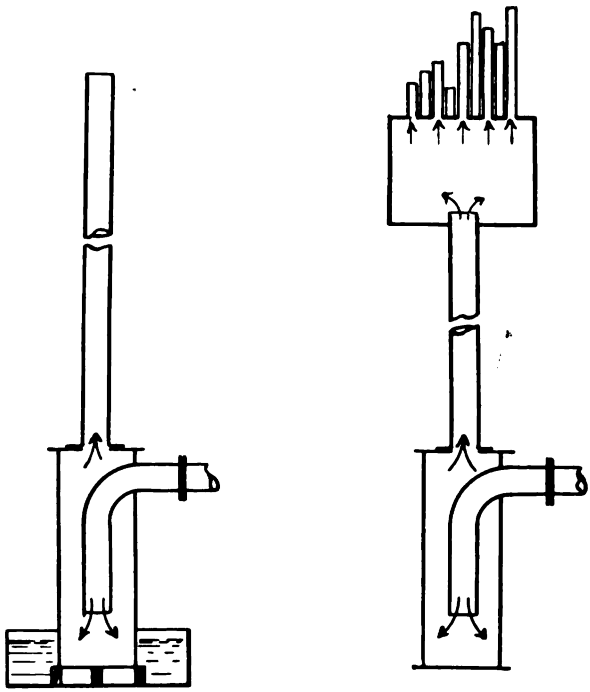
cooling water, and exhaust gases. The latter have a temperature of about 750 to 900° F., and the quantity of heat escaping in this way represents about 35 per cent. of the total number of units supplied to the engine. When large gas engines are used some of this heat should



Figs. X.—53. 54. 55.
Experiments in exhaust silencing.

be recovered. Different schemes have been adopted in this direction, and in some cases by raising steam in a special type of boiler which receives the hot gases, the steam thus generated being used either to drive an engine or some similarly useful purpose.

Upon this basis, supposing that about 3,000 to 3,600 B.Th.U. per B.H.P. developed by an engine were available, if this could be utilised in the production of steam, an output not exceeding 2·2 lbs. of steam at 70 to 85 lbs. per square inch per B.H.P. developed could probably be obtained. But in order to obtain this recovery of waste heat, the engine must work at least about two-thirds of its maximum power; if not, the exhaust gas is expanded down so much, by the modern



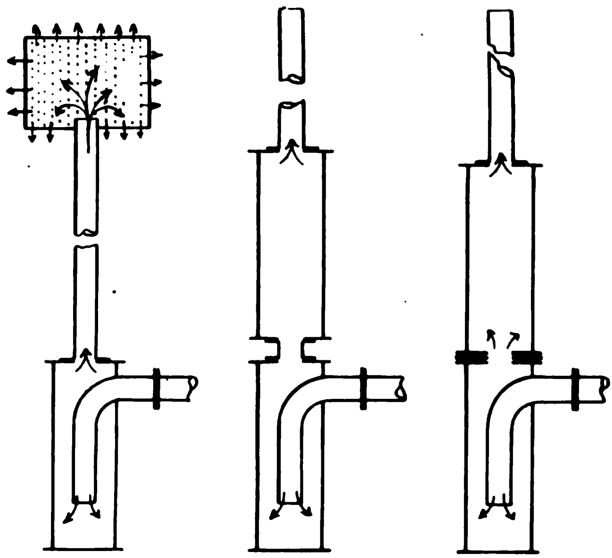
FIGS. X.—56. 57.
Experiments in exhaust silencing.

methods of governing, that it is not hot enough to give up any appreciable quantity of heat for recovery in this manner.

For instance, an engine of 100 B.H.P. working at full load would give 220 lbs. of steam per hour, which, used in a non-condensing engine, would produce 6 or 8 H.P. It will be seen, therefore, that the amount of heat recoverable is not of very great consequence, owing to the successive transformations of heat into work.

It is more advantageous to utilise the waste heat of the exhaust gas in a heating installation either by hot water or by means of a series of stoves, in the form of a drum, in which the gas circulates as shown in Fig. X.—52.

Several British firms, such as *Andrew Barclay & Sons, Ltd.*, of Kilmarnock, build steam boilers specially constructed for the recovery of the waste heat from exhaust gases. To obtain the maximum useful effect the boilers must be placed as near as possible to the engine in the immediate vicinity of the exhaust outlet. The boilers are relatively bulky, and it is necessary to arrange a specially appropriate site for their accommodation. They should be fitted with efficient arrangements to eliminate the condensed water contained in

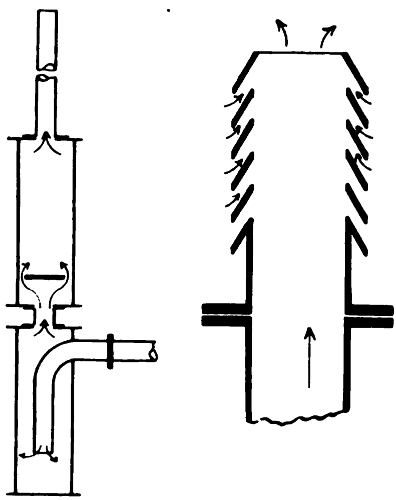


FIGS. X.—58. 59. 60.
Experiments in exhaust silencing.

the gas, as these are usually corrosive and cause rapid deterioration of the wrought iron plates.

With respect to the noise occasioned by the exhaust, the author has already discussed different arrangements which can be adopted to silence this, in the fourth chapter of his book, "Gas Engines and Producer Gas Plants." The solution of the problem sometimes presents great difficulties, and the following is an abstract from the *Gas and Oil Engine Record* of 15th March, 1906, relating to such difficulties in connection with two 200 H.P. three-cylinder vertical engines installed in a building on the side of a hill. The working of the engines had caused the neighbours to complain, those living *above* the level of the building objecting to the noise, while the air pulsations made the windows of some of the buildings *below* vibrate. Fig. X.—53, shows the original arrangement. The three cylinders were connected by one pipe common to all, which opened into a large expansion chamber from which the gases escaped to the free air. The vibration set up was felt for 150 yards from the works.

The first experiment to obviate the defects was to inject water in the exhaust pipe. The noise was reduced as regards the tenants living on the hill below the engine room, but no perceptible difference was made for those above. The next step, as shown in Fig. X.—54,

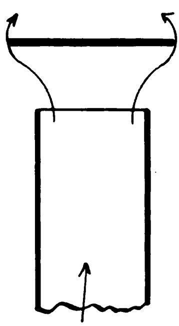


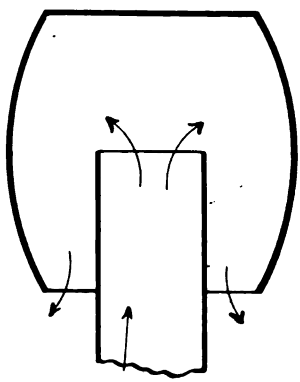
Figs. X.—61. 62.
Experiments in exhaust silencing.

was to lengthen the bend in the interior of the expansion chamber. No improvement was obtained. Two expansion boxes in series were next tried, as shown in Fig. X.—55, and while this arrangement made the noise disappear, the consequent back pressure on the engine was too great for proper working.

It was then noticed that the exhaust vapour travelled up the pipe with a rotary motion, and, to prevent this, the upper portion of the pipe was fitted with a wooden baffle, but the back pressure was not relieved. The expansion box was then fitted, without a bottom plate, above a leaden basin containing enough water to make a hydraulic joint as in Fig. X.—56. This was unsuccessful. The following experiment consisted in fixing a large box of about four feet square at the utmost end of the pipe. The top of the box was then pierced with a number of 3-inch diameter holes, and, through each hole, sheet iron pipes protruded of varying lengths of from 1 inch to 3 feet (Fig. X.—57). The end in view was to create a number of streams of exhaust, each out of time with any other, and thus to avoid air pulsations. No difference was noticed in the effects of these vibrations on the window panes. The diminution of noise was imperceptible. Following the same idea, the top of the box was renewed, and a large number of ½-inch diameter holes were bored on all sides, as shown in Fig. X.—58. This arrangement gave more satisfactory results, and the engine worked some time in this way.

In order to increase the expansion by using two boxes, which had



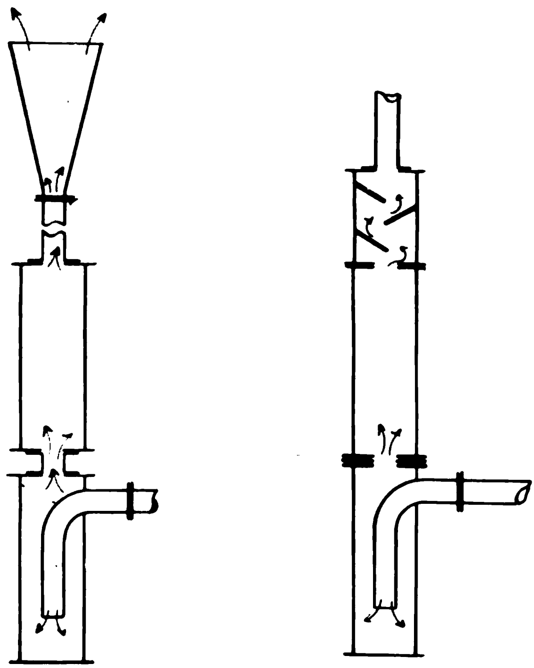


FIGS. X.—63. 64.
Experiments in exhaust silencing.

previously been done (Fig. X.—55), the second chamber was placed on the first by means of a collar of the same diameter as the exhaust pipe (Fig. X.—59). This arrangement appreciably reduced both the noise and vibration, but in order to make a permanent fixture, the collar was replaced by a metallic diaphragm, as shown in Fig. X.—60. However, on calm days, the casements of one particular house still rattled.

Fig. X.—61, shows that an iron baffle plate was directly placed above the first exhaust chamber; this was found to make a slight improvement. The end of the pipe was then fitted with a cowl, as shown in Fig. X.—62. Very little improvement resulted. A cast-iron plate, placed above the orifice (Fig. X.—63), produced vibrations, as also did a tub placed as shown in Fig. X.—64, while with the latter the back pressure was increased by about 3 inches of water.

A sheet iron funnel about 9 feet long, tapering from 9 to 18 inches (Fig. X.—65) reduced both noise and vibration by enabling the gases to pass away at low velocity, but a permanent cast-iron pipe



FIGS. X.—65. 66.
Experiments in exhaust silencing.

could not be fitted, owing to its excessive weight. A shorter funnel was useless.

Finally, the arrangement shown in Fig. X.—66, was the solution of the difficulty. This consisted in adding, above the second box, a baffling cylinder of the same diameter as the box and communicating directly with the air. The back pressure was practically nil, and the noise and vibration disappeared.

CHAPTER XI

GOVERNING AND VALVE GEARS OF INTERNAL COMBUSTION ENGINES

THE system of valve gear of a gas engine includes the admission and exhaust valve gear with the mechanical details that control the speed under the action of the governor.

In Chapter VIII. the methods of control in two-cycle engines have been described. In modern four-cycle, or "Otto" cycle engines, the valves are invariably operated by the side shaft (or cam shaft, or half-speed shaft), and their movement is obtained either by cams or eccentrics. On the whole, the different methods of general design do not permit the superiority of either cams or eccentrics to be determined. Both work equally well when properly designed for the work they have to do. The combination of eccentrics and rolling levers, so much in favour, for large gas engines particularly, because of their smooth and silent operation, involves the inconvenience of demanding great amplitudes of movement and bulky details.

The different systems of governing may be classed as follows:—

1. **Constant Ratio and Constant Volume.**—Hit-and-miss systems.
2. **Variable Ratio and Constant Volume.**—Variable stroke of the gas valve by a conical cam, or stepped pecker-block, either by continuous motion or automatic cut-off.
3. **Constant Ratio and Variable Volume.**—Throttle Butterfly valve, variable stroke of mixture valve by continuous motion or by automatic cut-off.
4. **Combined Regulation.**—By application of the two preceding methods.

1. **Constant Ratio and Constant Volume.**—The first method of control has been carried out by the "hit-and-miss" system, which for a number of years has been almost universally adopted for engines working with town gas or liquid fuel. This system has rendered substantial service to makers, and it has permitted very economical working to be realised. The development of producer gas and the building of engines in units of increasing power has shown that the "hit-and-miss" system, although excellent for town gas engines, is less suitable and even detrimental for large producer gas engines.

Although it has been definitely abandoned by the majority of makers, this method of regulating the speed and the output of the engine by varying the number of explosions constitutes the simplest, most efficient, and most economical means of control.

No other mechanical device can compare with this governor for simplicity, while its minute dimensions permit great sensibility. It has little or no resistance to overcome and is not exposed to any detrimental reaction. It limits to a fraction of an inch the displacement of a light piece of metal, which, upon the slowing of the engine, is interposed between the gas valve spindle and its actuating lever and which is pulled away when the engine speed increases. In this manner the gas valve opens and admits the charge or remains closed during several cycles, in the course of which a charge of air only is admitted to the cylinder.

The economic advantage of the system as regards consumption enables the proportions of gas and air in the mixture to be adjusted once for all, each charge admitted constituting a perfect explosive mixture and giving a maximum of useful work. But, in spite of the favour still given to the "hit-and-miss" system by several British and American makers, it must be recognised that this method of control does not result in smooth and silent working of the engines, which must be a feature if they are to compare with steam engines.

The temporary cessation or suppression of charges, after several cut-out cycles have been made, sets up sudden and violent shocks which may be without immediate serious consequences in small engines, but become prejudicial to engines ranging between 30 to 75 H.P., and involve real risks in engines of 100 H.P. and above.

Besides this, the "hit-and-miss" system does not answer well for engines working under light loads, or at no loads, with gas from suction gas producers. In these circumstances, the gas is taken after three, four, or even five cut-out cycles, and then the draught of air through the generator is not constant enough and the fire becomes extinguished, or, at least, produces very weak gas, due to the lack of furnace activity.

Those makers who apply the "hit-and-miss" system to their engines are familiar with the series of very special phenomena that it gives, and particularly (1) that after a cut-out the following explosion is stronger in some cases and weaker in others, according to the form of the combustion chamber and the position of the valves and ignition device; (2) that it frequently produces very high pressures from the violent explosions, the consequences of which are so much the more harmful to the working parts as the engines become larger, because the moving masses become heavier and the inertia more considerable; (3) that the intermittent explosion produces cyclical variations or irregularities extending over a certain number of revolutions of the fly-wheel, and that these cyclic irregularities are incompatible with the conditions of driving dynamos for electric lighting, &c.; (4) that to counteract these it is necessary to make use of extra heavy fly-wheels, which constitute an additional load on the engine and reduce the mechanical efficiency.

Against these defects it is true that the "hit-and-miss" system is the most simple governing arrangement to construct, the least liable to derangement, and presents the following further advantages:—(1) Low consumption at light loads, due to the fact that the mean pressure tends to increase rather than to diminish under such conditions, as has been mentioned above; (2) the proportion of the mixture, being adjusted once for all, remains constant and always realises a maximum mean pressure without necessitating high compressions; (3) regularity as regards variations of speed produced by variations of load. This regularity remains between the limits of $1\frac{1}{2}$ to 2 per cent. of variations in the number of revolutions between the "no-load" and "full-load" speeds of the engine.

The general statement that the "hit-and-miss" system gives a lower consumption than other governing methods cannot be maintained to-day. It has been practically demonstrated from a number of impartially conducted tests that a good system of governing by variable quantity concedes nothing to the "hit-and-miss" method.

In Germany all the makers have abandoned "hit-and-miss" governing, and those who do not yet build the larger engines with variable admission have chosen temporarily an intermediate system, which, if it is not more economical, has, however, the merit of ensuring smoother and more regular working without necessitating the use of extra heavy fly-wheels.

2. Variable Ratio and Constant Volume.—The characteristic of this system is an admission of air throughout the whole of the suction stroke whilst a variable admission of gas changes the ratio or composition of the mixture according to the load. The compression is therefore constant. The admission of gas commences sooner or later in the suction stroke and stops just before, or at the end of the stroke at about the time the main inlet valve closes.

This is contrary to the method adopted in the steam engine, in which the admission begins with the stroke and finishes earlier or later during the stroke.

It is with a view to obtaining the richest mixtures at the end of the piston stroke and to ensure that the explosive charge shall remain in the vicinity of the igniter, that many makers have arranged the admission of gas in this manner rather than at the beginning of the stroke. Some makers still seem to believe in the old theory of the stratification of fluids and to respect it as a valuable tradition in spite of the fact that specialists have proved that no stratification of gases is possible. One of the most reputable German builders has frankly resigned the theory of layers or stratification, formerly defended by Otto, and he has, without any apparent prejudice to the thermal and mechanical efficiency of his large engines, operated the variable method of governing by admitting the gas, not along with the admission of air, but from the commencement of the period of admission, cutting off its introduction, more or less early, during the forward movement of the piston, so that at the end of the stroke there is an admission of pure air only.

In such conditions, if the theory of stratification be a correct one, the back of the cylinder or combustion chamber should be filled with air alone, confined round the igniter, whilst a cushion or layer of rich mixture remains contiguous to the piston. To argue in this way it is necessary to admit the possibility of the layers of fluids remaining in their respective positions in spite of the swirling of the fluids during the admission period caused by the throttling of the gas and air passing through the ports, at a velocity of from 130 to 230 feet per second. It is also necessary to admit that the inertia of the gases drawn by the piston in one direction and suddenly pushed back in another, would allow these fluids of a sensibly equal density to escape diffusion, and to remain stratified and separated from each other until the end of the compression stroke.

This interesting question is, of course, entirely hypothetical, and therefore the author prefers to confine his remarks to some physical facts relative to the variable mixture at constant volume.

Theoretically, the variation of the ratio of mixture ought to be made exclusively under the control of the governor, so that it would only take place according to the work done by the engine. Hence, the mixture ought to remain constant for a constant load. But such is not the case, because the air, continuously admitted during the whole suction stroke, acquires a speed approximately corresponding to that of the piston, whilst the gas valve, which opens earlier or later, and closes invariably at the end of the stroke, sets up a restriction entirely depending upon the position of the piston when such opening takes place. Thus the speed or varying velocities of the gas do not synchronise with those of the air. Moreover, the influence of the inertia of the moving fluids plays here an important part.

Mr. Reinhardt has proved that the air acquires under the influence of the constant suction, an accelerated speed in the pipes and ports, whilst the gas, which is progressively admitted, must start from zero, and thence accelerates its velocity in passing through an orifice of a continuously varying area.

The gas valve opens during the outward stroke of the piston, and, when the position of the latter corresponds to its maximum speed, the gas which is at rest in the gas pipe is suddenly urged forward by the suction. It is this lack of synchronism between the velocities of the streams of gas and air that spoils the proportion of the explosive mixture and retards its combustion at low loads. This causes the back firings which are more often observed in engines governing on the mixture than in those governing on the volume.

The tendency to back firing at low loads is also due to the fact that the mixture may be impoverished and diluted to such an extent that it is no longer explosive in the real meaning of the word, but merely burns slowly from the beginning of the expansion stroke till the end of the exhaust stroke. This burning mixture is liable to fire the following charge entering through the inlet valve, which is usually opened in advance of the actual piston stroke. When this gas is burning slowly, it is much more liable to cause back-firing, as, owing to the absence of an explosion, it has hardly the necessary pressure to help it to escape at a great speed through the exhaust.

To prove that this explanation of a rather complex phenomenon has not at all a hypothetical foundation, the results of experiments on the inertia of fluids moving in the pipes and cylinders of the engines, which the author has already published, may be recalled.

If an indicator diagram be taken with a light spring, with a stop to restrict the movement of the indicator piston during the compression cycle of the engine, no explosion will be recorded from the engine cylinder, but only the first portion of the compression curve will be traced as well as the end of the expansion curve, and the entire exhaust and suction curves.

Fig. XI.—1 shows two consecutive diagrams taken from an engine. The first one (in thick lines) shows the pressures prevailing during a portion of a power cycle during which an impulse is given to the piston, and the second one (in thin lines) shows the pressures when a charge of mixture has been admitted, compressed, expanded, and then discharged by the sole effect of the in-stroke of the piston, the ignition device having been intentionally put out of action to prevent the explosion of the charge.

The exhaust back pressure in the case of the explosion diagram (thick lines), measured by the scale of the indicator spring, shows a resistance of only 2 lbs., whilst in the diagram without an explosion (thin lines) this back pressure reaches 6·3 lbs. This proves that the gases without pressure, when the exhaust period commences, have encountered the resistance due to the inertia of the fluids at rest in the exhaust pipes during the inward stroke of the piston.

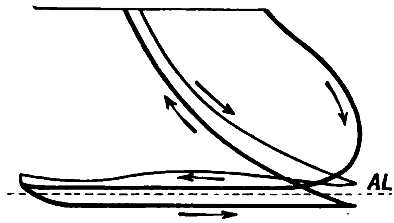


FIG. XI.—1. Resistance diagram, "power and "cut-out" cycles.

It will be observed that the expansion curve runs at a certain distance from the compression curve, owing to the heat borrowed from the walls of the cylinder, which has caused the charge to expand. Increasing back pressure to the exhaust takes place also when dealing with very poor mixtures, burning slowly instead of exploding. This phenomena is illustrated in Fig. XI.—2, obtained from different tests, in which the mixture has been impoverished to such an extent that the very late ignitions are recorded, causing much greater back pressures than when the charge was exploding, expanding, and then discharging under high pressure in a normal manner.

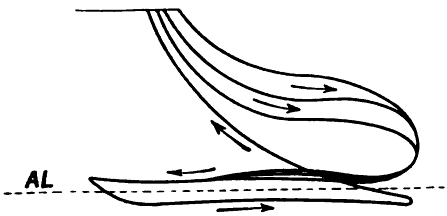


FIG. XI.—2. Resistance diagram, slowly burning, weak mixtures.

Back-firing may occur in any of the different ways which are shown by the cards which have been successfully isolated from the normal diagrams.

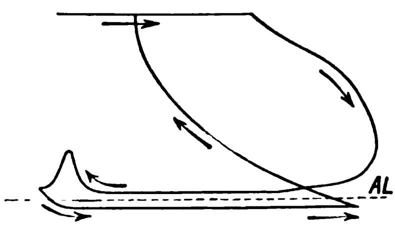


FIG. XI.—3. Resistance diagram showing back-fire upon admission of fresh charge.

In the first case (Fig. XI.—3), the burning gases have fired an explosive charge, remaining behind the inlet valve during the inward stroke of the piston, when the admission valve was opening in anticipation before the dead centre.

In the second and third cases (Figs. XI.—4 and 5), burning gases, confined in the combustion chamber till the beginning of the suction stroke took place, caused the new charge entering the cylinder to explode.

In order to avoid firing any mixture liable to remain in the valve box behind the inlet valve, this space must be scavenged or swept out by a final admission of air. This can be easily done by closing the gas valve at about the end of the suction stroke, so that air is admitted alone until the main inlet valve is closed. The space between the two valves must also be reduced to a minimum, such as by locating the air and gas valves in the main box concentrically to the spindle of the main inlet valve.

The arrangement of the gas and mixture valves within a common box presents some additional advantages by permitting rapid and close governor control. The existence of a large dead space between the regulating valve and the inlet port to the cylinder, permits the mixture

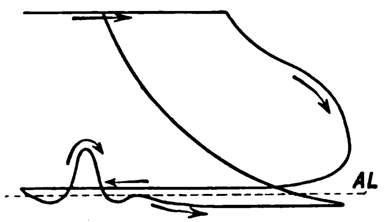


FIG. XI.—4. Resistance diagram showing back-fire during first portion of suction stroke.

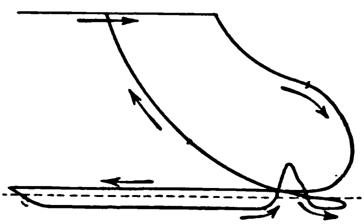


FIG. XI.—5. Resistance diagram showing back-fire towards the end of suction stroke.

contained in it to escape the control of the governor when the latter changes its position between two consecutive cycles.

The sensitiveness of the governor's action may especially be influenced in the case of multi-cylinder engines, as for example, a twin-tandem, four-cylinder, double-acting engine.

The single valve box, however, has some drawbacks in engines of the single-acting type, below 100 to 150 H.P., designed for work with producer gas previously submitted to but one purifying and washing process. In such conditions the gas valve is liable to be coated with tarry matters, and, consequently, needs frequent cleaning. This is more difficult when the valve is embodied with others than it would be if it were independent, and, therefore, much more accessible.

The system of governing with variable ratio and constant volume is applied to a special class of large engines where the load and the speed remain practically constant, such as in those designed for operating dynamos in central stations, spinning and weaving mills, corn mills, &c.

In these engines, it is necessary to have recourse to high and constant compression pressures and to carefully balance the reciprocating parts in order to obtain smooth working, to compare favourably with high speed steam engines.

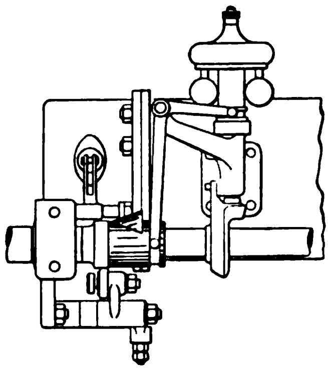


FIG. XI.—6. Variable quality governing by "Conical Cam."

The system of governing with mixture of variable ratio and constant volume can be effected either by means of a conical cam or a stepped-block, both designed for giving the gas inlet valve a variable lift, or by means of a mechanical device, which causes the stroke of the gas-valve to vary.

Fig. XI.—6 shows the system with "conical cam." This consists simply of a sleeve carrying a conical boss, which is moved by the governor longitudinally along the shaft so that the roller operating the valve is lifted to a greater or lesser extent, according to its position on the boss.

The same result may be obtained with the stepped-block fixed on the rod of the gas valve, such as in Fig. XI.—7. In this arrangement the governor moves the contact pecker up or down in such a way that the stroke of the gas valve is varied to suit the load upon the engine.

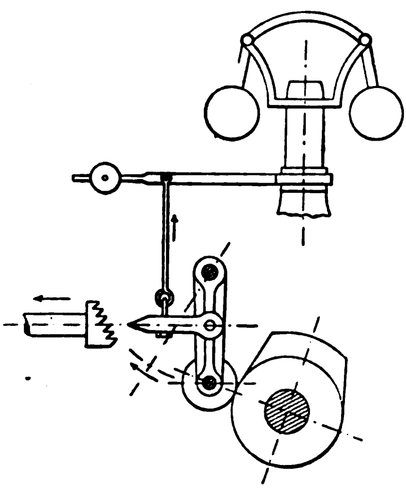


FIG. XI.—7. Variable quality governing by Stepped Pecker-block.

The first system is defective because the governor is subjected to violent reactions due to the obliquity of the cam, resulting in a lack of sensitiveness and steadiness, no matter how powerful the governor may be.

In the second system the governor is not subjected to these reactions because it does not actively interfere in the operation, but is only acting upon a light device incapable of transmitting any reaction, and is, in this respect, comparable to the "hit-and-miss."

While both the systems assure admission at every other revolution in the four-cycle engine, they both have the drawback of causing violent explosions at full load, which are prejudicial to the good working of the engine. Moreover, at low load, the ignition is necessarily late, owing to the impoverishment of the mixture, and, as a consequence, the efficiency of the explosive mixture is considerably lessened.

The indicator cards in Fig. XI.—8 and 9, illustrate these difficulties.

The systems above described are not suitable for higher compression pressures than 115 to 130 lbs., because premature ignitions would then result at full load, owing to the mixture being too rich. It must also be borne in mind that, with both of the systems mentioned, the admission of the gas to the cylinder does not take place at the same time as the admission of the air, except at full load, because the variable stroke of the gas valve is obtained by altering the times of both opening and closing the valve. To obtain a good homogeneous

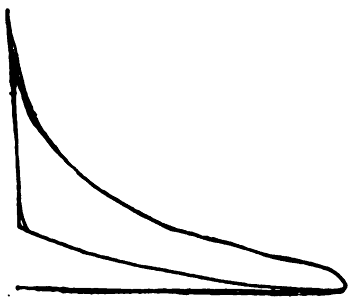


Fig. XI.—8. Indicator diagram showing violent explosion due to rich mixture.

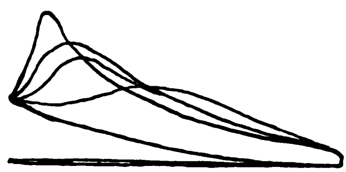


Fig. XI.—9. Indicator diagram showing loss of efficiency due to weak mixture.

mixture, the moment of opening and closing the gas valve ought to be constant and to correspond with the inner and outer positions of the piston. At low loads the gas valve is opened and closed for a very short period about the piston's midstroke, and when its linear velocity is at its maximum. This obviously is at variance with correct principles, as the valves should be operated so as to cause the fluids to move as much as possible at the same velocity.

The widespread popularity of suction gas producers for powers exceeding 30 H.P. has necessitated further study in connection with modern gas engines in order that the utmost reliability of such combinations may be secured.

It is obvious that the intermittent suctions caused by "hit-and-miss" engines when running light, are detrimental to the regular and continuous operation of the generator. It may, therefore, be predicted with certainty that the "hit-and-miss" governing, already abandoned by European gas engine firms, will soon fall into desuetude in the other gas engine manufacturing countries.

Some constructors have endeavoured to obtain the mixture at variable ratio and constant volume by operating on the stroke of the gas valve, either by using a continuous motion or by automatic cut-off; others have designed devices that vary the time of the opening or closing of the gas valve.

As the mechanical devices which have been applied are similar to those used for operating the admission valve in the system of governing classified under No. 3, the latter will first be examined and afterwards the similar contrivances used in the No. 2 system of governing will be dealt with.

3. Governing at Constant Ratio and Variable Volume.—The characteristic of this method of governing is to cause a variation of compression pressures proportional to the volume of mixture admitted into the cylinder.

It is a well-known fact that compression from 140 to 170 lbs. and over is indispensable in order to obtain low consumption and the complete combustion of very poor mixtures. Particularly is this so for engines fed either with producer gas or with blast furnace gas. The latter especially requires high compression owing to the fact that the almost entire absence of hydrogen in the gas is unfavourable to the quick propagation of flame.

With engines governed by variable volume of mixtures at constant ratio it is necessary to arrange for very high compression of full load volumes in order to ensure ignition of the small quantity of mixture admitted at low load. This compression should not, of course, be carried above the critical limit at which spontaneous ignition might occur.

Governing by admission of variable volumes of mixture at constant ratio is obtained by throttling the gaseous mixture before entering the cylinder. This throttling may be produced either by means of a butterfly valve in advance of which an automatic valve is sometimes fitted, or by varying the lift of the inlet valve itself, or by means of a mixture valve.

It would be premature to declare that this system of governing as applied in its different forms will preponderate over other systems, as both the methods 2 and 3 are used by makers in equal numbers.

Butterfly Valve.—The butterfly valve has been applied by the firms Schmitz, Soest & Co., Hornsby-Stockport, Tangye, Koerting and Benz, the two latter using an automatic mixing valve before the butterfly itself. The butterfly is worked by the governor, and is fitted immediately before the admission valve to the cylinder, where it operates on the mixture previously effected. Sometimes two butterflies are fitted, one controlling the gas and the other the air as in Fig. XI.—10.

With the butterfly arrangement a free space or clearance is formed between the butterfly and the inlet valve. This is filled with mixture formed before the action of the governor upon the butterfly, and cannot be scavenged by air only, since both air and gas are entering at the same time. In order to lessen the detrimental effect of this bulk of mixture the butterfly should be fitted as close to the combustion chamber as possible. The only advantage that can be claimed for this device is the facility with which the butterfly can be dismantled for cleaning purposes, a consideration of particular importance when producer gas is used, as the necessity of lifting the heavy admission valve and its cage, &c., is thus avoided.

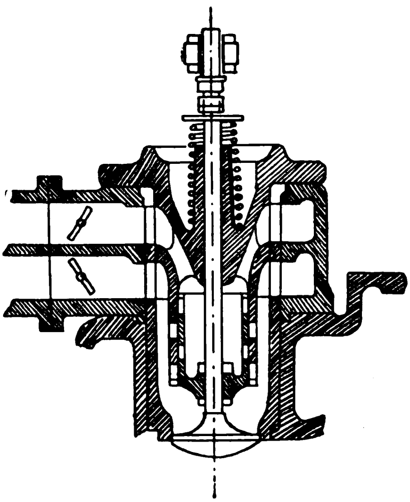


Fig. XI.—10. Governing by variable volume: mixture throttled by separate Butterfly Valves.

The butterfly valve in the arrangement under notice is the first detail that restricts the gas or the charge before its entry into the cylinder, and as it remains cold, these two features contribute to the precipitation of tar. In gas producing plants with a single purifier it constitutes one of the principal causes of disturbances in working, and it is necessary to remedy this as much as possible by providing means for its ready cleaning.

The arrangement of butterfly valves shown in Fig. XI.—11 (Koerting), prevents the gas under pressure from passing into the air supply pipe by means of an automatic back pressure valve which closes both air and gas passages at the same time. It acts also as a mixture regulator, and forms a useful complement to the butterfly valve owing to the fact that it ensures the air and gas remaining in the same proportion whatever the lift of the valve.

The partial vacuum caused by the suction of the engine operates the mixing valve, and, being modified by the position of the butterfly valve controlled by the governor, the lift is thus always in accord with the working of the governor. The mixing valve is necessarily exposed to the deposit of tar, but as it can be readily removed it does not involve much interference with the good working of the engine.

The Westinghouse Co. apply throttle governing to both air and gas by means of a double equilibrium valve actuated by the governor as indicated in Fig. XI.—12.

A. H. Alberger & Co., of Buffalo, for their four-cycle, single-acting,

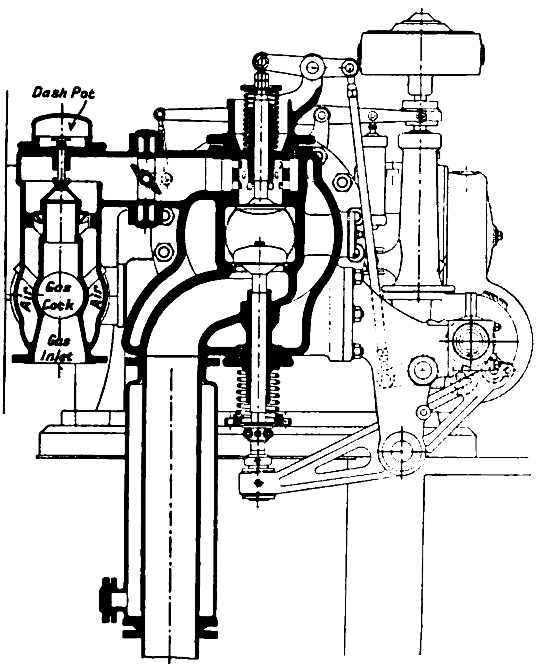


FIG. XI.—11. Butterfly Mixture Valve and automatic Gas Valve (Koerting).

tandem engines use a piston valve, shown in Fig. XI.—13. Longitudinal ports are provided in the piston valve and in the box which contains it, the two portions being shown separately in the drawing. The rotary movement given by the governor throttles the air and gas passages to a common port leading to the two cylinders. The piston valve is placed between the gas inlet above and the air inlet below, and in case of need a small injector for feeding petrol or gasoline can be provided, as shown in Fig. XI.—13, if the engine is to work with this liquid fuel. The diaphragm separating the two ingredients within the valve box can be moved by hand and adjusted so as to obtain the proper mixture ratio. Governing is thus effected by cutting off the admission of mixture sooner or later according to the power required.

On account of the frictional surfaces comprised in this arrangement it is not suitable for imperfectly purified gas, which might cause deposits to collect upon the piston valve. But it is, of course, well understood that American anthracite does not produce tar.

On the other hand, as the slide valve is suspended entirely from the

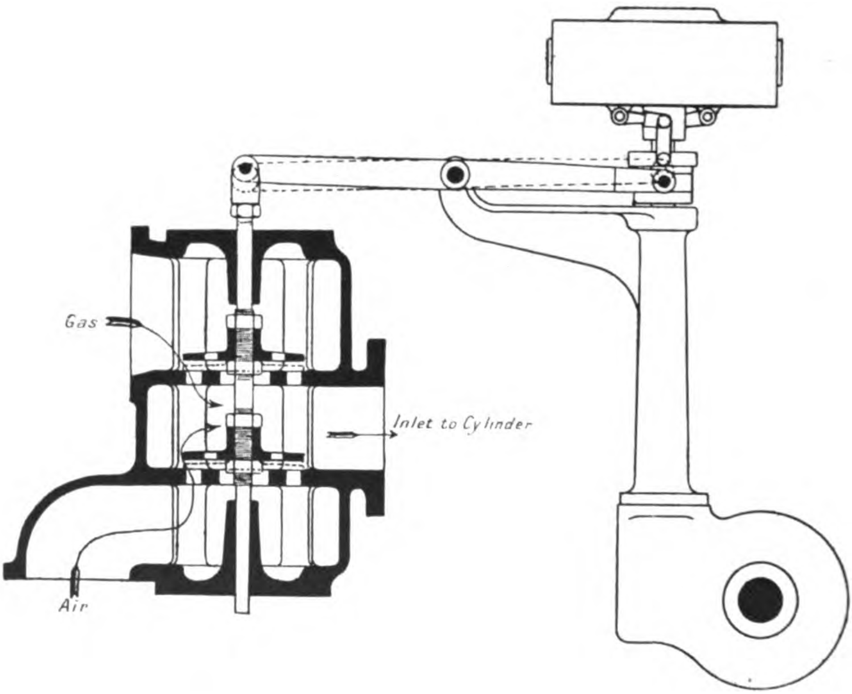


FIG. XI.—12. Westinghouse Air and Gas Throttle Governor.

governor and does not intercept the communication between the gas and air inlet, the arrangement does not permit any scavenging of the gas contained in the mixture port into the cylinder by means of a blast of air only. Moreover, if the gas be supplied under pressure, there is a tendency for it to pass into the air pipe displacing the air, while, if the gas is supplied at less than atmospheric pressure as from a suction gas plant, the air has a tendency to enter the gas supply pipe. The result is that there is a lack of stability in the proportions of the mixture.

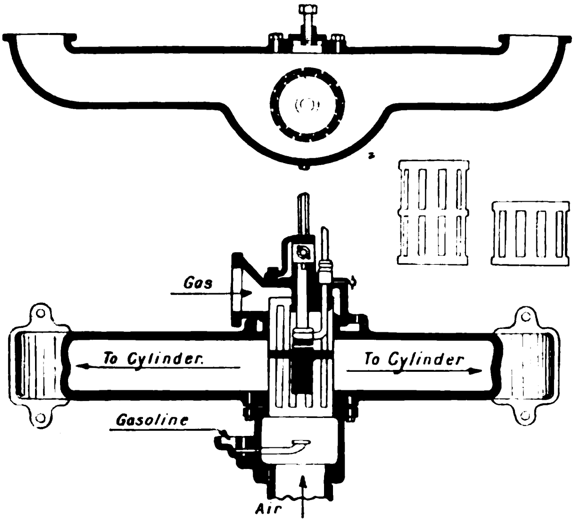


FIG. XI.—13. Alberger Piston Throttle Valve.

Variable Valve Lift.—If the governing be effected by the mixture inlet valve, it may be described as follows :—

"System of governing with mixture at constant ratio obtained by a variable opening of the main inlet valve under the control of the governor."

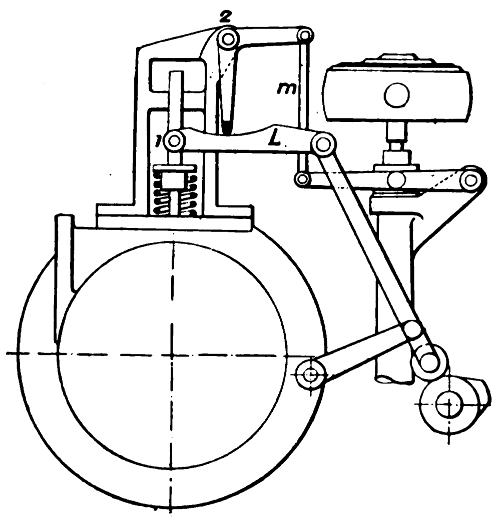


FIG. XI.—14. Gasmotoren Fabrik Deutz variable Valve Lift Device.

This method, already complex by its definition, presents some practical difficulties in application, to which it is desirable to call attention. It may be designed in several ways, all more or less commendable, and be operated by a variety of mechanical devices.

To ensure a good arrangement and to avoid defective working, it is indispensable to observe the following points :—

1. The time at which the opening begins or ends should be invariable.

2. Any direct reaction should be transmitted to the governor through the operating levers of the valves, and, if this reaction cannot

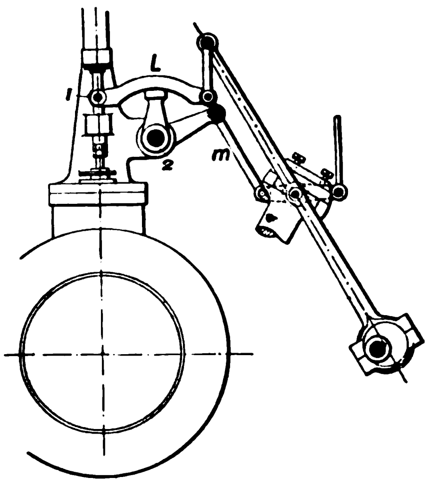


FIG. XI.—15. König Governing Device.

be altogether avoided, it should be so small that it can be easily overcome by means of springs or dash-pots.

3. The position corresponding to the minimum opening of the valve should be very near the position where there would be no opening at all.

4. The rods and operating levers should be of rigid design in order to avoid any vibration.

5. The different parts should be designed so as to afford free access and easy dismantling of the valve.

Several manufacturers have obtained the monopoly of certain designs of gear by duly acquired patents. Naturally the most simple and rational arrangements are the property of those makers who first investigated the question, and it is inevitable that those who have to consider the problem at a later date must be content with more complicated, though probably equally efficient devices, to avoid infringement of the prior rights of the original patentees.

In all designs the mixture inlet valve is operated by means of levers actuated by a cam, or an eccentric fixed on the side shaft. The simplest arrangement uses a single lever; other systems utilise two or sometimes three levers combined with each other to secure the

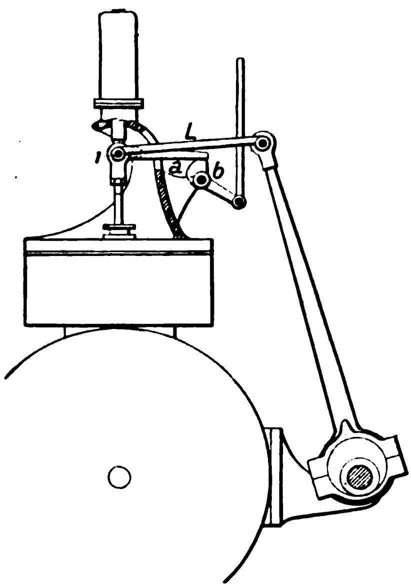


FIG. XI.—16. Gilissen Governing Device.

proper variation of the valve lift. The various systems may be divided into two principal classes :—

(a) Those with a single lever; (b) Those with combined levers.

In order to avoid confusion the movements should be differentiated with respect to the lever which is in immediate contact with the valve spindle. The following definitions of terms will be useful when comparing the various examples noted in the following pages :—

Resistance.—The force to be overcome when moving the valve by movement of its rod in the direction which corresponds with its opening. (The point of application of this resistance must have the variable stroke that meets the predetermined requirements.)

Stress.—The effort exerted more or less directly by the motion of the cam or of the eccentric, so as to vary either the stroke or the point of application of said stress.

Support.—The centre of oscillation of the rocking lever. This support may be moved as well as the lever itself, so as to vary the position of the point of application of the stress as well as the point of application of the resistance.

Single Lever.—The combinations of the single lever arrangement

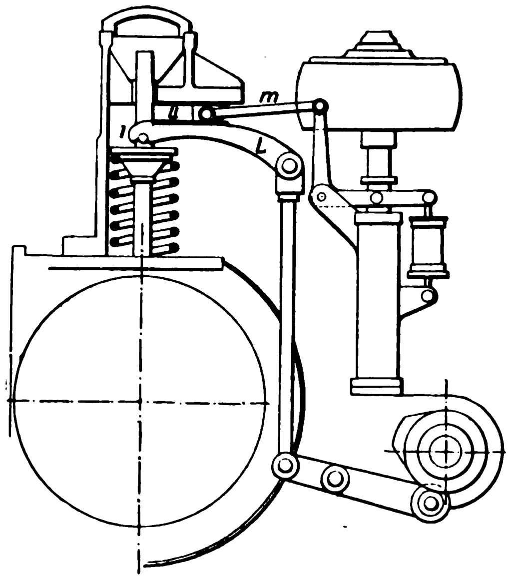


FIG. XI.—17. Güldner Governing Device.

comprise either the displacement of the support, or the variation of the stroke of the point of application of the stress.

Displacement of the Support.—The device employed by the Gasmotoren Fabrik Deutz utilises the displacement of the support, and is illustrated in Fig. XI.—14.

It is protected by a German patent applied for in July, 1902, and granted in May, 1904. It involves a lever *L*, attached at 1 to the spindle of the valve to be operated. The fulcrum of this lever is formed by the end of an arm which oscillates around its suspension point 2, and which is controlled by the governor by means of the rod *m*.

The claim of the patent is as follows:—

"Governing device for inlet valve of explosion engines, characterised by the displacement of the support of the operating lever, effected by the governor according to the load of the engine in such a way that one arm of the lever is elongated, and thereby the other shortened, and consequently the stroke of the valve is varied whilst maintaining a constant duration of the opening."

The author is not entitled to offer any opinion upon the patent

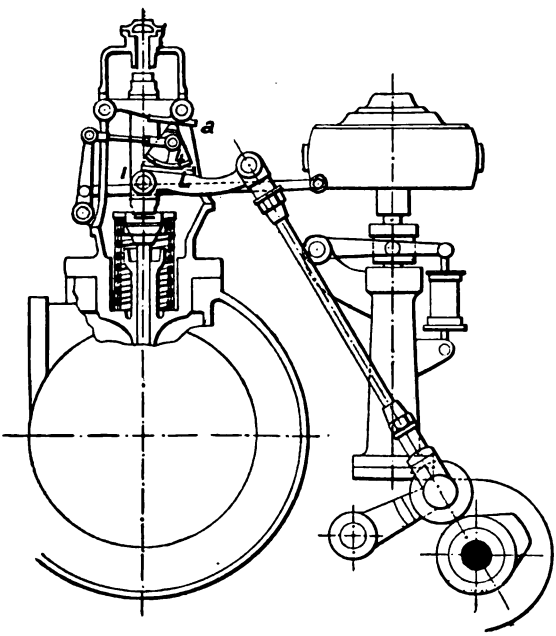


FIG. XI.—18. Bollinckx Governing Device.

rights that the Gasmotoren Fabrik Deutz has acquired in Germany. From the long period which elapsed from the time of application till the time of granting this patent, it is obvious that the German Patent Office thoroughly investigated the matter, and carefully examined the different anticipatory claims which might have been alleged. If, after this examination, the patent has been granted upon the text above mentioned, this constitutes a recognition of the right of the Gasmotoren Fabrik Deutz to benefit by this invention and to prevent any other constructor from making use of mechanism based on similar principles to those which are defined in the patent.

The arrangement under notice may be considered as one of the best yet designed. It is simple, rational, and compact, and in practical use it has worked exceedingly well. But since it is patented in certain countries, other makers will have to content themselves with the use of devices less simple, but perhaps quite as ingenious.

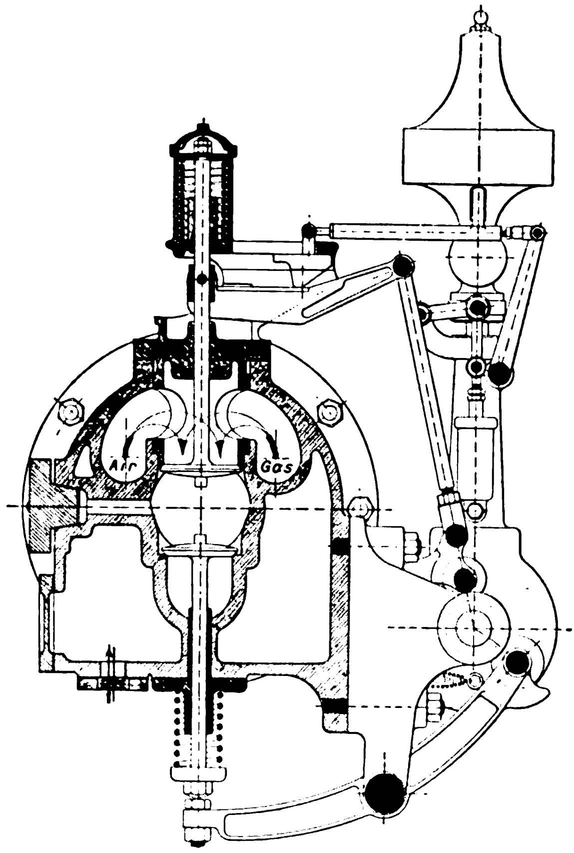


FIG. XI.—19. Mora Governing Device.

It will accordingly be interesting to examine similar devices to the Otto-Deutz, which have been the subject of previous patents, and which have, in consequence, been opposed by the Patent Office against the application for the patent. The Gasmotoren Fabrik Deutz were one of the first, if not the very first, to design gears for giving a variable stroke of the mixture valve of gas engines, and the prior applications deal almost entirely with devices relating to the governing of steam engines. The König and the Gilissen systems may be particularly mentioned in this connection.

In the König system, Fig. XI.—15, the arrangement is absolutely similar to the Otto-Deutz gear, except that the valve is opened by lifting, and that the motion is obtained by an eccentric. In the Gilissen system, Fig. XI.—16, the operating lever *L* rests upon a curved rolling lever *a*, the inclination of which is varied by the governor by means of the cam *b*. The result of this is the variation of the support of the principal lever *L*.

The Güldner system, Fig. XI.—17, used by the Güldner Motoren

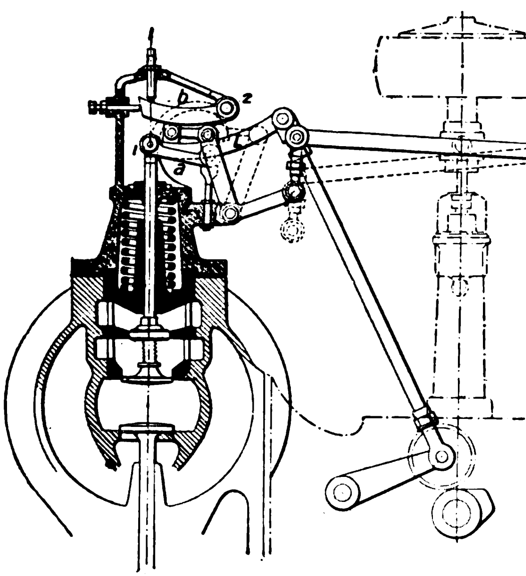


FIG. XI.—20. Anzin Governing Device.

Gesellschaft, differs from the Otto-Deutz gear, by the fact that the support is formed by a block with a partial rolling surface, 4. This block is moved along the straight sliding surface by means of the rod *m*, operated by the governor. This device infringes the claims of the Gasmotoren Fabrik Deutz, but they have permitted Messrs. Güldner to make use of it.

A Belgian patent has been obtained for the gear shown in Fig. XI.—18, by M. Bollinckx, of Brussels. In this the support is replaced by a sector 4, the circular edge of which rolls on the lever *L*, whilst the centre slides against the straight rolling surface *a* and is fulcrumed to a rod operated from the governor.

M. Mora, of Paris, adopts the arrangement shown in Fig. XI.—19, The Société de Mécanique Industrielle d'Anzin (France) has patented an arrangement which is an improvement on the Güldner patent, Fig. XI.—20, and includes a curved rolling surface *b*, instead of a straight one, fulcrumed at 2, and held firmly in position by two set-screws, which provide for hand adjustment. The lever *L* is not fixed to the valve, but rests upon the spindle, and is guided by a secondary lever *a*.

Variable Stroke of the Point of Application of the Stress.—M. Bollinckx proposes the arrangement shown in Fig. XI.—21. The arm *L* of

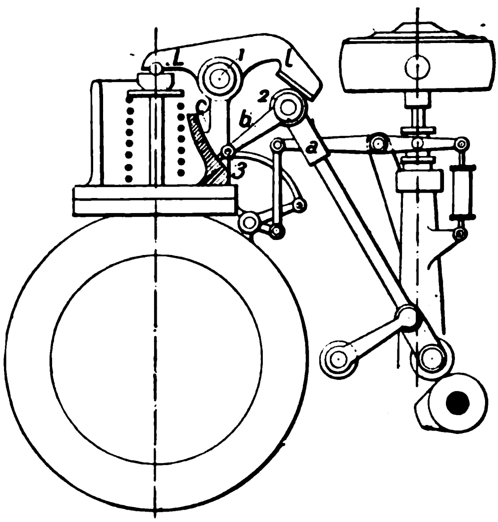


FIG. XI.—21. Bollinckx Governing Device.

the lever *Ll* is in contact with the rod *a*, and is kept in position by the connecting rod *b*, fulcrumed at 2; the end 3 of this connecting rod is moved on a curved rolling surface *C*. According to the position of the point 3 along the surface *C*, the stroke of the point 2, resulting from the constant movement of the rod *a*, is varied, and, owing to the reaction of the lever *Ll*, the stroke of the valve is also made to vary.

Combined Levers.—In making use of two or more levers, the variable stroke of the valve can be obtained by means of various combinations :—

1. The variable stroke of the point of application of the stress.

2. The variable stroke of this point combined with its own displacement or with the displacement of the point of application of the resistance.

3. The displacement of the point of application of the resistance.

(1) Variable Stroke of the Point of Application of the Stress.—In this class is included the arrangement formerly used by the Nürnberg

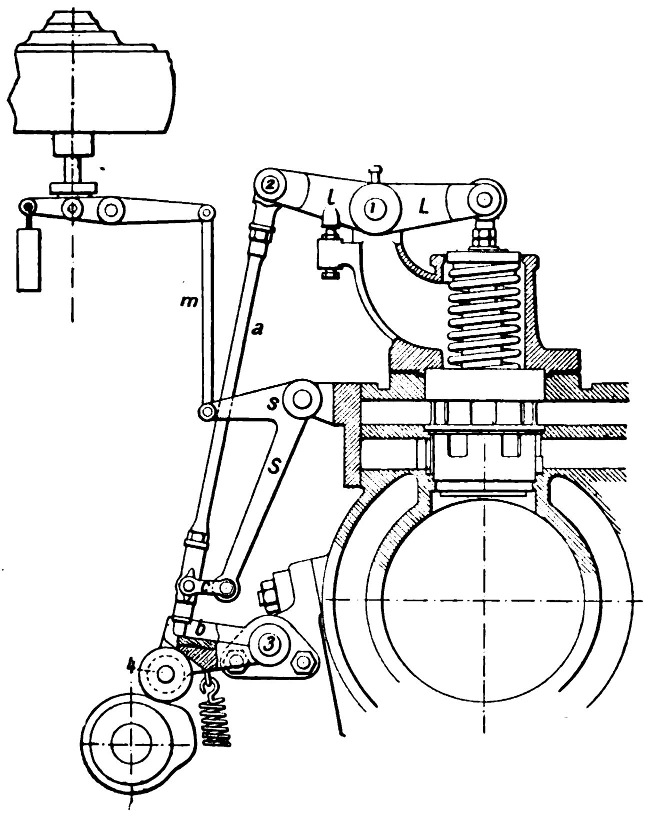


FIG. XI.—22. Otto (French) Governing Device.

Co., similar to that used by the French Otto Co., Fig. XI.—22. The main lever *Ll* is fulcrumed at 2 with a rod *a*, resting at its lower end on a curved link *b*.

One of the ends of this link is fulcrumed to the frame at 3, whilst the free end 4 rests on the operating cam. The governor moves the rod *a* along the link by means of the bell crank lever *S*, the connecting rod *C*, and the governor rod *m*.

Accordingly, as the position of the rod *a* varies on the link *b*, the stroke of the valve is caused to vary from *nil* to maximum.

It should be mentioned, however, that this arrangement has been practically abandoned by the original users, because of its defective working due to the short connecting rod (C), causing prejudicial reactions on the governor. It has now been superseded by a recently patented device.

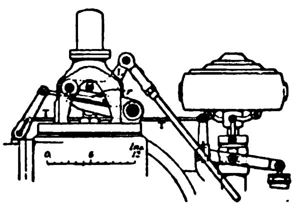


FIG. XI.—23. Nürnberg Governing Device.

(2) **Variable Stroke of the Point of Application of the Stress Combined with its Displacement.**—In this class are the well-known and standard devices of Nürnberg, Fig. XI.—23, and of Schmitz, Fig. XI.—24. The two designs differ from each other only by the position of the levers and their mode of operation. They each contain a secondary lever, which rests upon the

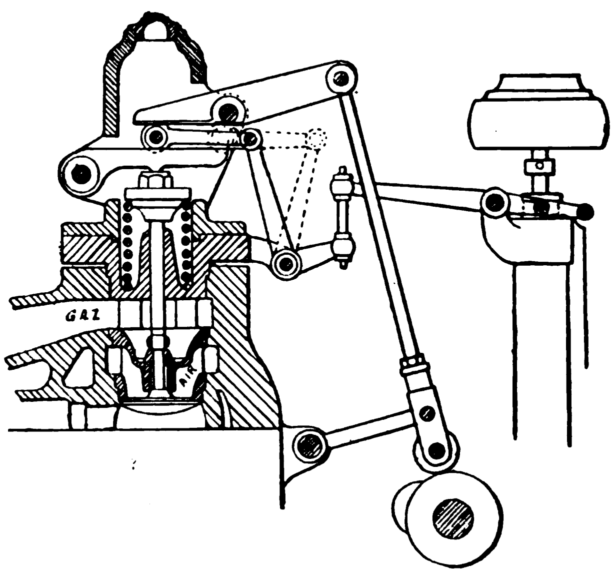


FIG. XI.—24. Schmitz Governing Device.

valve spindle, as well as a principal lever, the arm of which is parallel to the secondary lever. Contact between the two levers is ensured by means of a block or a roller, which moves under the action of the governor. As the block, or roller, moves more or less, so it causes the valve to open from *nil* to maximum.

(3) Variable Stroke of the Point of Application of the Stress Combined with the Displacement of the Point of Application of the Resistance.—The combination, Fig. XI.—25, patented by Tangyes Ltd., and J. Robson, of Birmingham, comprises a principal lever *Ll*, resting at 2 on a secondary lever *b*. The latter rests at 5 on the spindle of the valve. The rod *m* of the governor is fulcrumed at 4 to the lever *b*,

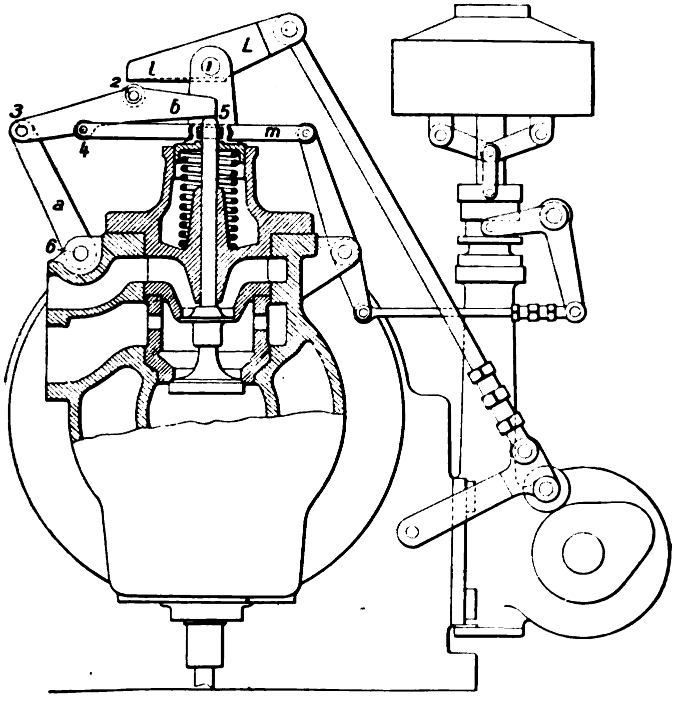


FIG. XI.—25. Tangye Governing Device.

so as to move the latter upon the pivot 6, to which is fulcrumed the connecting rod *a*.

This motion varies the point of application of the stress 2, and at the same time alters the position of the resistance 5, with respect to the point of application 3.

Displacement of the Point of Application of the Resistance.—Some combinations of this class have been recently constructed. They are too recent in application to enable any judgment to be formed of their reliability in actual work.

Mixing Valve.—In considering the hypothesis of governing with constant ratio and variable volume, the use of the "mixing valve" must be considered. This valve is, as a rule, composed of two parts; one is an ordinary double-seated equilibrium valve, which closes the annular gas ports; the other is formed with a circular slide that moves over the air ports, which are, in almost all cases, placed below the ring of gas ports.

In the design used by the "Schweizerische Locomotiv und Mas-

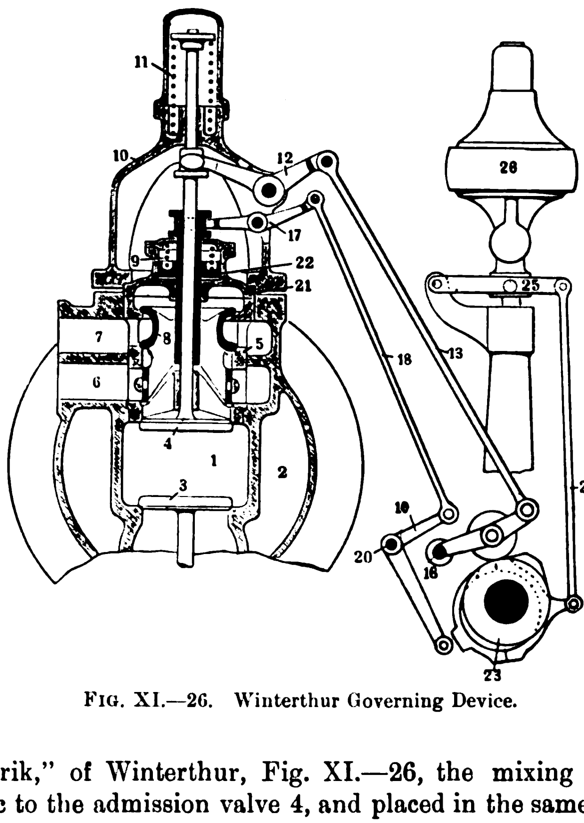


FIG. XI.—26. Winterthur Governing Device.

chinenfabrik," of Winterthur, Fig. XI.—26, the mixing valve 8 is concentric to the admission valve 4, and placed in the same box, which can easily be taken to pieces for cleaning. The main inlet valve, kept on its seat by the spring 11, is operated by the ordinary combination of lever, rod and cam, so as to keep the valve open during the full suction stroke. The mixture valve 8 consists of a double-seated gas-valve and a circular air slide, made in one piece. This valve is operated from outside through a sleeve fulcrumed to another combination of lever 17, rod 18 and cam, with the difference that the latter is part of an eccentric 23, the position of which can be altered by the governor 26, through the rod 24.

Owing to the special shape of the cam carried by the eccentric, the mixture valve opens always at the same instant, whatever the position of the governor, but closes earlier or later during the piston stroke according to the power developed by the engine. The air dash-pot deadens the shock caused by the rapid closing of the valve by the action of the spring 9.

Fig. XI.—27 illustrates the former arrangement used by the

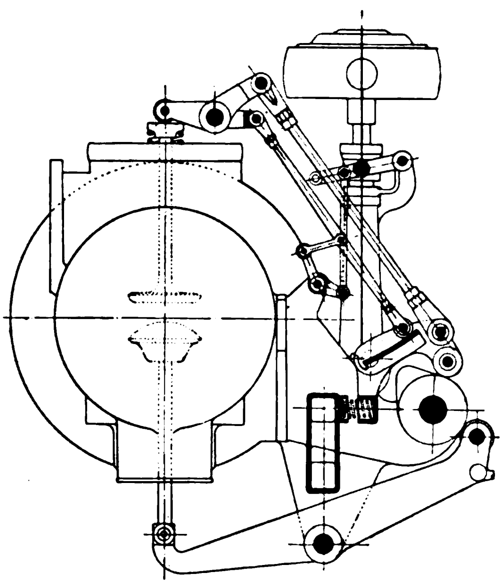


FIG. XI.—27. Nürnberg Governing Device.

Nürnberg Co. in their single-acting engines. The lever operates a mixing valve through a rod, the end of which is moved on a link by the governor. This link is part of the lever which receives a constant motion through the cam and roller. By means of the link a variable opening to the mixing valve is obtained.

Governing at Variable Ratio and Constant Volume.—The mechanical devices designed to secure the second system of governing with variable ratio and constant volume by operating upon the valve stroke of the gas valve will now be reviewed.

If the gas valve be separately considered, irrespective of the admission valve to the cylinder, the different combinations may be divided into systems with single lever, and systems with multiple levers just as for the third system of governing. In practice, the mechanism of the gas valve is always combined with the inlet valve gear, and the arrangement consequently involves the use of several levers.

These combinations may be classified as follows :—

1. The displacement of the support of the operating lever.
2. The variable stroke of the point of application of the stress.
3. The displacement of the point of application of the resistance.

1. The displacement of the point of application of the support has been applied by the Gasmotoren Fabrik Deutz, and will be discussed later on when dealing with the large gas engines.

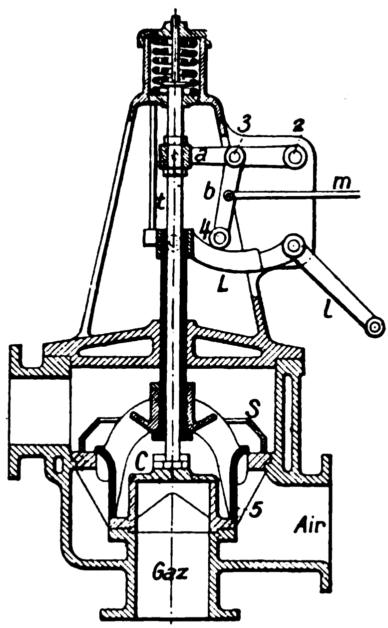


FIG. XI.—28. Fahlencamp Governing Device.

2. The variable stroke of the point of application of the stress is used in the device of Fahlencamp, Fig. XI.—28, whose patent was opposed to the Gasmotoren Fabrik Deutz by the Patent Office, although the two systems are absolutely different. Fahlencamp operates the air-valve 5 by means of a lever *L* at constant stroke. Through the sleeve of this valve passes the spindle of the gas valve. On the curved air-valve lever *L* rests an arm *b*, fulcrumed at the other end to the spindle of the gas valve. The arm *b* is moved by the governor by means of the rod *m*, and so causes the variable opening of the gas valve.

3. The displacement of the point of application of the resistance has been realised in the device of Recke, of Rheydt (Germany). This arrangement, Fig. XI.—29, has a principal lever, *Ll*, connected to the inlet valve, to which it gives a constant motion. To this lever is fulcrumed a connecting rod *b*, fulcrumed to a lever *Aa*. The latter can turn upon the pivot 3 through a movable support displaced by the governor.

The arm a of the lever Aa is so curved that it keeps in constant contact with the roller C, fitted to the end of the gas valve. The stroke of the latter varies in accordance with the displacement of the

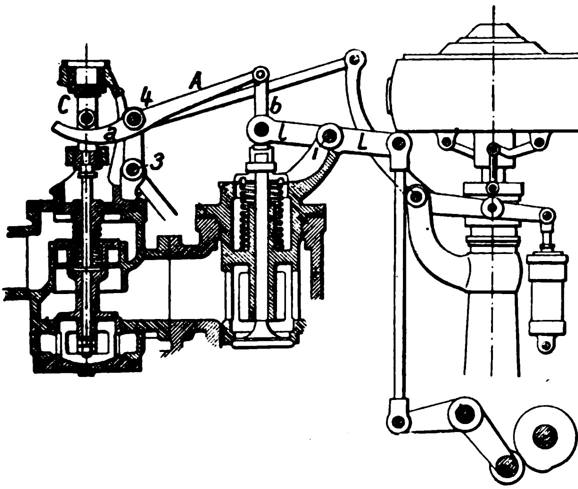


FIG. XI.—29. Recke Governing Device.

lever Aa, as the movable support is brought more or less near the roller.

Governing at Variable Ratio and Volume.—The governing arrangement designed by the Hallesche Maschinenfabrik can simultaneously

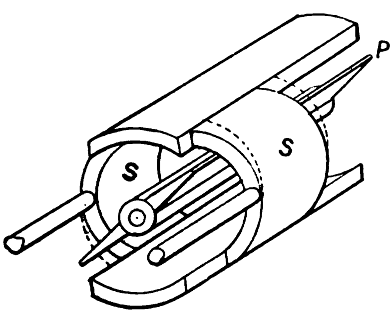


FIG. XI.—30. Hallesche Throttling Device.

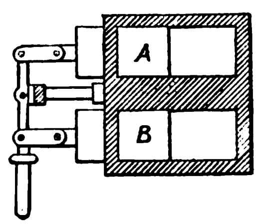


FIG. XI.—31. Hallesche Gas and Air Regulator.

vary the quality and the quantity, and enters, of course, into the fourth class. It consists of a cylindrical box, fitted in the prolongation of the gas and air pipe. In this box is a throttling device S, Fig. XI.—30, containing a butterfly valve *P*, which is under the control of the governor and regulates the quantity of mixture admitted to the cylinder. On opposite sides of the throttle are two movable segments *SS*, which can be moved as shown by the dotted lines. According to the position of the segments, the supply of one or another of the two fluids composing the mixture can be altered so as to modify the ratio. This modification can be effected either by the governor or by hand.

The patent granted to this firm for their governing device also includes an arrangement for separating the gas and the air in two passages, as shown in Fig. XI.—31, and in securing the variation of the mixture by means of two slides *A* and *B*, operated by a lever in such a way that one slide opens when the other closes, and *vice versâ*.

Governing of Large Gas Engines.—From a practical as well as from an experimental point of view, and more particularly when dealing with large engines, it will be noticed that several important firms after having made use of the one system of governing, have abandoned it in favour of another. Other firms have designed a combination of variable volume *and* variable ratio, thus endeavouring to get the good qualities of each method while eliminating the defects. It is interesting to notice how these makers have solved the problem, but at the same time it is open to question whether the combination of the two methods has not been obtained at the cost of great mechanical complication.

As has been already mentioned, considerations of fuel economy have not encouraged improvements in the constructional details of large gas engines with respect to high efficiency on the part of gas engine makers in America and England as much as they have in Germany, France, and Belgium. It is in the latter countries that within the last seven or eight years the use of large engines has been rapidly developed, and the leading engineers have had to deal with the difficult and complex problem of building large internal combustion units, comparable to steam engines in regard to safe working and reliability.

During the four or five years occupied in developing the new ideas English and American makers have watched the laborious though rapid revolution of the blast furnace gas engine; and now, when the principal practical difficulties have been solved, they are able to make a selection from the best Continental designs and commence construction of such engines under licence. In their turn they have endeavoured to simplify the design, and to incorporate such further improvements which are supposed to be best suited to local requirements.

British and American engineers have always opposed the tendency of European makers to complicate the mechanical arrangements, and in consequence, we may perhaps expect that in the very near future the Continental designs will be considerably simplified. The author has always insisted upon the simplification of the designs of the Continental engineers, and when, in December, 1906, Dr. Lucke, Professor Fernald, and Mr. Junge read their interesting papers at the annual meeting of the American Society of Mechanical Engineers in New York, he took advantage of the discussions to set forth his opinions in this respect.

The author must, however, take exception to certain statements that have since been made, namely, as to the necessity of building gas engines after a so-called "American practice." He is of the opinion that in the future there will not be any "American practice" nor will there be an "English" or a "German," but there will be a universal practice, and one that will be based upon the ripest experience and the most scientific principles. It will be the best practice suggested by pioneers like the German and Belgian engineers. With the exception of certain constructional details and rules with which American engineers seem at variance, notwithstanding the fact that these principles have been evolved by European makers as the result of long experience, there is no doubt that the Continental engine will be improved when slightly Americanised. That, perhaps, will be the moment for European makers to derive some benefit from American methods, and they may be permitted to do so without any prejudice to the honour due to them as the originators and pioneers.

The devices that have been already described and commented upon are more especially applicable to small single-acting engines of powers up to 100—125 H.P. per cylinder. Although they have the merit of great constructional simplicity, they cannot, in many cases, be applied to large double-acting engines, because of the weight of the moving parts, the inertia of which would be detrimental to the sensitiveness of the governor. Moreover, the partial vacuum caused in the cylinder at low loads, which may reach from 7 to 10 lbs., causes negative work which has to be considered when dealing with large engines. The result is not only the loss of useful work, which affects the mechanical efficiency, but also the tendency of the admission and exhaust valves to open, due to the suction. In a single-acting engine of 150 H.P. these valves have an area of about 65 square inches, which, with a vacuum of 7 lbs., would entail a suction effort of about 455 lbs. upon the valve. To ensure tightness, a tension three times as great as that needed for keeping the valves on their seat is used for these springs, because, in addition to the tendency of such springs to yield to the action of the vacuum, they have to overcome the frictional resistance of the guides. In large engines, where the valves reach 15 inches in diameter and are subject to a suction effort of about 1,250 lbs., these springs offer, of course, an enormous resistance to the motion of the engine.

In spite of certain complications of mechanical parts, which cannot be avoided, makers have endeavoured to replace the negative action caused by springs on the valves with mechanical devices that maintain the valves tight on their seats, a spring only being used to compensate the play in the fulcrums. This spring may, of course, be light and small, since it is not used to draw the valve back on its seat. Hence, the triple function of opening, closing, and maintaining the valve on its seat is entirely dependent upon a positively actuated mechanical device obtained by means of connecting rods operated from the side-shaft of the engine. But as a consequence, and with very few exceptions, recourse has been made to eccentrics instead of cams. In order to assure the sensitiveness and independence of the governor, an automatic cut-off has been applied in many cases. In steam engines of the automatic cut-off type, the function of the governor is but that of an apparatus which will show, in a passive way, the moment where admission must start or stop, without itself operating these functions. The active forces are obtained by the operating device.

For certain special services, such as operating rolling mills, hoisting machinery, blowing engines, &c., the engines must be built so that their speed can be varied. For instance, in the case of engines directly coupled to large blast furnace blowers the number of revolutions must be capable of variation from half to full.

In such conditions, engines running at a low normal speed are best suited to the purpose, because the inertia of the moving parts is less prejudicial to smooth working and there is no need to adopt constant compression for the sake of balancing the moving parts. In the latter case, of course, the governing with constant ratio and variable volume seems to be the most suitable owing to the simplicity of the mechanical devices and to the high efficiency at low loads, while ignition faults occur less frequently with this system.

The gears applied by the leading Continental makers for governing large gas engines will be reviewed and described in the order which corresponds, as far as possible, to the classification previously adopted.

(a) Method with constant ratio, variable volume (3rd category).

EXAMPLES.

1. Gasmotoren Fabrik Deutz.
2. Société John Cockerill.
3. Gutehoffnungshütte.
4. Elsassische Maschinenbau.
5. Dinglersche Maschinenfabrik.
6. Gebrüder Koerting Actiengesellschaft.
7. Société Française de Constructions Mécaniques.
8. Snow Steam Pump Works.

(b) Method with variable ratio and constant volume (2nd category).

9. Maschinenfabrik Augsburg, Nürnberg.
10. Crossley Brothers, Ltd.
11. Société John Cockerill.
12. Ehrhardt and Sehmer.

(c) Combined method.

13. Thyssen & Co., of Mulheim.

(d) Method with constant ratio and constant compression.

14. Schuchtermann & Kremer.

(a) Method with Constant Ratio and Variable Volume.

(1) *Gasmotoren Fabrik Deutz—Double-acting Engine.*—In their double-acting engine, the Otto-Deutz Co. has abandoned the use of the gas and air valves fitted concentrically to the admission valve. They consider that this arrangement involves excessively heavy working parts, and that for the sake of easy dismantling and cleaning it is better to have the mixing valve, which is the most liable to get dirty, fitted in a separate box.

This is common to the three systems of governing here described.

In Figs. XI.—32 and 34, the motion of the admission gear is obtained by the cam fitted on the side shaft by means of rods and levers to the main valve. The mixing valve has a double seat, the upper one used for the air and the lower one for the gas, whilst in the air pipe is a butterfly, which is operated by hand, and in the gas pipe a hand cock, Fig. XI.—32, the latter being replaced by a valve in the device shown, Fig. XI.—34.

In both cases the mixing valve is operated by the rod used for the motion of the admission valve, in such a way that when the latter drops the former rises, and its lift is variable according to the position given by the governor to the movable support, which moves between the two levers, forming the rolling path.

It is this variable stroke, always beginning and ending at the same points, that assures the admission in variable quantity of a predetermined mixture.

The double seat of the valve has also the effect of balancing it and of opposing the minimum resistance to the governor, whatever the

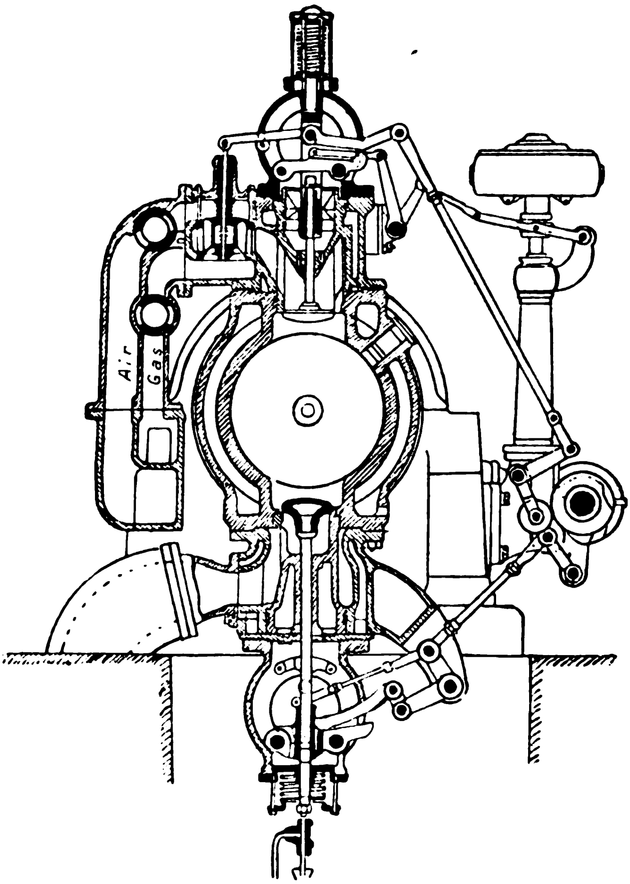


FIG. XI.—32. Governing Device, Otto-Deutz double-acting Engines.

amount of the vacuum existing between this valve and the main inlet valve.

In the device, Fig. XI.—33, a special inlet for town gas is also provided under the mixing valve. This inlet is controlled by a cock, to be used, of course, when the fuel gas inlet is closed.

The mixtures obtained with town gas must be more correctly proportioned, and in order to avoid leakages, which easily take place with double-seated valves, a special town gas valve has been provided with a spring which keeps it tight upon its seat. The construction of the device shown in Fig. XI.—34, differs from Fig. XI.—33, by the operating mechanism of the main inlet valve, in which the strong opposing coil spring is no longer used, but is replaced by a set of levers which draw back the valve on its seat by means of a spring

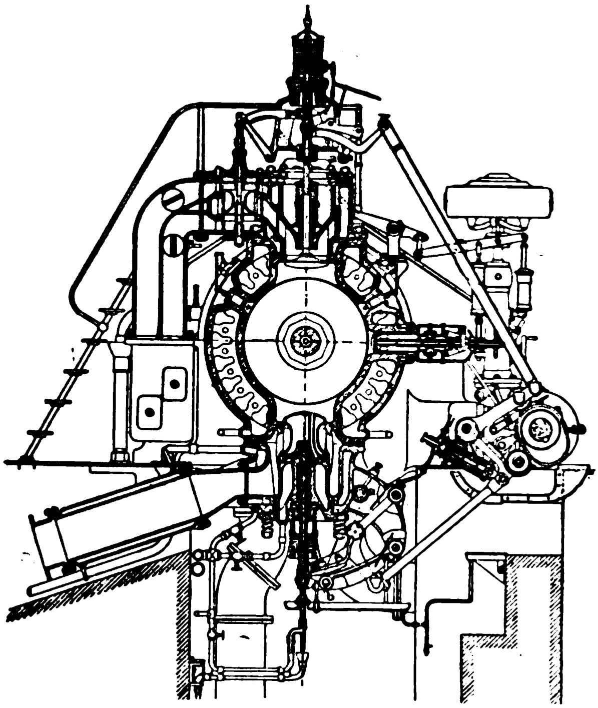


FIG. XI.—33. Governing Device, Otto-Deutz double-acting Engines.

fitted to the lower lever of the operating rod. As soon as this valve is brought back on its seat it is mechanically retained by the device of levers and fulcrums in the following manner :—

The main crank lever, Fig. XI.—34, the pivot of which is supported on the casting, is connected at one end by two small connecting rods to the spindle of the valve, and at the other end to the operating rod by means of two other short connecting rods. At rest, the latter are fitted to follow the same straight line and to constitute a toggle joint arrangement that prevents the valve opening when it is submitted to the action of the suction during the forward stroke. As soon as the main rod enters in operation, this straight line is broken, causing a reaction that makes the valve open.

At the end of the valve there is only a small spring used for compensating the play in the fulcrums and pivots.

A circular slide is also shown in Fig. XI.—34, on the spindle of the

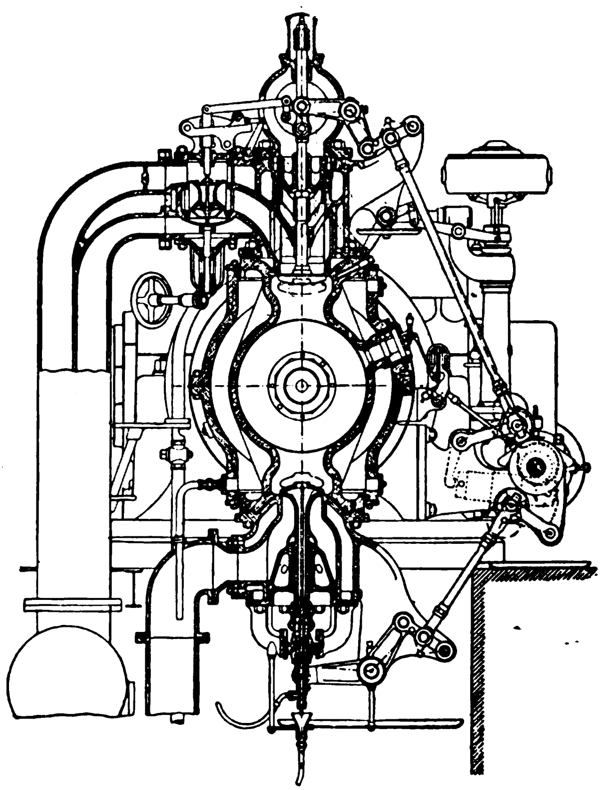


FIG. XI.—34. Governing Device, Otto-Deutz double-acting Engines.

inlet valve, between two guides. This is used either for making an additional independent air inlet, in order to impoverish the mixture when using rich gas or for scavenging the mixture chamber by admitting air after the mixing valve is closed at the outer dead centre of the piston, or when the exhaust valve is still kept open after the inner dead centre.

The third device applied by the Gasmotoren Fabrik Deutz to their very large machines (for instance, 2,000 H.P.) presents some very interesting and entirely novel features. These are illustrated in Figs. XI.—35, 36, 37, and 38. The main inlet valve, which, instead of being opened in the usual way by a cam on the side shaft, is, on the contrary, retained closed under the action of this cam, presents a development three times larger than the normal cam. The usual operating lever is fulcrumed to the spindle of the valve by means of two small connecting rods, a small coil spring absorbing the

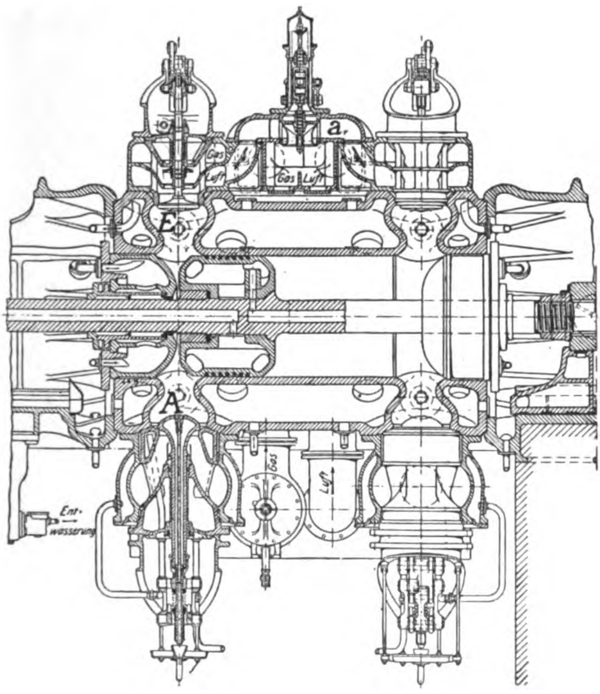


FIG. XI.—35. Governing Device, Otto-Deutz double-acting Engines.

play. As soon as the cam releases the motion and allows the operating rod to drop, a strong coil spring, attached to the operating lever, and which was compressed during the whole period the valve was closed, relaxes its tension and opens the valve during the whole period corresponding to the continuity of the cam. This arrangement ensures the positive mechanical locking of the valve by preventing it from opening under the action of the suction.

The lift or stroke of the valve determined by the strong coil spring is invariable.

Another novelty consists in the use of a single set of gas and air valves which are common to the two inlet valves, each fitted at the end of the cylinder. This arrangement reduces the number of the operating mechanical parts by one half, and has the further advantage of supplying an identical mixture to both ends of the cylinder.

It must, however, be remembered that the length of the passage

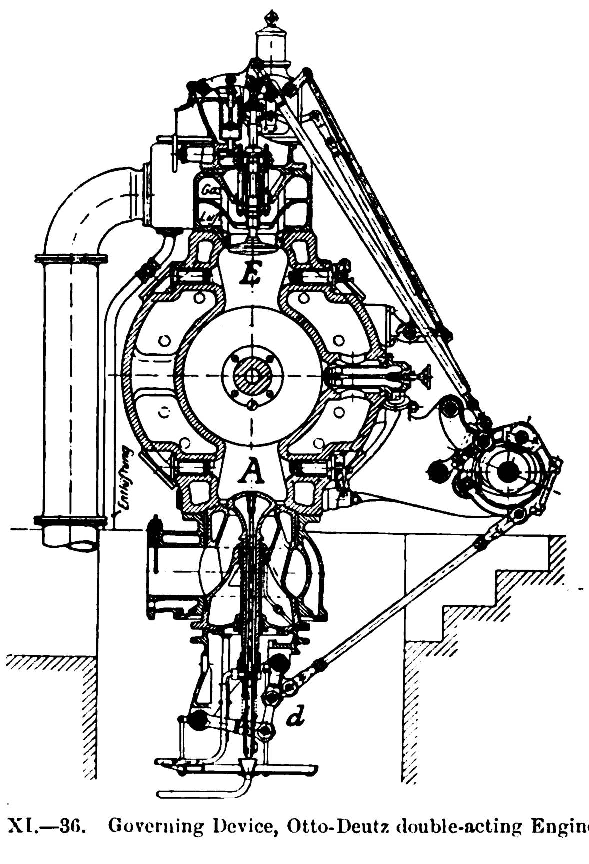


FIG. XI.—36. Governing Device, Otto-Deutz double-acting Engines.

from the mixing valves to the admission valves creates a dead space, the contents of which may partially escape the action of the governor in such a way that the governor would not control the speed as closely and as quickly as with valves fitted concentrically, according to the practice of many other makers.

The gas and air valves are fitted in a cage located between the two admission valves, in the middle of the cylinder, and in its transverse axis. The variable admission of both air and gas is obtained by means of a device with variable fulcrum giving a variable valve lift as used in the other controlling gears of the Gasmotoren Fabrik Deutz, Figs. XI.—37 and 38. The air valve carries the rocking lever operated by the rod and the cam on the side shaft, and the governor acts directly upon this valve, which is connected to the gas valve by a similar set of levers in such a way that both valves are dependent upon each other in their motion. The two valves drop for the admission, and, as they are of the double-seated type, they open at

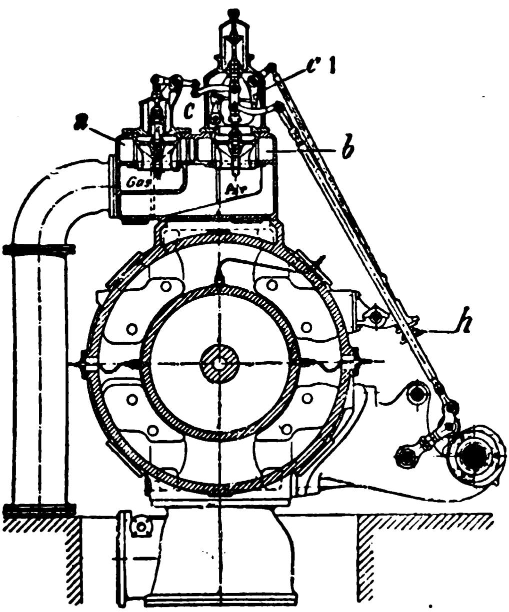


FIG. XI.—37. Governing Device, Otto-Deutz double-acting Engines.

both top and bottom. The channels for the air and for the gas both end in the shape of tuyeres, immediately behind the admission valves where the mixing takes place.

The gas inlet can be increased or diminished by means of a circular disc, fitted to a sleeve, moving along the spindle of the inlet valve. This can be operated by hand to regulate the quantity of gas, according to its quality. The air supply can also be regulated by means of butterfly valves fitted in the supply channel.

(2) *Société John Cockerill, Seraing.*—The system of governing at constant ratio used by this firm consists of a main inlet valve with

invariable stroke, which is kept on its seat by means of the usual coil spring. On a sleeve, moving along the spindle of this valve, a double-seated balanced valve and a circular slide are fitted. The balanced valve controls the gas inlet at the top, and the slide controls the air

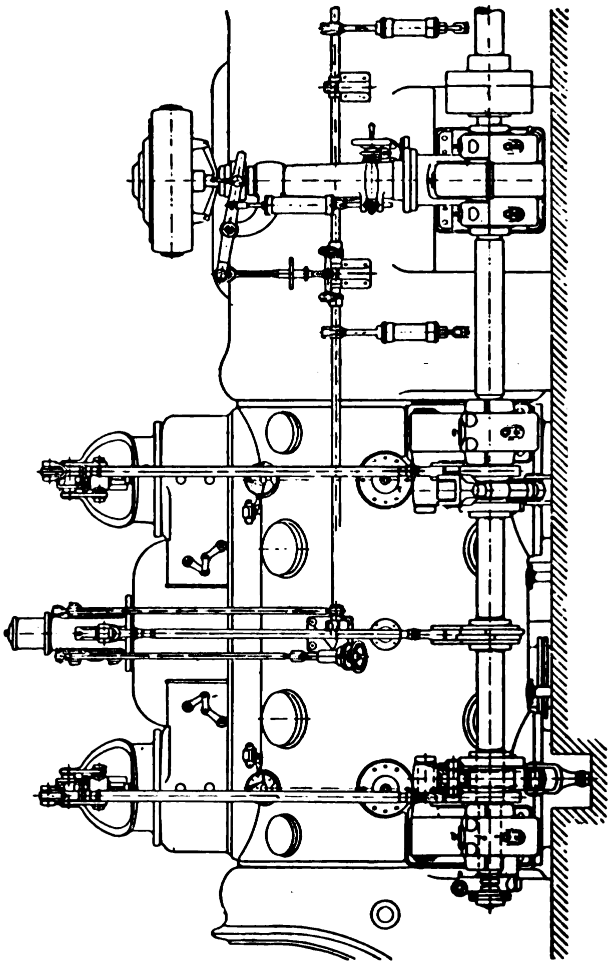


Fig. XI.—38. Governing Mechanism, Otto-Deutz double-acting Engines.

inlet at the bottom. The main inlet valve is moved by the usual system of cam, rod, and lever, shown in Fig. XI.—39.

The gas valve and the air slide are fitted upon a sleeve joined to a rod L which is provided with a stop at its free end. Against this stop rests the push rod P, which slightly turns on the free end of the lever A, which, in its turn, receives an oscillating movement by means of a set of levers from the engine cam shaft.

To support the upper ends of these two levers, the lever B is fitted, terminating in a small roller, the position of which is determined by the governor.

When the lever A is drawn downwards by the operating rod, the lever L participates in the movement and lifts the gas and air valve

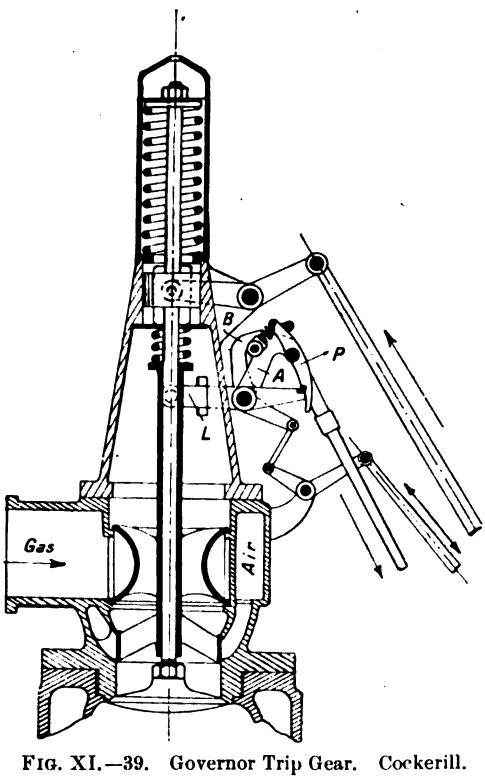


FIG. XI.—39. Governor Trip Gear. Cockerill.

cages until that portion of the stroke is reached, when the contact between the roller B, and the end of the push rod P, releases the catch points and allows the coil spring to act, thus shutting off the admission of mixture by movement of the valve cage sleeve sooner or later under the control of the governor. The mechanism is similar to that represented in Fig. VIII.—12 (Oechelhäuser).

This device has been abandoned recently by the Cockerill Co., and they now govern all their engines with variable ratio and constant volume.

(3) *Gutehoffnungshütte, of Oberhausen.*—The inlet valve shown in Fig. XI.—40 is of the ordinary type, closed and maintained on its seat by a powerful coil spring and opened by the usual device, consisting of lever, rod, and cam. Two rolling levers secure a smooth motion for the constant stroke of this valve. The mixing device is located in a side box in which two slide valves are fitted on the same rod, one for gas and the other for air. Their simultaneous lift determines the

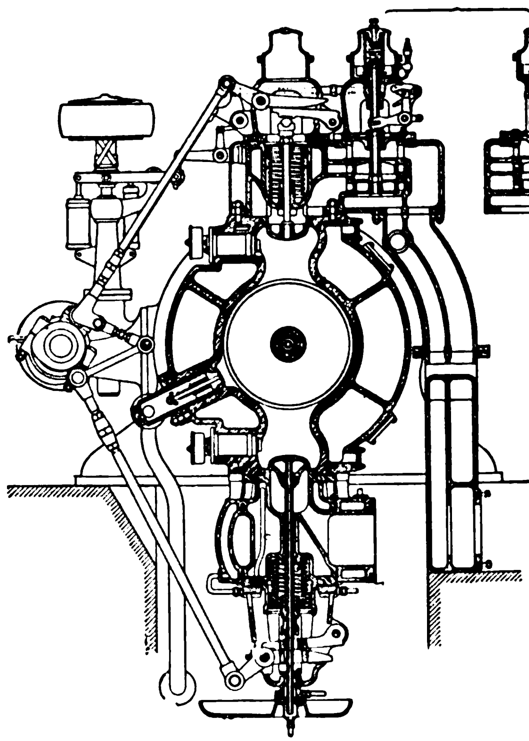


FIG. XI.—40. Gutehoffnungshütte Governing Device.

mixture at constant ratio; the latter can be regulated by hand, while the total quantity admitted to the cylinder is varied by the duration of the opening.

The mixture slide valves are opened by eccentrics fitted on the side shaft through a set of rods and trip gears controlled by the governor, so as to cut off the inlet of the mixture, earlier or later, along the suction stroke. When the cut-off takes place, the slide valves are suddenly closed under the action of the spring placed within an air dash-pot to ensure smooth operation.

(4) *Elsassische Maschinenbau of Mulhausen.*—This company, which started the construction of large gas engines under the licence of the Cockerill Co., now make engines of their own design, in which the controlling gear is of a different character. This is shown in Figs. XI.—41 and 42. They have adopted eccentrics for the valve motion. The inlet valve is opened by means of a combination of levers, one being fulcrumed to the eccentric rod, the other to the valve spindle. Both levers are supported by pivots carried by the casting, and the combination assures the smooth operation of the

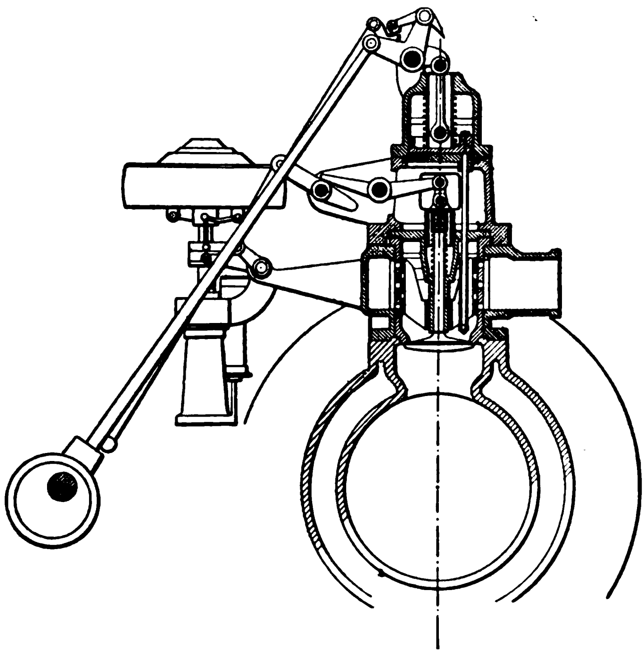


FIG. XI.—41. Elsassische Governing Device.

rolling path. The admission valve has a constant opening, and is brought back and kept on its seat by the coil spring fitted on its spindle. A cylindrical mixing slide with multiple ports is concentrically fitted to the main valve, and guided by means of an independent sleeve.

The cylindrical slide is connected by means of two vertical rods to the air dash-pot fitted at the top of the casing, and is operated by a trip motion. The channel which surrounds the mixing slides, contains gas in one half and air in the other. Fig. XI.—42 shows a horizontal and vertical cross sections of the valve box.

To obtain the admission of a variable quantity of predetermined mixture, the trip raises the mixing slide and opens it. At the same time the admission valve is dropped, admitting the mixture, whilst the governor causes the cut-off of the mixture to take place along the suction stroke. The effect of the cut-off is to permit the mixture slide to close by the action of the spring and dash-pot. The slide then suddenly and smoothly stops the supply of air and gas, whilst a corresponding partial vacuum takes place in the cylinder. This combination is very simple indeed, and secures even velocities to the streams of both air and gas.

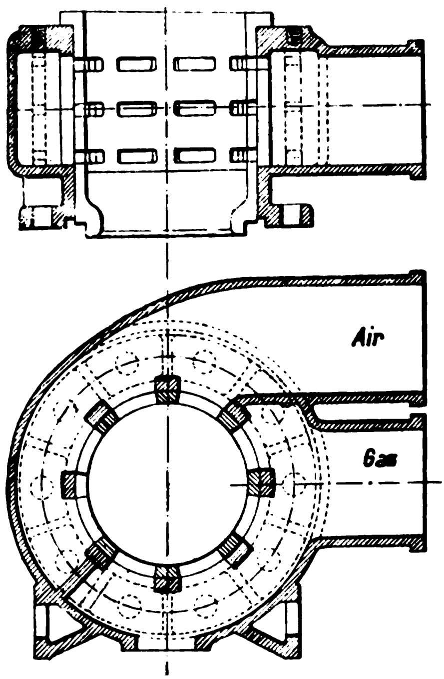


FIG. XI.—42. Elsassische Mixing Valve Box.

As an inlet gear for the gas, however, the slide is subjected to the inconvenience of sticking or of grinding the rubbing surfaces if the gas is not well freed from tar or dust. The makers state that in practical work no trouble of this sort has occurred; but it must be borne in mind that the engines made by the firm are almost all intended for use with blast furnace gas or coke oven gas, which are thoroughly washed and cleansed, and contain imperceptible traces of tar, and less than 10 grains of dust per 1,000 cubic feet.

(5) *Dinglersche Maschinenfabrik, Zweibrücken.*—This gear, illustrated in Fig. XI.—43, presents some distinctive features, and has two side

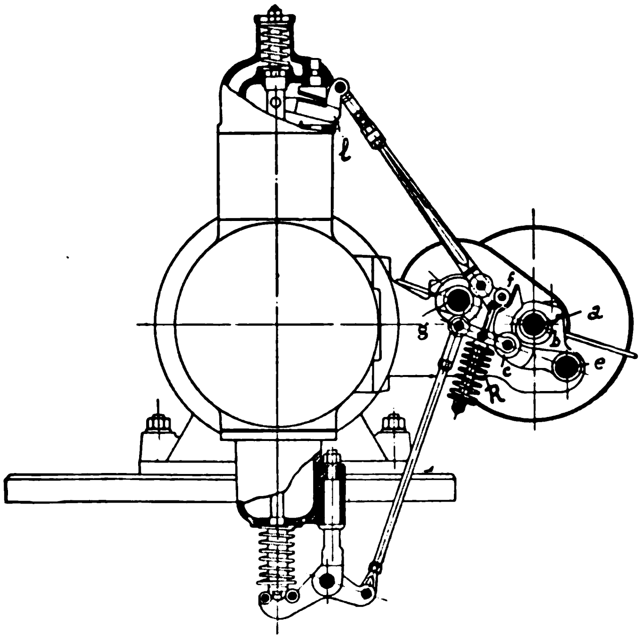


FIG. XI.—43. Dingler Governing Device.

shafts for its operation; the one (a) is an auxiliary shaft, running at the same number of revolutions as the crank shaft, and carrying an axial governor; the other (g) is the usual half-speed cam shaft. The

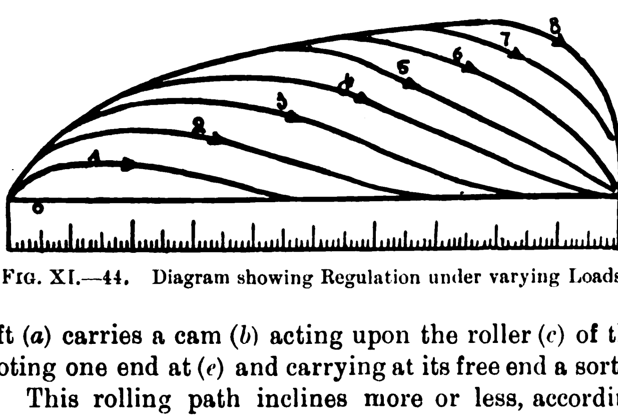


FIG. XI.—44. Diagram showing Regulation under varying Loads.

first shaft (a) carries a cam (b) acting upon the roller (c) of the curved lever pivoting one end at (e) and carrying at its free end a sort of rolling path (f). This rolling path inclines more or less, according to the movement of the governor cam (b). The main inlet valve, with the circular air and gas slide which it carries, is lowered by the usual device of a rolling path (l) with rod and cam operated from the half-speed side shaft. Hence, the motion transmitted to the device is longer or shorter in consequence of the reaction of the roller of the operating rod when the cam raises it, whilst it is kept on the movable rolling path by the spring (R). The roller of the operating rod moves concentrically or eccentrically with respect to the pivot (e) of the rolling path lever.

The makers illustrate the effect obtained on the admission by the diagram, Fig. XI.—44, in which the curves illustrate the different

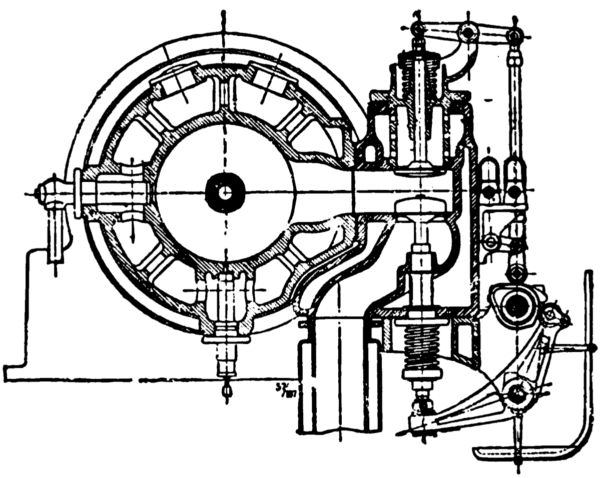


FIG. XI.—45. Koerting double-acting, four-cycle, Engine.

admissions, and show that the strokes and the duration of admission increase in proportion to the load under the control of the governor.

The air and gas slide being part of the admission valve, the mixture remains at a constant ratio, and the volume admitted is variable.

(6) *Gebrüder Koerting, of Hanover.*—Four-cycle engine. The new Koerting double-acting Otto cycle engine, Fig. XI.—45, has been recently introduced to meet the demand for engines of this class, rather than for the two-cycle type. The design of gear applied with such advantage to the single-acting Koerting engine has been retained. It consists of a butterfly valve placed in the passage between the mixing valve and the main admission valve. Instead of being arranged with the combustion chambers at the back of the cylinder, as in double-

acting engines, the new construction has these chambers fitted laterally.

The side shaft operates the main inlet valve with a constant motion obtained from a cam. This valve is located in a box provided with a series of ports for the introduction of the mixture. This box is connected with an automatic flat valve, common to both the air and gas ports. The mixture is of constant ratio, and is admitted in a

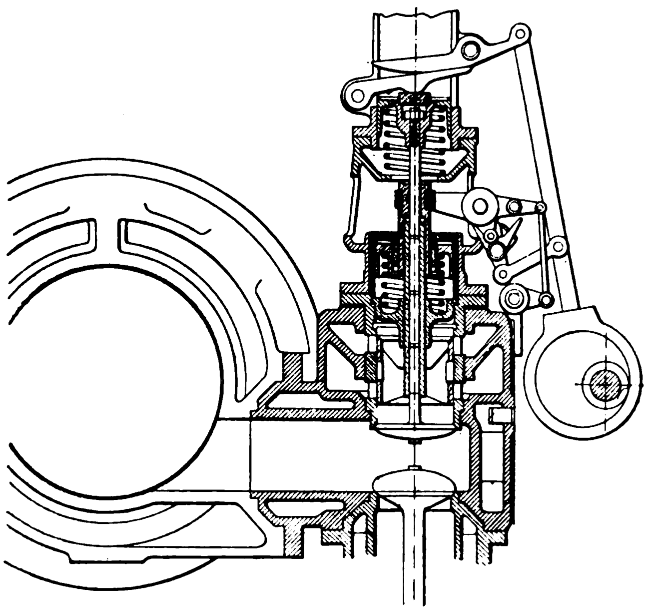


FIG. XI.—46. Denain Governing Device.

variable quantity by the butterfly valve operated by the governor. See Fig. XI.—11.

(7) *Société Française de Constructions Mécaniques, Denain.*—This construction, Fig. XI.—46, has also the inlet and the exhaust valves arranged at the side of the cylinder, but in this case the valve box is on a level with the bottom of the cylinder. The inlet valve is operated by a set of levers, rolling path, and eccentric, giving an invariable stroke. In the box behind the inlet valve the gas ports are arranged, and above them, the air ports. A cylindrical slide opens simultaneously both series of ports, so that air and gas enter at the same time and form a mixture at constant ratio behind the inlet valve. The slide is quite independent of the valve, and is connected a

by a sleeve to an outer lever. The latter is rendered either dependent or otherwise upon the rod of the eccentric that operates the inlet valve, by means of a trip motion under the control of the governor. The mixing slide uncovers the air and gas ports at the same time as the inlet valve opens—*i.e.*, at the beginning of the suction stroke. Along the latter, and according to the load on the engine, the trip motion releases the slide to the action of its coil spring, causing a sudden cut-off of the supply, whilst the shock is broken by the air dash-pot in combination with the spring. The slide is then brought back as in Fig. XI.—46. The vacuum that follows does not affect the slide, and in the absence of any tendency to be re-opened, its spring is very weak, and has the least possible influence on the sensitiveness of the governor.

However, the reliability of the device depends on the tightness of the slide and on its intimate contact with the whole surface of the box provided with the ports. This surface is continuously exposed to dust and tar carried by the gas, and is, moreover, difficult to lubricate efficiently. The slide would in this case be liable to stick and to resist the action of the light spring, with the result that the engine would "hunt." The device, therefore, would answer better with well-purified and washed gas, such as coke oven or blast furnace gas, than with suction producer gas. The position of the valve box involves rather a long channel, which is detrimental to the quick propagation of the flame. In consequence, and in order to avoid very early firing, a second ignition device has to be fitted in the combustion chamber. It is exceedingly difficult to obtain a short and evenly shaped combustion chamber when placed laterally, so that this arrangement must be taken with its inconveniences and advantages. The Société de Constructions Mecaniques of Denain, when designing their engine, seem to have realised the practical and valuable qualities of simplicity and facility of access.

(8) *Snow Steam Pump Works, Buffalo.*—This firm have employed a new governing arrangement acting upon the volume of mixture in their new four-cylinder, double-acting, twin-tandem engine, installed in 1908 at Ceres, New York, for the Western New York and Pennsylvania Traction Co.

Fig. XI.—47 shows a longitudinal section and end view of the mixture chamber with the valves and mechanism, and is taken from an article published in *Power and the Engineer* of New York.

The governor controls the action of two trip valves, one for air and the other for gas. The arrangement of the inlet and mixture valves for one end of the cylinder is shown in the left hand drawing of Fig. XI.—47.

Air and gas enter the mixing chamber *M* by separate openings, shown closed by the valve discs *A* and *G* respectively for air and gas. These valves are connected by a cylindrical distance piece and therefore move as one. In the chamber *M* the mixture enters the cylinder by the main inlet valve *I* at the commencement of the suction stroke, and sooner or later during this stroke the valves *A* and *G* are closed by means of a trip gear by the coil springs which encircle the spindle, as determined by the governor.

The disc *B* is adjustable by hand from outside the casing to regulate the proportions of air and gas. The shoulder of the disc *B* serves as a guide for the lower end of the valve spindle.

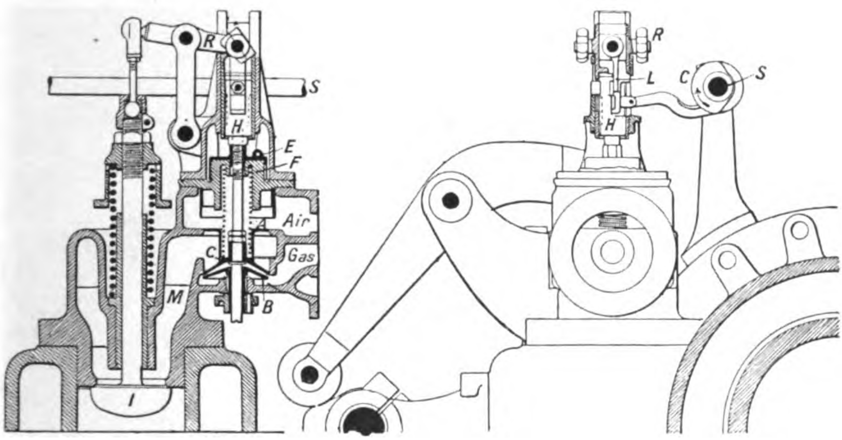


Fig. XI.—47. Snow Steam Pump Co.'s Governing Device.

The gas valve *G* moves in a conical passage in such a manner as to admit more gas in proportion to its lift. The air valve *A* has a bevelled seat.

The inlet valve *I* opens and closes always at the commencement and end of the suction stroke. It is operated by a rocking lever from the cam shaft, and is connected by means of a ball and socket jointed rod to another lever *R*, the other end of which is in connection with a sliding block moving up and down in guides. The sliding block is joined to a fulcrum lever *L* at the end of which is a catch block engaging with the end of the gas and air valve spindle, the catch block also serving as a guide for the spindle. This lever *L* is connected at the bottom to a kind of connecting rod the free extremity of which slides in a socket cam fixed on the shaft S. The latter operates the trip gear under the action of the governor and revolves at the same speed as the cam shaft.

As the piston moves forward along the suction stroke, the admission valve I opens, and, by the engagement of the arm L, the two valves for air (A) and gas (G) also open, until at a definite point of the stroke, determined by the governor, the cam shaft C of the trip gear shaft S, in revolving, releases the trip action and allows the mixture valves to close by the action of the spring. The top of the mixture valve

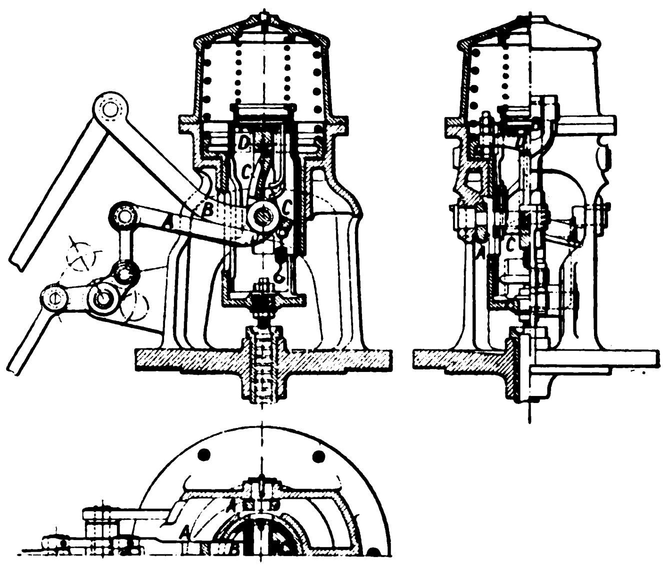


FIG. XI.—48. Nürnberg Governing Device.

spindle is fitted with a cylindrical box E, which, descending over the piston F, forms an air cushioning device.

It will be noticed that the designers of this gear have not been concerned in providing means whereby the mixture contained in the chamber M can be scavenged into the cylinder by a retarded closure of the air valve. The normal shape of the air passage and the conical shaped passage for the gas do not seem capable of obtaining a synchronous movement of gas and air which meet between the two discs of the mixing valve, and, therefore, it would appear that the proportions of gas and air would not be constant for all positions of the mixture valve trip gear, while the throttling of gas in the conical passage would have a tendency to cause deposit of tar and other matters in the event of imperfectly purified gas being employed.

(b) Method with Variable Ratio and Constant Volume.

(9) *Maschinenfabrik Augsburg of Nürnberg.* — The best known standard type of engine governing on the ratio of mixture with constant volume is certainly that of the Maschinenfabrik Augsburg. This engine has been extensively described. The early design of

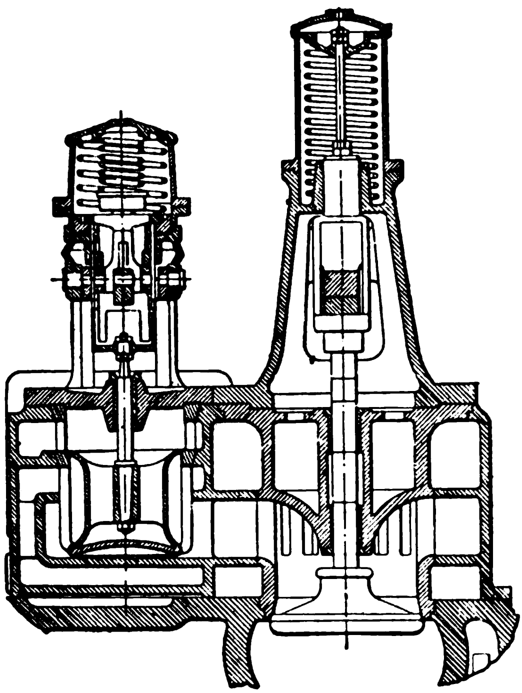


FIG. XI.—49. Nürnberg Governing Device.

control gear has now been replaced by the device shown in Fig. XI.—48, and therefore the illustration, Fig. XI.—49, of the mixing and inlet valves is given merely to assist recollection with respect to the leading principles.

In the general arrangement of the Nürnberg construction the main inlet valves are fitted at each end of the cylinder and are operated by eccentrics with sets of rolling path levers providing a constant motion for opening the valves, the latter being drawn back upon their seats by the usual coil spring fitted at the top. The gas valves are fitted between the two inlet valves on the longitudinal axis of the cylinder (Fig. XI.—49). They are of the double-seated equilibrium type, through which the gas is admitted earlier or later, near the end of the suction period, whilst air is passing around the body of these valves during the whole suction stroke of the piston. The mixture is made in the common channels ending at the main inlet valves. The mechanism shown in Fig. XI.—48 works as follows:—

An eccentric on the side shaft gives a rocking motion to lever *B*, fulcrumed on a pivot in a cylindrical box, traced in thick lines on the illustration. On the same pivot moves freely a piece *C*, which is maintained in position by a small coil spring. This piece *C* is the trip of a cut-off motion in which *D* is the fixed abutment. *D* is part of a cylindrical slide, concentric and external to that mentioned above, and to *D* the spindle of the double-seated gas valve is fixed. Lever *A*, used as a support to the operating lever *B*, terminates inside in the shape of a fork, the two branches of which are carried by pivots fitted on the frame whilst the outer free end is attached by means of a small connecting-rod and a crank lever, to the governor rod. Lever *A* moves under the action of the governor, and so causes earlier or later contact between the pieces *C* and *D* of the trip motion, and thus the gas valve cuts off the supply invariably at the end of the admission stroke when abandoned to the pressure of the strong spring and air brake piston surmounting the device. This method of governing answers the aim of the makers in ensuring the existence of a layer of air immediately behind the piston, and of a richer mixture close to the igniters, at the end of the admission stroke.

(10) *Crossley Brothers, of Manchester*, have recently applied a new system of governing to their large engines designed by their well-known chief engineer, Mr. J. Atkinson, after trip gears had been tried and abandoned, the method now adopted being simpler, quicker in action, very practical, and giving considerably better working.

The cross section of the valve arrangements is represented in Fig. XI.—50, and consists of a main casting *A* fixed on the top of each cylinder end.

The air is admitted by the connection *B*, and controlled by a hand throttle *C*, whilst the gas is admitted in the upper part of the valve arrangement by a gas connection *D* containing a gas cock *E* for regulation.

F is the admission valve, operated from the beginning of the inlet stroke in the usual way by means of cam, rods, and lever, and closed by the top spring *G* about the end of the stroke.

H is the gas valve with a reversed seat, provided at its upper end with a vacuum piston *I*; the quantity of air admitted behind this piston, so as to fill the vacuum cylinder entirely, partially, or not at all, is controlled by a small cylindrical plug *J*, operated by the governor and provided with a circular groove for opening or closing the air connection.

A very slight motion of this governing plug controls the work from full to no load.

A nut *M*, keyed on the spindle of the valve *F*, provided with a

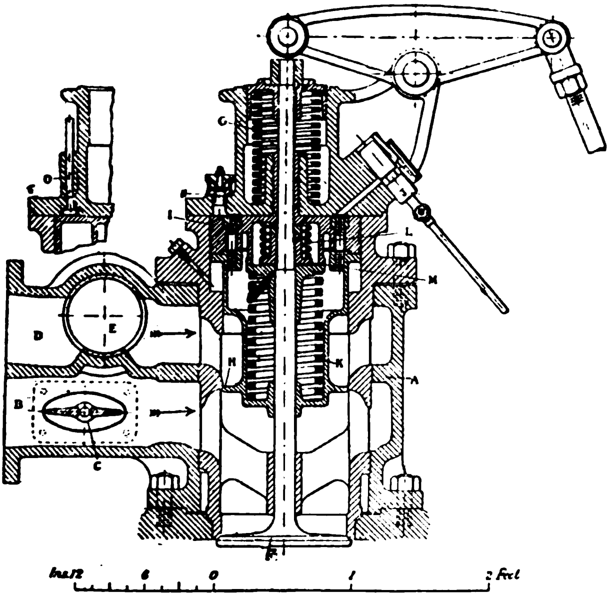


FIG. XI.—50. Crossley Governing Device.

buffer spring *L*, forces the vacuum piston *I* to the end of its upper stroke, when the admission valve *F* is closed.

The inner spring *K* is neutralised when the valve *F* is closed, and as soon as it opens the pressure of this spring at its upper end is taken by the nut *M*, and the pressure at the lower end tends to open the gas valve *H*.

If the governing plug *J* allows air to enter too freely behind the vacuum piston *I*, the gas valve *H* moves with the spring *K*, and closes with it, so as to allow a uniformly rich charge of mixture to be drawn through the admission valve during the whole suction stroke for realising the full power.

If, on the contrary, the plug *J* cuts off the admission of the air behind the vacuum piston, the latter prevents the gas valve being opened, a partial vacuum in the cylinder being sufficient to overcome the pressure exerted by the spring *K*. Under these circumstances air alone is admitted into the cylinder, and, of course, no impulse takes place.

For regular working, however, the plug *J* moves in an intermediate position more or less restricting the air passage, causing a partial vacuum behind the valve piston and restraining proportionately the opening of the gas valve, making it later and slower. As in the Nürnberg system, the gas is admitted more or less late along the suction stroke, but is stopped off always at the end of this stroke, causing first an admission of air and afterwards of mixture.

The little snifting-valve *N* opening outwards from the vacuum cylinder ensures a prompt return of the valve *F* in closing. A small indicator rod (*O*) (see separate section) is also fitted to the vacuum piston so that its working can be made visible to the attendant.

It will be realised that the governor has no resistance to overcome, the cylindrical plugs moving easily in their cylinders, with the result that accurate control can be ensured to the four separate valve arrangements which constitute the gear of a double-acting, two-cylinder engine. The internal working parts are protected from dust and tar that might cause them to stick.

It is said that in the course of the experiments made upon the first engines fitted with this gear by Messrs. Crossley Brothers the power was suddenly dropped from 600 to about 50 H.P., and loaded again without causing more than 1·65 per cent. total variation in speed. That is, of course, a wonderful result, and undoubtedly proves the sensitiveness of the governing device. Time will show whether the effects of wear and lack of tightness that may take place between the rubbing surfaces of the air piston and plug will affect its sensitiveness.

(11) *John Cockerill Co., at Seraing.*—In the section of this chapter dealing with the mechanism for governing with a mixture of constant ratio and variable volume, one of the constructions of the Cockerill Co. has already been described. The author believes that this firm have very wisely solved the question so much discussed as to the advantages of governing on the quantity compared to governing on the quality, by designing a type of governing for each method, and using either. according to the duty of the engine. Their system of governing on the mixture admitting a constant volume to the cylinder (and, of

course, at constant compression) is realised in a very simple and commendable manner, as shown in Fig. XI.—51. The inlet valve is

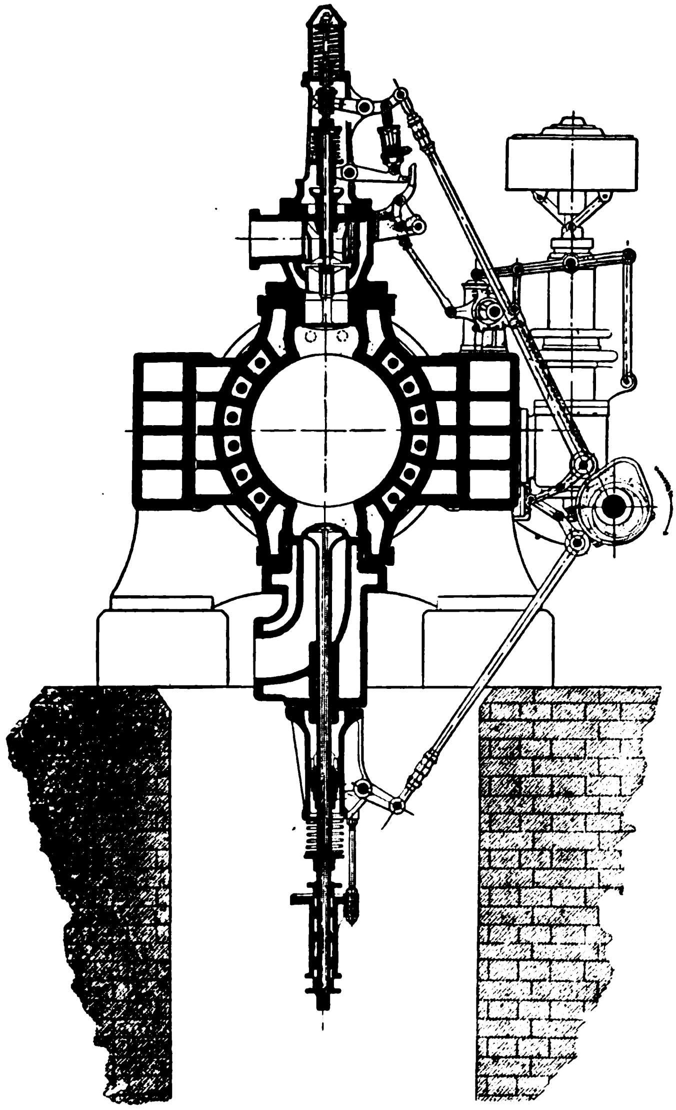


FIG. XI.—51. Cockerill Governing Device.

opened by the usual device of lever, rod, and cam, and is closed and kept on its seat by means of a powerful spring, fitted on the top of the spindle. On this spindle, and behind the valve itself, is also fitted a cylindrical slide that controls the admission of air during the

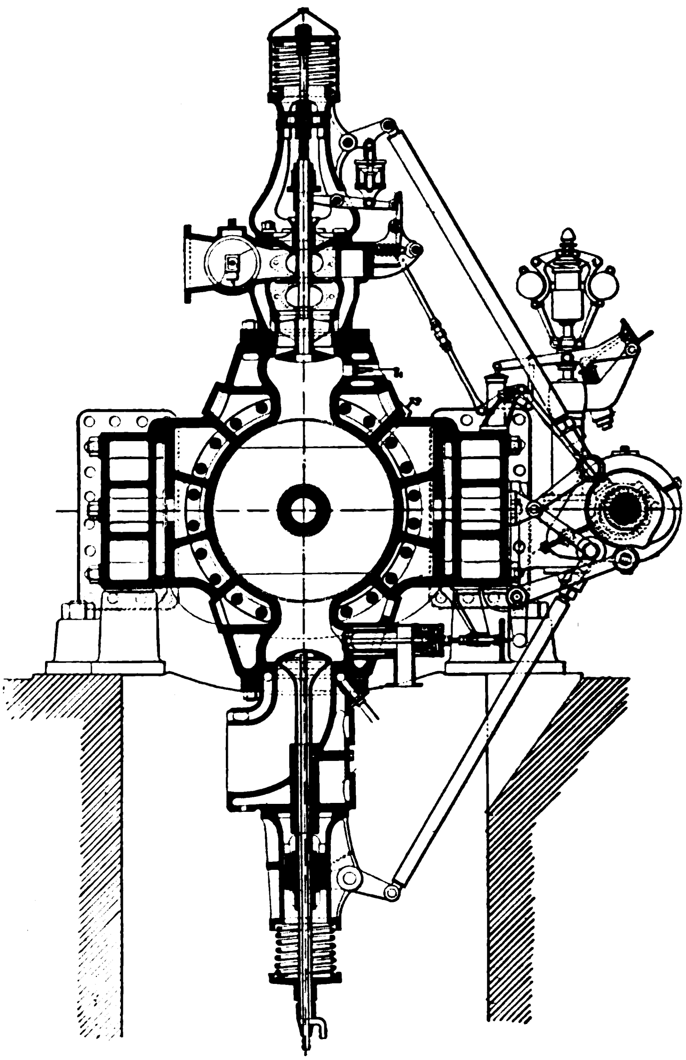


FIG. XI. -52. Cockerill Governing Device.

whole suction stroke. Above this slide is the double-seated equilibrium gas-valve, fitted on a sleeve moving concentrically with, and independently to, the spindle of the main inlet valve. The equilibrium gas valve is connected to one of the ends of a lower lever, pivoted on the frame. The other end of this lever is held by means of a

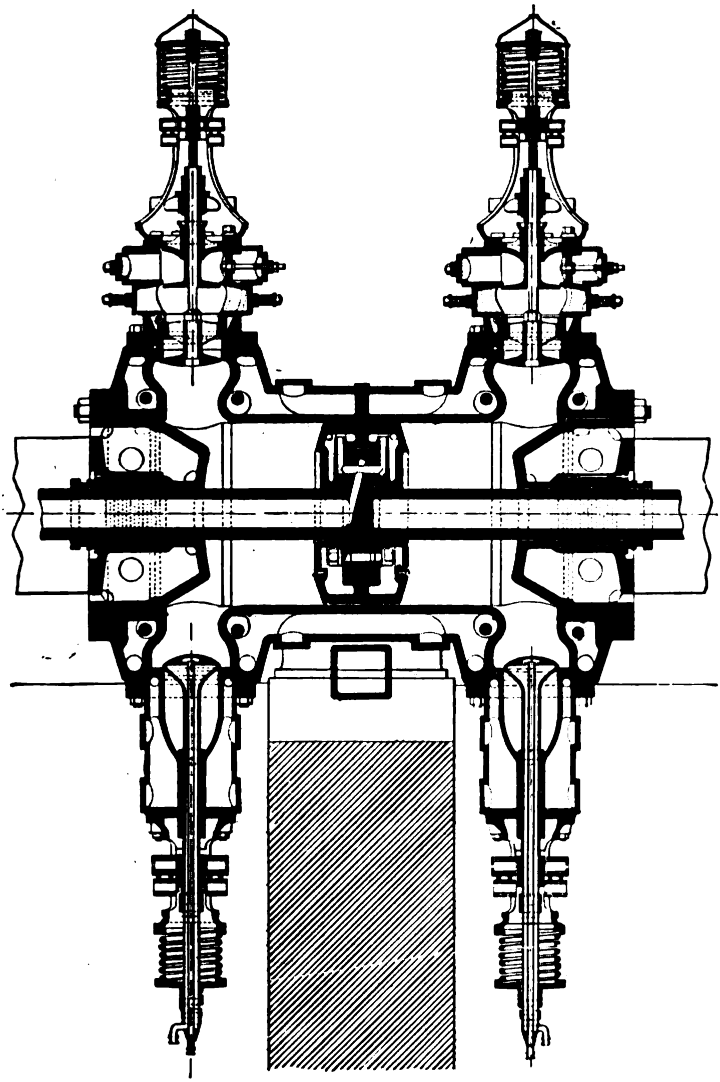


FIG. XI.—53. Cockerill Governing Device.

governor-controlled trip gear, also connected, through an air-brake piston, to the operating upper lever of the main inlet valve, the latter having a uniform lift.

The gas valve is connected by a coil spring to the inlet valve spindle. At the beginning of the inlet, air alone, coming from the slide, is admitted through the main inlet valve, whilst the lever of this valve compresses the spring of the gas valve. The gas valve is kept closed during the time the air is entering until towards the end of the suction stroke, when the governor actuates the trip gear of the air piston, which, abandoning the gas valve to the action of its spring, causes it to open suddenly. The gas then enters until the end of the stroke. The mixture is thus made at the end of the suction stroke, and therefore remains in the bottom of the cylinder near the ignition plugs when compression starts.

It will be noted that the valve casing carrying the supports of the levers used in this system of governing is, at the point of its connection with the cylinder, of the same shape and dimensions as the valve casing used when governing by the regulation of the quantity. This fact makes the transformation of one system of governing to the other quite a simple matter. This is the only example of which the author is aware where both systems are interchangeable, and where they are designed in such a rational way viewed from both constructional and working stand-points.

The engineers of the John Cockerill Co. were the pioneers of large blast furnace gas engines of the double-acting type. In spite of their wide experience, improvements on their early makes have been relatively slowly carried out, but have resulted in the construction of engines which to-day may be cited as the model of their kind. Just recently they have given up their system of governing on quantity, and they have unified all their designs in accordance with the type described above. At the same time, certain alterations have been made in detail, particularly with regard to the equilibrium gas valve being replaced by an ordinary disc valve with a single seat, as shown in Figs. XI.—52 and 53.

(12) *Ehrhardt & Sehmer, of Schleifmuhle.*—This firm first constructed their gas engines under licence from the Gasmotoren Fabrik Deutz, with cam gear for both inlet and exhaust valves. The mixing valve was of the double-seated type, admitting the air from above and the gas from below. The quantity of mixture admitted at constant ratio was dependent on the stroke of the valve as varied by the governor. The variable stroke device was obtained by the regular Deutz mechanism, the basis of which is the displacement of the support of the operating lever, without intervention of any cut-off gear. The system of governing by cut-off operating on the mixture,

as now applied by Ehrhardt & Sehmer was designed by their chief engineer, Mr. Drawe, and is developed from original theories which were very elaborately expounded in a report from which the author proposes to quote some of the interesting statements in support of the new system of governing. Referring to the diagram Fig. XI.—54, if the lift of the ordinary valve is represented by the curve *a b*, in functions of the piston stroke x, x_1, the ascending period of the curve $a-b$, corresponds to the period during which the valve is mechanically operated in its progressive opening, whilst its closing, when the cut-off takes place along the stroke, is shown by the line b, b^1, which is the maximum ordinate corresponding to the highest lift of the valve. Now, this would be correct, if the quantity of gas admitted remained constant for any position of the piston, so as to avoid the inertia effects due to the relative variations of the speeds between the admission of air and that of the gas.

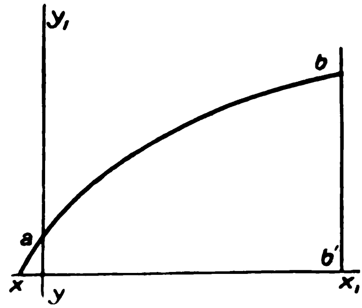


FIG. XI.—54. Diagram of ordinary Valve Lift.

The curve representing the openings of the gas valve would, of course, be parallel to the axis of abscissæ x, x^1, independently of the variations of the valve lift.

Messrs. Ehrhardt & Sehmer have adopted a special shape of valve, as shown in Fig. XI.—55, for the purpose of assuring a constant supply, because the dimensions of *a* and *b* may increase, whilst *e* and *f* remain constant. Fig. XI.—56 shows the curve 1 of the opening of the main inlet valve; the curve 2 is the opening of the gas valve parallel to the axis of X. The intersection of these curves at m, for example, corresponds to the point of maximum introduction of mixture in the cylinder, obtained with an equivalent opening of both the gas valve and the main inlet valve. Contrary to what happens in other systems where gas is *admitted* more or less early before the end of the suction stroke, in the Ehrhardt system the gas is *cut off* more or less early before the end of the stroke, so that the

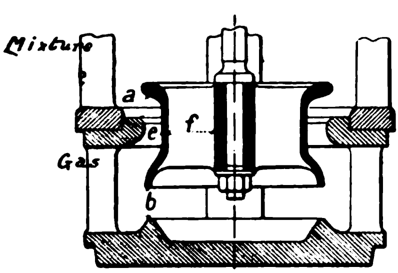


FIG. XI.—55. Ehrhardt & Sehmer Admission Valve.

latter always finishes with an admission of pure air which sweeps out all the mixture that may be contained in the gas valve passage. The rich mixture would thus remain near the piston, whilst poorer mixture, or even pure air, would be confined in the combustion chamber near the ignition plugs. This is an absolute denial of the hypothesis of stratification of fluids, and, in this respect, the method applied by Mr. Drawe is as daring as it is interesting. Notwithstanding, it is of great interest to note that it is said to secure perfect practical results.

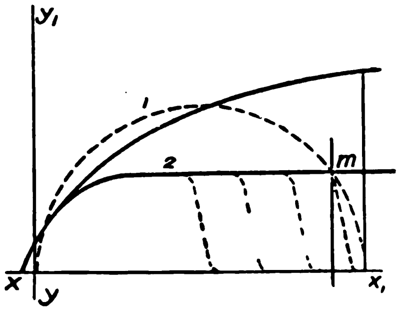


FIG. XI.—56. Diagram of Ehrhardt & Sehmer Valve Lift.

The main inlet valves have peculiar features. They are operated by rods and cams, and are fitted on the top of the cylinder on the longitudinal axis. There are no separate air valves. The gas regulation valves are placed between the main inlet valves, and their working

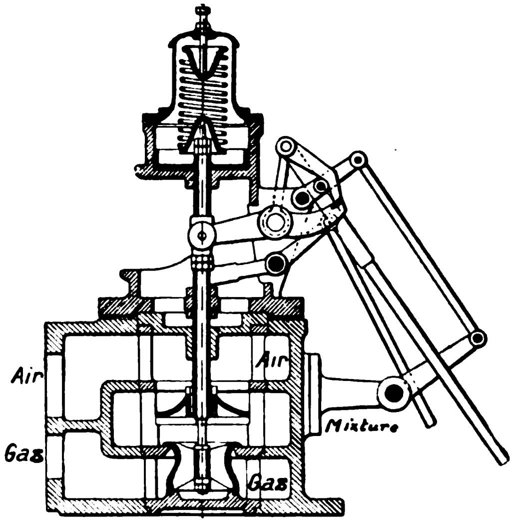


FIG. XI.—57. Ehrhardt & Sehmer Governing Device.

will easily be understood from Fig. XI.—57, in which an independent regulating disc can be seen on the rod of the gas valve. Although of a very simple design, the disc arrangement presents the peculiarity of operating simultaneously on the one relatively to the other, and, at the same time, acting as a baffle preventing both fluids from rushing against each other before mixing.

The gas valve is operated by the rod of an eccentric, through a trip resting on a lever fulcrumed to the valve, and supported by the casing. When gas must be cut off in relation to the load, about the end of the suction stroke, the governor causes a roller to slide off the abutment of the valve lever. The latter is then abandoned to the pressure of the spring with an air cushioned piston, fitted on the top of the casting, giving a sudden but smooth closure. The air alone continues to enter the cylinder by the central channel, and sweeps out the space between the gas valve and the main inlet valve, thus preventing back firing. It will be noticed that, in this governing gear, no rubbing surface exists, so that neither sticking nor hanging of the valve can take place owing to the presence of tar or dust.

(c) Combined Method with Constant Mixture Ratio and Constant Volume.

(13) *Thyssen & Co., Mulheim on Ruhr.*—The general arrangement of this construction, like the Nürnberg, consists in placing the two main inlet valves at the top of the cylinder with the regulating valves between them. The latter have been the object of special investigation on the part of Mr. Richter, formerly chief engineer of the Nürnberg firm, but now in charge of Thyssen & Co.'s gas engine department. In this mode of governing, it is not the admission of the gas which is controlled by the governor, but the admission of a mixture ready made by valves which open more or less late along the suction stroke, and are automatically cut off, invariably at the end of the stroke, when the main inlet valve itself closes. The contrivance consists of a double-seated valve and a slide, which are both fixed on the same spindle. The equilibrium valve in the lower part controls the gas inlet, whilst the circular slide on the top admits the air. When the valve is closed, the slide permits the air to enter the cylinder alone through the by-pass shown at the right hand of Fig. XI.—58.

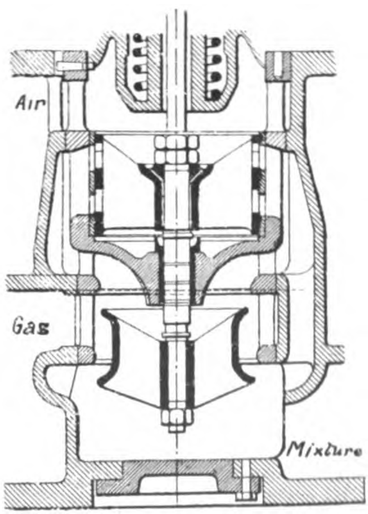


FIG. XI.—58. Thyssen Governing Device.

When the gas valve opens, the slide uncovers the air ports progressively in proportion to the passage areas for the gas. The result is that for any position of these valves, and until they are closing, the speed of both fluids remains constant. The equilibrium valve being used at the same time for gas and air secures a uniform mixture which, in moving, depends upon the accelerated speed of the pure air admitted before the equilibrium valve opens. It would be interesting to know precisely whether this accelerated speed of the pure air, admitted in advance, does not prevent the formation of a uniform mixture, because if the regulating device opens more or less late during the admission of pure air a lack of synchronism between the speed of both fluids may take place, owing to the fact that the gas has a velocity depending on the governor, whilst the air enters continuously at the velocity depending on the speed of the piston. In this device disturbances in the formation of the mixtures may be so much the more prejudicial, because the system of governing on the mixture has an inherent tendency to cause misfirings at low loads when the mixture is much diluted and impoverished.

(d) **Combined Method with Constant Ratio and Constant Compression.**

Schuchtermann & Kremer, of Dortmund.—This firm has taken up Mr. Reinhardt's patent, who has combined the devices with the object of obtaining a constant mixture without varying the compression at different loads, by means of increasing the amount of air admitted for filling up the cylinder. As shown in Fig. XI.—59, the main admission valve has a constant lift obtained by eccentrics, rods and rolling lever, and is fitted above the cylinder. Behind this valve is an independent cylindrical slide, operated through its sleeve by a trip gear. This slide controls three series of ports, of which the upper one *I.* serves for the gas and both the intermediate *II.* and the lower one *III.* serve for the air. In the beginning of the suction, the openings for the gas *I.* and air *II.* are covered by the slide, whilst the lower ports for the air *III.* are open. Through the latter pure air is first drawn, until the instant when (depending on the work of the engine) the trip gear, which up to then kept the slide up, abandons it to the action of its spring. Then the latter suddenly lowers the slide, to the position shown in the figure. The primary air port *III.* is then closed, whilst the gas port *I.* and air port *II.* are opened. When the piston has reached the end of its suction stroke, and when the main inlet valve closes, the regulating slide is re-engaged by the trip gear and lifted to its initial position, closing the upper gas and air ports.

The cut-off gear operated by eccentric and fitted with an air cushion piston, is similar to the systems usually applied in the Cockerill and Nürnberg constructions.

It will, however, be observed that in the position for admitting the

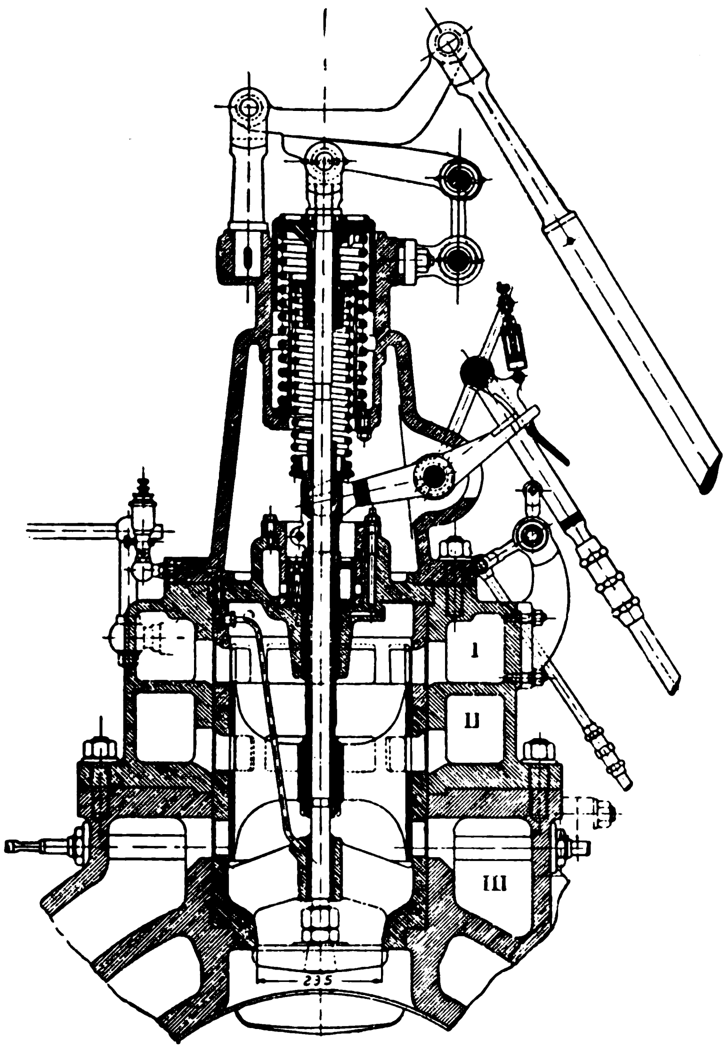


FIG. XI.—59. Reinhardt Governing Device (Schuchtermann & Kremer).

mixture, in Fig. XI.—59, the ports *I.* and *II.* being simultaneously open, there must be in the zone which is separating them inside the box a smaller depression than in the zone between ports *II.* and the inlet valve, since for an equal area in the first zone only one fluid moves, whilst in the second two fluids are moving—the latter being a mixture of equal volumes of air and gas in the case of blast furnace gas. The result of this might cause disturbances in the formation of a homogeneous mixture, and it would be easy to remedy it by placing those ports in the same horizontal plane, and by alternating the air and gas ports, thus dividing both fluids into streams favourable to the formation of a perfect mixture. It should, however, be taken into account that the gas alone is under pressure.

Notwithstanding, Mr. Reinhardt seems to have solved the problem of realising the displacement of an equal column of air and gas in a very ingenious manner, and without causing a very heavy vacuum at low loads. At the time of the Congress of Mechanics, held at the Universal Exhibition at Liège in 1905, the author, in a paper on the governing of large gas engines, enumerated different principles that the makers should endeavour to realise. Among these he mentioned "The realisation of governing with a practically constant mixture to be admitted in variable quantity without causing depression in the cylinder." The device designed by Mr. Reinhardt in his combined arrangement for governing at constant ratio and compression is said to answer the above-mentioned desideratum.

The engine constructed by Schuchtermann & Kremer presents another peculiarity in its construction. This consists in a side chamber for the exhaust valves at both ends of the cylinder, thus making them more accessible and also avoiding the interruption of the continuity of the foundation underneath the cylinder (Fig. XI.—60).

An automatic oil-drain valve is fitted at both ends of the cylinder, thus answering another desideratum set forth in the above-mentioned report at the Congress of Mechanics.

The principal and most interesting governor gears for four-cycle engines, which may be cited as European examples of sound mechanical conception and perfection of workmanship have now been examined.

Apart from the classification that has been adopted in accordance with the method of regulating the explosive charge under governor control, other classifications may be made by considering the relative positions of the admission valves to themselves and to the cylinder. Some makers have placed the gas valve and mixing valve in separate casings from those containing the main inlet valve, and have either fitted them on the longitudinal axis of the cylinder between the main inlet valves, or behind these valves. In the first classification relating to the large engines are the Nürnberg engines (Figs. XI.—48 and 49); the latest system of the Gasmotoren Fabrik Deutz (Figs. XI.—35 and 36) applied to engines of 2,000 H.P.; the constructions of Ehrhardt & Sehmer (Figs. XI.—55 and 57), and Thyssen (Fig. XI.—58). These devices have a separate and independent operating gear

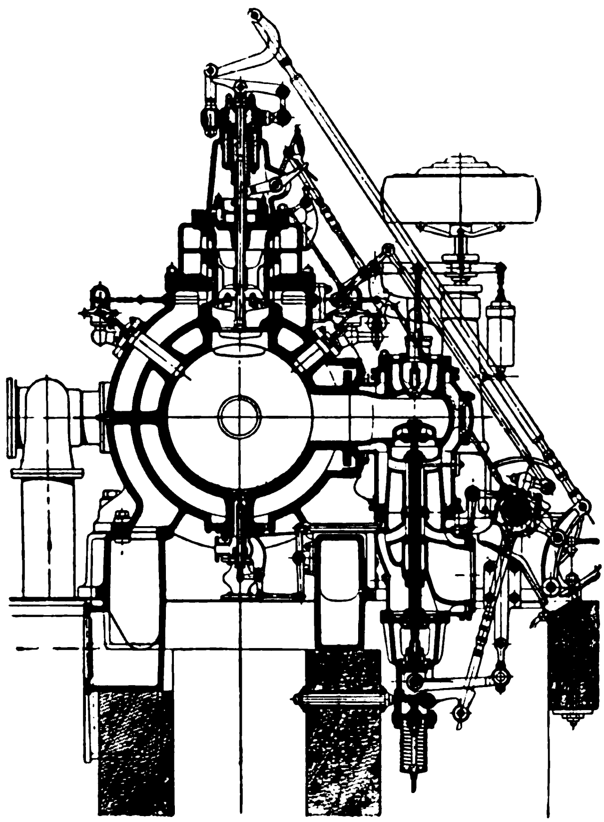


FIG. XI.—60. Schuchtermann & Kremer Engine.

for the gas valve or the mixing valve, whilst the combination with a mixing chamber behind the admission box can be operated by a simple arrangement from the side shaft, either by means of a cam or an eccentric, such as in the constructions of Gasmotoren Fabrik Deutz (Figs. XI.—34 and 35) and Gutehoffnungshütte (Fig. XI.—40).

The mixing devices, if placed between or at the side of the admission valves, are obviously more accessible, and this presents a real advantage with respect to the ease of dismantling for inspection and cleaning, and especially in those cases where the gas might not be thoroughly purified and in consequence would be likely to foul the inner working parts with dust or tar. From this point of view, this type of construction is particularly suitable for large central power houses such as central electric lighting stations.

The next construction to be considered is the arrangement with regulating valves concentric to the main inlet valve and located in a common box, as in the type of the Cockerill engine (Fig. XI.—51), Elsassische (Fig. XI.—41), Dingler (Fig. XI.—43), Crossley (Fig. XI.—50), Schuchtermann (Fig. XI.—60). The simplicity and neat appearance of these devices are very attractive, necessitating very few joints and fittings and resulting in the minimum of mechanical operating parts. The shape of the mixing box, immediately behind the admission valve, is simple and rational, and the air scavenging at the end of the suction stroke prevents any mixture remaining in the recesses or pockets of the mixture box. When it happens that, due to either of the causes examined and illustrated in Figs. XI.—3, 4, and 5, a back-fire takes place, it will only affect a very small volume of mixture, and the resulting products of combustion will only slightly influence the following new charge. In the latter types, characterised by a small dead space, the action of the governor is obviously more accurate and quick than if the mixing valves were separated from the inlet valve by a long channel. From both constructional and working points of view, the governing device, placed concentric to the inlet valve and located with it in a common box, suggests the possibility of its acceptance as a standard in the future.

Wherever possible the gas or the mixing valve will be used in preference to the slide, because the rubbing parts of the latter need lubrication, which it is difficult to secure with any certainty. Moreover, it is the oiling of these parts which invites the dust in suspension in the gas to become deposited on the rubbing surfaces. As a mixing device, the double-seated valve seems to answer better than the regular mushroom valve; first, because it affords a double passage, the section and direction of which can be varied so as to help the circulation of the fluids, and also because it particularly suits the automatic cut-off action. It is balanced and does not feel the variations of resistance due to the gear which are liable to influence the governor. It is this feature which has caused it to be so extensively used in steam engineering.

The trip gears, which secure the independence of the governor, are now applied to almost all large gas engines of modern construction in which, in spite of all other advantages, similar arrangements to those applied to vary the stroke of the admission valve in single-acting engines, would not work satisfactorily. Of course, with mechanisms for small engines of the kind of Gasmotoren Fabrik Deutz (Fig. XI.—14), Schmitz, Fig. XI.—24, &c., the movable piece controlled by the governor is necessarily engaged and locked during the whole suction stroke, and becomes free only during the three consecutive strokes of compression, explosion, and exhaust. Now, in a double-acting engine with two cylinders there is always one end of the pistons at the suction period, and the result would be that the movable piece moved by the governor would always be engaged, and the action of the governor annihilated. This would be the case with the large tandem double-acting type of Gasmotoren Fabrik Deutz, if these skilled constructors had not overcome the difficulty by making use of a sort of ball coupling which gives the governor liberty to operate the gear of one of the cylinders when the other is locked.

It will be seen from this practical study of the different methods of governing which are in favour for modern large four-cycle engines, that all makers endeavour to secure an explosive mixture giving the highest thermal efficiency at different loads. Less attention seems to have been given to the question of the most favourable shape of the explosion chamber, although this factor of efficiency becomes of great importance as the sizes are increased.

The Duisburger Maschinenfabrik Co., at Duisburg, however, have made a praiseworthy attempt in this direction, and in their double-acting engine have shifted the vertical axis passing through the admission and exhaust valves to a position lateral to the axis of the piston. With this arrangement they claim a more homogeneous mixture, and that when ignition takes place, the flame is propagated in the mixture in a spiral wave creating whirls of burning gases, and thus securing a more thorough combustion of the whole bulk of mixture. In the author's opinion the useful effect is more in favour of complete combustion than that obtained from mechanical combinations for timing the ignition automatically by means of mechanical gears, in such a way that the ignition is advanced under the action of the governor in proportion as the mixture is more diluted.

As a matter of fact, ignition is only one of the numerous factors on which the combustion of the very variable mixtures fed into large engines depends. The temperature of the metallic enclosures which change continuously with the charge, the temperature of the air and of the gas, their pressure, their state of dilution, the variations in composition of the gases, &c., are other factors that will greatly alter the conditions of combustion of a mixture in a given cylinder. This explains why, although everything may be apparently equal, and the engine be carefully maintained at constant load, one is very seldom able to record five or six consecutive indicator diagrams which are identical with respect to initial explosive pressure, the expansion curve, and to the mean effective pressure.

From the British and American stand-point, especially, the question of automatic timing of the ignition is of no interest, because it leads to mechanical complications, and none of the Continental firms whose constructions have been discussed, seem to have taken it into consideration. A very elaborate report on the subject of unevenness in the combustion of the mixtures, independently of the ignition, has been presented by Dr. Ch. E. Lucke to the American Society of Mechanical Engineers, and is referred to elsewhere.

During the translation of this work two interesting methods of governing have been introduced to which reference should be made. The first is illustrated in Figs. XI.—61 and 62, and has been designed by the National Gas Engine Co., Ltd., of Ashton-under-Lyne. The main inlet valve to the cylinder receives a constant stroke by means of the usual cam and lever, but graduated charges of variable volume but constant ratio are obtained by varying the lift of supplementary gas and air valves, arranged upon a common spindle, and operated by a separate cam and a special but extremely simple form of mechanism.

The movement of the supplementary valve cam is transmitted to the spindle by means of two short levers, 1 and 2, fulcrumed respectively at *A* and *B*. The roller *C* is always in contact with the cam, and therefore the lever *B* is always moved through the same distance. The movement of *B* is, however, communicated to *A* by means of the governor die *D* interposed between them. The governor regulates the position of *D*, and it will be clear from the illustrations that the travel of the valve spindle is thus directly affected. In Fig. XI.—61, the mechanism is shown in the position taken up by the governor when the engine is running under no load, and thus the die is at the minimum distance from the fulcrum *B* and the maximum from the fulcrum *A*, and, as a consequence, the lift of the gas and air supplementary inlet valve is reduced, so as to admit only sufficient gas to keep the engine in rotation. In Fig. XI.—62, the governor die is shown in the position that is at a maximum distance from *B* and the minimum from *A*, giving a lift sufficient to ensure full load charges.

An advantage of this system is that little or no work is thrown on the governor, while all details are of great simplicity, with but a few wear-

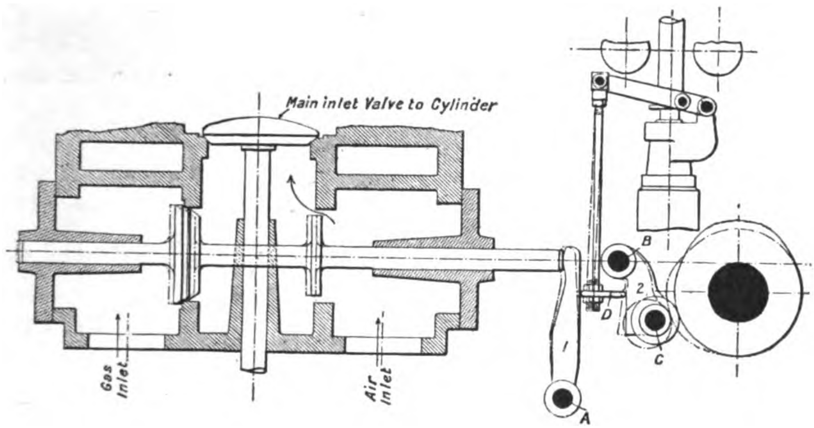


FIG. XI.—61. National Governing Device (light load).

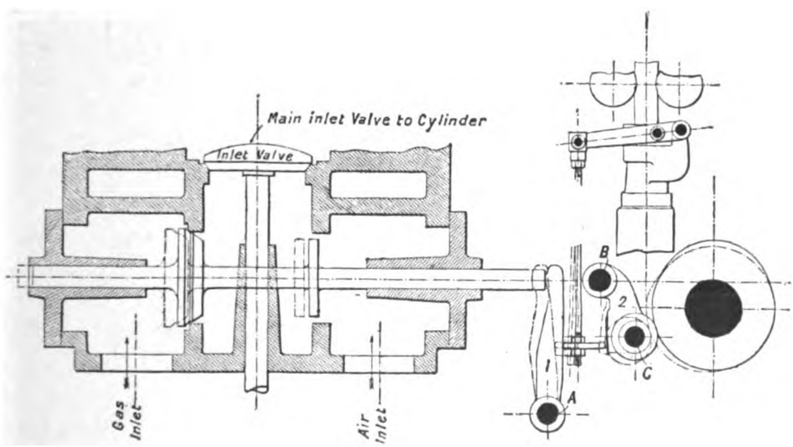


FIG. XI.—62. National Governing Device (full load).

ing parts, such as links and pins, to cause erratic working due to ordinary wear and tear.

Fig. XI.—63 illustrates the governing mechanism of the latest American Westinghouse horizontal gas engines of large power as fitted to those made for the Indiana Steel Co.'s blowing plant at Gary, U.S.A. This is similar to the German Mietz arrangement. The inlet valve is operated by a single eccentric with rolling contact levers, as is also the exhaust valve, this independence permitting the valve

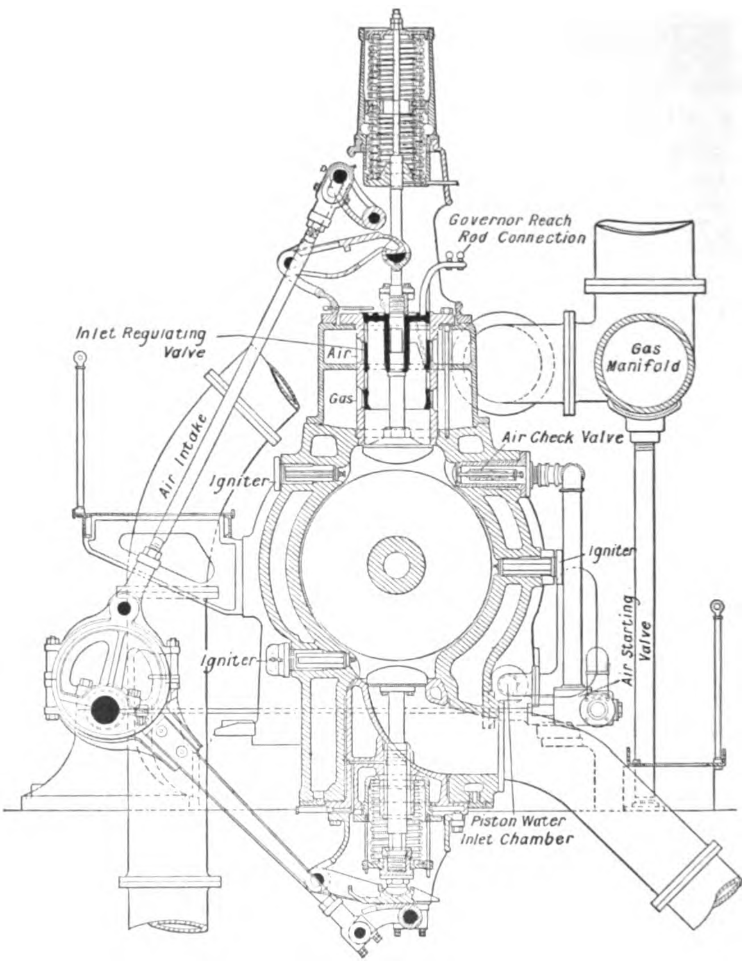


FIG. XI.—63. American Westinghouse Governing Device.

settings to overlap so as to encourage scavenging by inertia and to obtain fuller charges.

The main inlet valve is given a constant lift, but the volume of mixture is controlled by a cylindrical slide-valve with apertures which register more or less with others in the valve casing and thus adjust the charge to suit the load. The cylindrical valve is rotated slightly as required by a rod, which couples all the similar valves in each line of double-acting, tandem cylinders, and this rod is actuated by oil at a pressure of 50 to 60 lbs., provided by a plunger pump driven from the engine crank shaft. The centrifugal governor is thereby relieved of all work except operating a small pilot valve which controls the supply of oil to the working cylinder of the system. The oil pressure can always be noted by a gauge conveniently disposed, and should the pump fail, a small gravity accumulator serves to maintain pressure until reserve valves can be opened. In addition to the main governor, a centrifugal safety-stop device is arranged at the fly-wheel rim which trips the main igniter switch at a predetermined speed and causes the engine to stop.

The Premier system of governing is described on p. 404. The recently designed method adopted for Hornsby-Stockport engines is referred to and illustrated in Chapter IV., p. 42.

Figs. XI.—61 and 62 are reproduced from *The Engineer*, London, and Fig. XI.—63 from *Power*, New York.

CHAPTER XII

DETAILS OF CONSTRUCTION

FORMULÆ. DIMENSIONS OF STATIONARY PARTS

In giving some practical formulæ to simplify the calculations with regard to the dimensions of the different details of gas engines, the author desires to point out that, being empirical, they should not entirely substitute those established after more precise principles based upon the resistances of materials. The latter, however, studied in detail by several authors, are often presented in such a complex form as to render them scarcely intelligible and their application difficult. Especially is this so when account must be taken of all the factors that come into play, of which some cannot be precisely determined.

With empirical formulæ, a portion of the factors are neglected, and certain coefficients substituted, deduced from experimental data. In this way determination or verification of the dimensions of parts can be effected with sufficient exactitude for general purposes.

The formulæ here given, therefore, may be employed to approximately fix the required dimensions, recourse being made to more precise calculations in some particular cases so as not to subject the metal to abnormal stresses.

It should be remembered that the most precise and most complicated formulæ contain nearly always some coefficient to take into account some hypothesis or other, which, in consequence, destroys their character for absolute exactitude. They frequently conduce to unduly strong dimensions which uselessly increase the weight and the selling price of the engines.

Practical engineers and designers, therefore, make use of empirical formulæ, leaving the others to the theorists who are more particularly interested in pure mathematics. British and American makers generally apply the former, and the engines made in accordance therewith are strong and substantial, yet relatively light.

Different tables are given showing the dimensions that should be given to the various parts according to the power of the engine. The author suggests the following procedure for deciding the details and the dimensions of an engine to be built:—

1. To take the normal power to be developed.

2. To choose the type of engine—four-cycle or two-cycle, single or double-acting, single-cylinder, tandem or twin, horizontal or vertical, &c.

3. To deduce the diameter of the cylinder and the stroke of piston. These should be within the limits fixed by practice, and decided with respect to the number of revolutions per minute and the mean linear piston speed. The ratio between the stroke and diameter of the cylinder depends upon the mean pressure per cycle upon the piston, according to the fuel used equivalent to those shown in the table on p. 278.

4. To determine the weight of the fly-wheel to permit a certain degree of regularity according to the purpose for which the engine is designed. A profile to be chosen for the rim and the different dimensions calculated.

5. According to the manner in which the power is to be transmitted by the fly-wheel or by a pulley, and according to the number of bearings that it is proposed to employ, to fix the different parts of the crank shaft, and to deduce their dimensions to suit the stresses they will have to withstand.

6. To calculate the connecting rod from knowing its length; to determine the dimensions of the bearings and of the bearing bolts.

7. To calculate the piston and fix its different dimensions.

8. The previous essentials being known, to draw the frame to show the form and stiffening webs.

9. To design the combustion chamber according to the compression pressure to be attained, the method of governing to be adopted, and the type of engine.

10. To determine the different details of the valve gear—cam shaft, cams, levers and rollers, valves and their springs, governor, ignition device, &c.

In the following examples, the author takes into consideration fuel gas engines, single-acting, calculated to produce a mean effective pressure Pm of 70 lbs. per square inch per cycle, and a mechanical efficiency R of 80 per cent.

The majority of the formulæ will be in terms of the piston diameter, and will assume an initial explosion pressure of 430 lbs. per square inch, and a resistance to tensile stress of 3,000 to 4,000 lbs. per square inch for cast iron as a safe working load.

Constant Effective Work. B.H.P. . .	15.	25.	35.	50.	75.	100.	150.
Piston . . diameter D inches	8	$10\frac{1}{4}$	12	$13\frac{3}{4}$	$16\frac{1}{2}$	$18\frac{3}{4}$	$22\frac{1}{2}$
stroke L „	18	19	21	$22\frac{1}{2}$	$25\frac{1}{2}$	28	$32\frac{1}{2}$
Ratio . . $\frac{\text{stroke}\ L}{\text{diameter}\ D}$ „	2·25	1·85	1·75	1·64	1·54	1·5	1·44
Revolutions per minute, N	230	220	210	200	190	180	160
Linear-mean-piston speed, in feet, per minute, V	690	700	735	750	810	840	870

The above figures are derived from data collected on engines of the best modern practice.

Single-acting Gas Engines—Principal Details.—The majority of engines are designed to drive dynamos exclusively or simultaneously with other machines. There is a tendency, therefore, to increase the number of revolutions and mean linear speed of the piston as being favourable to efficiency.

In small engines considerations of wear and vibration render it necessary not to exceed a certain angular speed, and consequently the linear piston speed is restricted thereby.

According to *Haeder* the ratio $\frac{L}{D}$ should be about 1·6 for engines over 27 inches of stroke. Güldner suggests 1·2 to 1·3 for large engines, and advises the ratio to be made as large as possible.

Tangyes, Ltd., of Birmingham, in their former type of engines adopted the ratio of from 2 to 1·2. In the latest type they have increased this ratio, and have also increased the number of revolutions per minute.

Koerting Brothers, of Hanover, generally use long strokes and low rate of revolutions, the ratio varying between 1·9 and 1·6.

The ratio depends, also, upon the form and volume of the compression chamber. If this be large, as in cylindrical or hemispherical chambers, it necessitates a long stroke in order to attain a high compression.

With respect to the number of revolutions per minute, the figures of the above table are higher than those applied several years ago. The care taken in balancing the moving masses, and the finish of construction, now enables such limits to be reached without prejudice to the engine's smooth and silent working.

To determine the dimensions which enter into the ultimate power of the engine it will be logical therefore to decide, first of all, the number of revolutions suitable for the particular service and the corresponding linear piston speed. This for industrial engines should not exceed 900 to 1,000 feet per minute, while for automobile engines it may be 1,200 to 1,500 feet per minute. At the present time these limits are considerably exceeded for special types.

The formula for mean linear speed is:—

$$V = \frac{2 \times L \times N}{12} = \frac{LN}{6} = \text{Velocity in feet per minute,}$$

the stroke being:—

$$L = \frac{6V}{N}.$$

The stroke being fixed, the diameter of the cylinder is deduced by the formula:—

$$\text{B.H.P.} = \frac{\frac{\pi D^2}{4} \times \frac{V}{4} \times Pm \times R}{33000}$$

therefore

$$D = 4 \times \sqrt{\frac{33000\ Pe}{\pi\ V\ Pm\ R}}$$

in which:—

B.H.P. = Effective or brake H.P.
V = Linear piston speed in feet per minute.
L = Length of stroke in inches.
N = Revolutions per minute.
Pm = Average mean pressure on the piston in lbs. per square inch.
D = Diameter of piston in inches.
R = Coefficient of mechanical efficiency $\frac{\text{B.H.P.}}{\text{I.H.P.}}$

The formula applies to single-cylinder, single-acting engines for which the expression $\frac{\pi D^2}{4}$ represents the piston area.

For engines whose pistons are furnished with a piston-rod or tail-rod of d diameter, the effective area will be expressed by

$$A = \frac{\pi}{4}(D^2 - d^2).$$

The expression $\frac{V}{4}$ applies to a single impulse during 4 strokes of the piston or 2 revolutions of the fly-wheel. To calculate the power of multi-cylinder or double-acting engines the expression $\frac{V}{4}$ is replaced by the following values according to the arrangement of cylinders shown in Figs. V.—4, 5, and 6.

For arrangements II. to VII. inclusive; $= \frac{V}{2}$

„ „ VIII., IX. and XIII. „ ; $= V$

„ „ XII. to XIV. „ ; $= 2V$

and so on according to circumstances.

The mechanical efficiency $R = 0{\cdot}80$ is an average applicable to the different engines with a heavy wheel. For engines with lighter wheels R can be taken as $= 0{\cdot}85$.

In four-cycle engines, single and double-acting, R may vary between 0·80 and 0·87 and fall to 0·75 or even 0·70 in two-cycle engines with one or two cylinders, owing to the work absorbed by the gas and air pumps.

The mean pressure $Pm = 70$ lbs. per square inch is an average value allowing a certain margin, and not the limit of power the engine can reach. Under some favourable conditions the mean pressure attains 85 lbs. per square inch and more, with power gas.

The following is a list of different fuels used and the mean pressures which may be realised in some engines of good modern construction, with the figures for corresponding compression adopted in practice.

Fuel.	Compression in lbs. per square inch.	Mean Pressure in lbs. per square inch.
Blast Furnace Gas	170 to 200	70
Producer Gas	140 to 170	70
Natural Gas (800 to 900 B.Th.U. per cubic foot)	140 to 160	80
Coke Oven Gas	130 to 140	80
Town Gas (normal)	110 to 140	85
Refined Petroleum	40 to 70	60
„ „ with water injection	70 to 85	65
Spirit and Benzol (industrial engines)	70 to 85	70
Alcohol (industrial engines)	100 to 130	85
Spirit and Benzol (automobile engines)	85 to 100	85—100
Alcohol (automobile engines)	110 to 160	100—110

In order to examine the different parts, which together compose a gas engine, in a clear and methodical manner, they will be classified as follows:—

A. **Stationary Details.**—Frames; Foundation bolts; Main bearings: Outer bearings; Side shaft bearings; Cylinders; Liners and Water jackets; Cylinder covers; Stuffing-boxes; Breech ends and explosion chambers; Joints; Water circulation; Valve boxes.

B. **Moving Parts.**—Fly-wheels and pulleys; Crank shafts; Pistons; Connecting rods; Piston rods; Crossheads.

C. **Governing and Operating Details.**—Valves; Half-speed shafts; Gear wheels; Rollers; Bearings and supports; Valve gear; Pivots; Cams; Levers; Eccentrics; Rolling paths; Trip gear; Governors; Springs.

A. Stationary Details.—Frames.

Formula.—The complex form of frames do not permit any general formula to be used for the calculation of their strength. It is

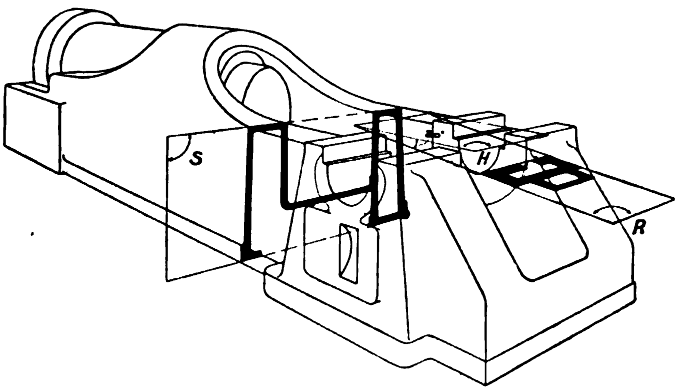


FIG. XII.—1. Design of Frame Casting.

necessary to give each section such form as will correspond to a maximum of resistance.

Fig. XII.—1, indicates in perspective a frame or base casting with sections of two important portions R and S. The right-hand side of the frame forming a separate chamber is used as an air suction chamber.

Section R.—This is made in the horizontal plane of the back portion of the crank shaft bearings. This portion should be strengthened by extending the frame towards the front so as to form a buttress or strut. It must, in fact, compensate the loss of resistance due to the separation caused by the accommodation for the main bearings.

If E is the total explosive force acting on the piston at the rate of 430 lbs. per square inch, $\frac{E}{2}$ will be the load on each bearing, acting horizontally.

To produce a fracture along the horizontal section, the latter is calculated by taking into account the shearing stress by the formula

$$\frac{E}{2S} = R$$

in which :—
E = the total force acting on the piston in lbs.
S = the area of the metal in square inches.
R = the resistance of cast-iron to shearing in lbs. per square inch.

But the fracture, if produced, would not follow the horizontal plane but that of R (Fig. XII.—1), making an angle with the horizontal plane, which Güldner estimates as 20° in the case of a bearing inclined about 50° towards the interior.

To calculate the section the following must be considered :—

1. The *tensile stress* F acting at the centre of gravity of the section, expressed by :—

$$F = \frac{E}{2} \sin \beta,$$

and if n is the area of fracture in square inches, the tensile stress will be

$$Rt = \frac{E \sin \beta}{2n}.$$

2. The *bending moment*, expressed by

$$Mt = \frac{El}{2}$$

l being the distance between the horizontal plane passing by the axis of the forces (centre of cylinder).

The resistance to bending is given by the equation :—

$$Rf = \frac{\frac{El}{2}}{\Sigma i}$$

Σi being the sum of the moments of inertia of the portion under consideration.

$$Rf + Rt = \frac{\frac{El}{2}}{\Sigma i} + \frac{\frac{E}{2} \sin \beta}{n} = R$$

should not exceed 2,800 lbs. per square inch.

Usually the thickness given, determined either by the form of the frame or by the foundry limitations, is such that R is in the neighbourhood of 1,400 lbs. per square inch.

Section S.—This is made at about halfway between the bearings and the beginning of the cylinder. The calculations are made by taking into account the tensile and bending stresses.

Tension.—If $S =$ the total area of the section in square inches, the tensile stress will be

$$\frac{E}{S} = Rt \text{ foot lbs. per square inch,}$$

E being the total load upon the piston in lbs.

Bending.—If γ_1 is the distance between the neutral axis of the section and the power axis (axis of cylinder), the bending moment will be :—

$$Mf = E\gamma_1.$$

The tensile stress for the most remote portion of the metal is given by the equation :—

$$\frac{E \times \gamma_1 \times \gamma_2}{\Sigma i} = R$$

in which,

$\Sigma i =$ the moment of total inertia with respect to the neutral axis.
$\gamma_2 =$ the distance between the neutral axis to the most remote fibre.
$R = Rt + Rf$ should be lower than 2,800 lbs. per square inch.

General Form.—For horizontal single-acting engines the most recommendable form, in the author's opinion, is that which is similar to the German models, a frame with a large base area supporting the cylinder for practically the whole of its length and forming one piece with the water-jacket (Fig. XII.—2). This arrangement of cylinder and frame constitutes a strong and compact casting which opposes all tendency to bend at the moment of explosion. The author also advocates the arrangement, shown in Fig. XII.—3, in which all the side shaft brackets are fitted upon the frame itself, leaving the breech end as independent as possible.

For purposes of cleanliness a groove should be arranged at the base of the frame to receive the excess of lubricating oil (Figs. XII.—2 and 5), a small drain cock being fitted at the lowest point or a communication made with the trough between the bearings within which the crank revolves; the same arrangement being carried out to all parts forming a basin where oil can accumulate.

For suction producer-gas engines the air chamber can be arranged without inconvenience in the frame. With town gas it is not permissible, owing to the liability of the frames to become fractured by the ignition of explosive mixture that may be accidentally formed. The risk does not exist with suction gas engines, as the mixture is produced by the suction of the engine itself. Nevertheless, the frame should be kept low and strong so as not to present large flat areas against which abnormal pressures might act.

The number and area of air apertures should be sufficient to avoid

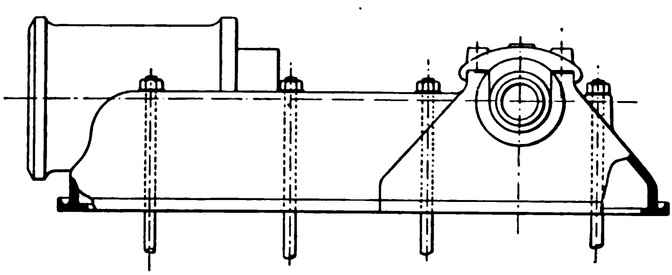


FIG. XII.—2. German form of Frame Casting.

all whistling or hissing during suction, and for the same reason the walls should be smooth and without seams or blisters.

The general form of frame which has been referred to as the German type has been applied to the largest powers reached from a single cylinder by the Maschinenfabrik Augsburg Nürnberg (Fig. XII.—4).

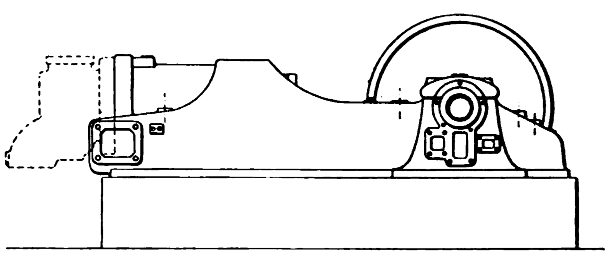


FIG. XII.—3. Frame Casting with provision for Cam Shaft Brackets.

This design was used in a blast furnace gas engine developing 700 H.P. at a speed of 90 revolutions per minute, with piston 52 inches diameter and stroke of 55 inches. The engine was fitted with two admission valves and two exhaust valves in simultaneously operated pairs.

The two separately-cast side pieces forming the supporting beams were tied together by a central tie rod. Their total height was $94\frac{1}{2}$ inches, the breadth at the base $35\frac{1}{2}$ inches, and at the upper portion $23\frac{1}{2}$ inches.

The construction of single-acting engines of this size has been definitely abandoned, and to-day the output of 150 or 200 H.P. maximum is hardly ever exceeded for engines of this type.

Fig. XII.—5 represents a section through the frame of a single-acting gas engine made by the Société Winterthur. M. Reichenbach has

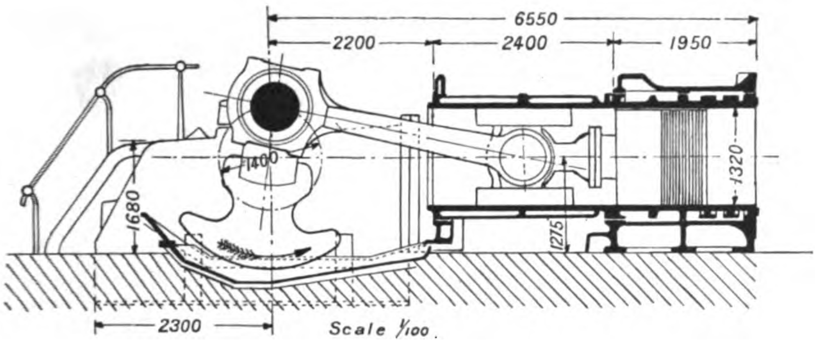


FIG. XII.—4. Frame of 700 H.P. Nürnberg Engine.

designed a type of engine—constructed by the Union Society of Dortmund and the Görlitzer M. A. G. of Görlitz—which enables the frame to be used indifferently for either single or double-acting or tandem engine (Fig. XII.—6). In the single-acting and the tandem

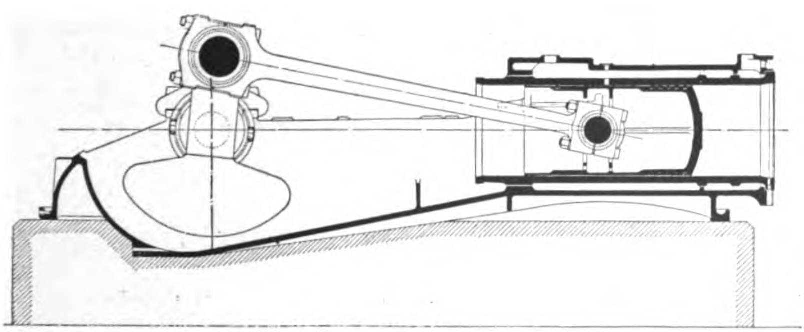


FIG. XII.—5. Frame of Winterthur single-acting Engine.

arrangements the frame casing serves as the cylinder water-jacket ; in double-acting engines it is used as the crosshead guide.

English and American makers until recently have remained faithful to less imposing forms of frames than the German. Some further remarks will be made later on when dealing with cylinders and jackets, but, as has been already mentioned, in all countries there seems to be a very marked tendency to adopt the German type of frame. Several of the more important firms in America and England have consulted the author with regard to the conversion of their existing engines,

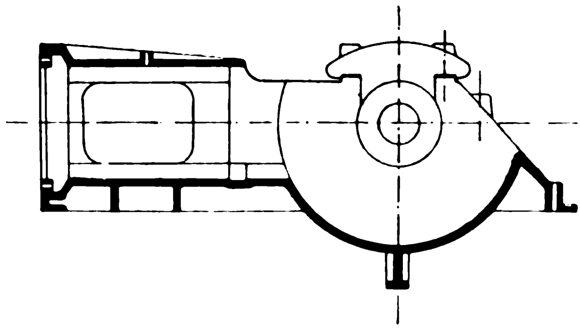


FIG. XII.—6. Frame suitable for various Cylinder Combinations (Reichenbach).

and, as a result of serious consideration of this question, have decided to adopt this type of frame.

It has been already mentioned that the frame, forming one piece with the cylinder water-jacket, presents the advantages of being very rigid and obviates the objectionable overhang.

The partisans of jackets independent of the frame, in support of

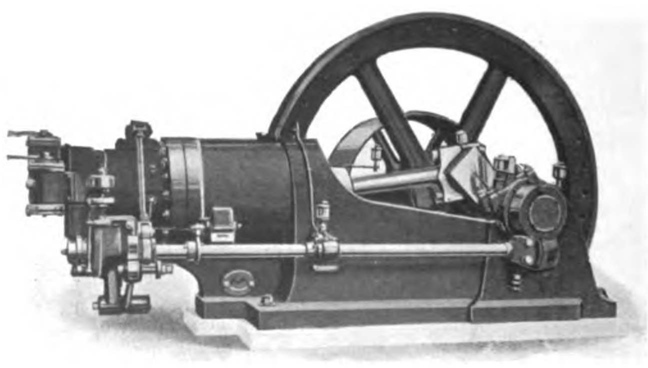


FIG. XII.—7. Supported Cylinder (Dudbridge).

their preferences, put forward the specious argument of the great ease of renewal of this jacket should it at any time be fractured by frost. But it must be conceded that an engine is not designed to be exposed full of water to frost. Such accidents were frequent when gas engines first began to be employed and when those placed in charge were incompetent with respect to the attention necessary, but to-day, with better education of the attendants, such fractures rarely occur.

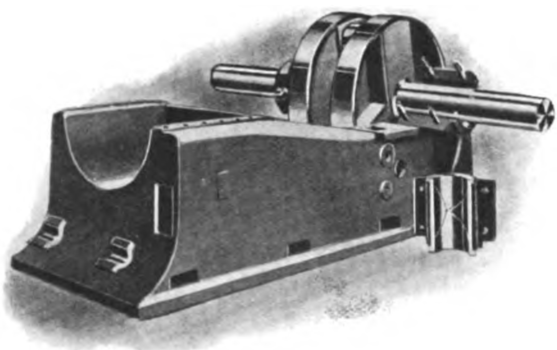


FIG. XII.—8. Cylinder supported on Side Bearers (Foos).

In certain cases, water jackets independent of the engine frame may prejudice the stability and complicate the appearance of the engine. Some makers, however, remain partisans of this separate casing, which, if it is subject to criticism, does not always present grave

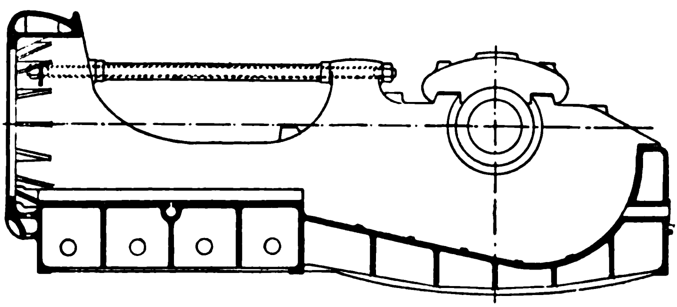


FIG. XII.—9. Nürnberg Frame with Tie-bar.

inconveniences when applied to single-acting engines of lower power than from 20 to 30 H.P.

The Dudbridge Ironworks Co., Ltd., for some of their engines, shape the frame with a rear bracket, all in one piece with the water jacket, as shown in Fig. XII.—7.

Fig. XII.—8 represents the frame of the Foos engine, with bearings inclined towards the rear, fitted with two longitudinal bearers to receive the cylinder.

In large double-acting engines the frame usually differs from those used for single-acting engines on account of the size of the parts and the necessity of providing for the disposition of the valve gear in a rational manner. The engine is nearly always made of different members joined up together, some of the segments occasionally reaching considerable dimensions and weights. The frame con-

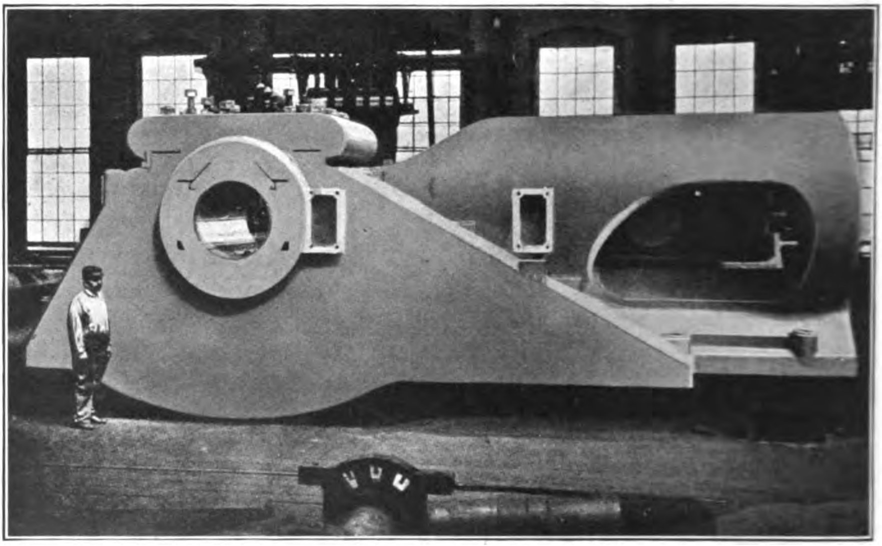


FIG. XII.—10. Frame of 5,400 H.P. Engine (Snow Steam Pump Works).

structed by the Maschinenfabrik Augsburg Nürnberg for the engines supplied to the steelworks at Rombach weighs over 26 tons.

Fig. XII.—9 is a section of a frame constructed by this firm. The openings arranged in the sides to give access to the slides and cross-head are strengthened by a powerful steel tie-rod.

Fig. XII.—10 shows the frame of the 5,400 H.P. engine of the Snow Steam Pump Works of Buffalo, weighing 77 tons.

The frame of the large Cockerill engines is formed by two strong cast-iron beams running the whole length of the engine, and between which the cylinders are fixed. They terminate in the crosshead guides and the crank shaft bearings (Figs. XII.—11 and 12).

The Ehrhardt & Sehmer engines have a strong frame, with a large bearing surface, fixed to the masonry by a series of uniformly spaced bolts (Fig. XII.—13). This frame is furnished with a crosshead guide, the latter being machined at the same time as the facing for the cylinder so as to ensure absolute concordance in erection.

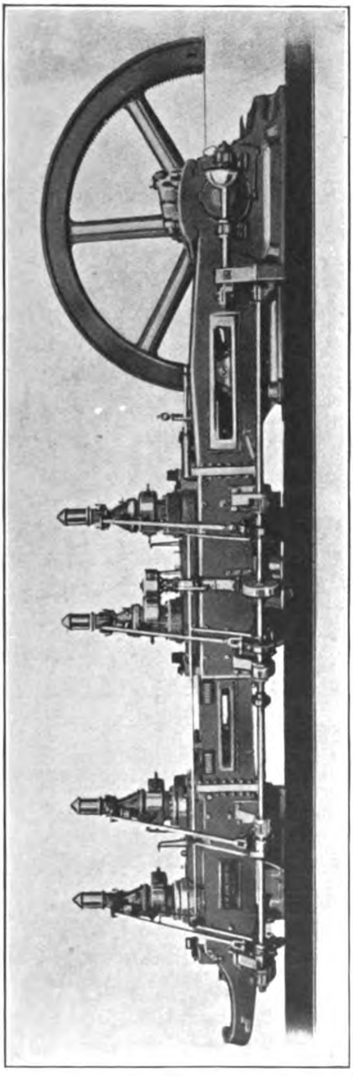


FIG. XII.—11. Cockerill Girder-type Frame.

The Haniel et Lueg frame bears upon the foundation throughout its length. It is hollow and fitted with internal ribs. Two side openings are arranged on each of the side frames near the site for the first cylinder to allow the front stuffing-boxes to be inspected, the openings being closed during work by a wrought-iron slide. The bore of the portion which joins on to the cylinder is concentric to the piston crosshead guides.

In the tandem engines the intermediate frame is similarly hollow

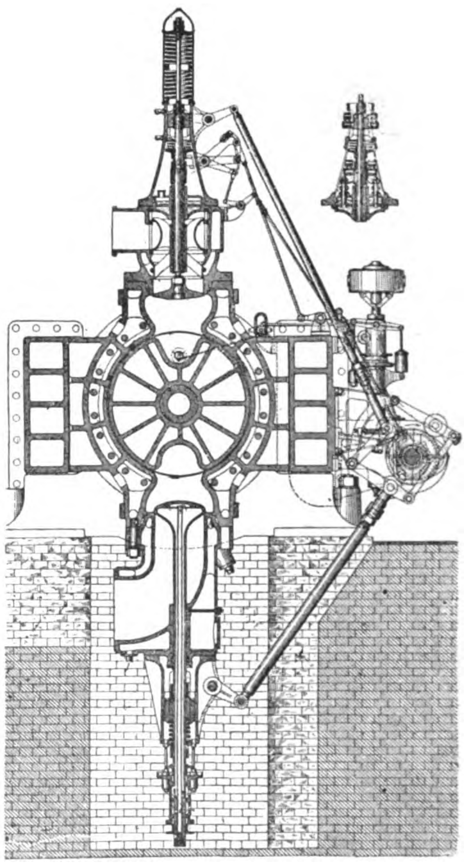


Fig. XII.—12. Cockerill Girder-type Frame.

with a large number of internal ribs designed to give the maximum of rigidity with least weight. A lateral opening in this portion permits the erection of the piston rods and inspection of the middle crosshead. This opening is strengthened by two strong steel rods.

Fig. XII.—14 is a transverse section of the frame of the engine built by the Elsassische Maschinenbau of Mulhausen. It is of the two symmetrical beams type, strongly braced by the crosshead slipper guide and by the oil trough. The cylindrical guide is bored at the

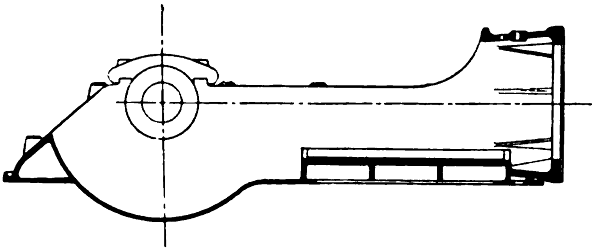


FIG. XII.—13. Ehrhardt & Sehmer Frame Casting.

same time as the flange for the cylinder attachment. It is cooled by water circulation and fitted with means of adjustment.

FOUNDATION BOLTS.

Dimensions.—Only experimental data exists for the dimensions of foundation bolts. Their number and position depends upon the length and shape of the base. To strengthen the base at the crank shaft

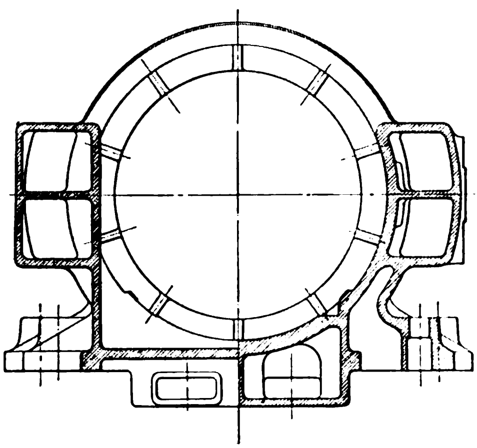


FIG. XII.—14. Elsassische Frame Casting.

bearings it is a good plan to place a foundation bolt on either side in their immediate vicinity (Figs. XII.—2 and 25).

The following table gives the dimensions which should be employed for single-cylinder engines upon the basis of using from six to nine bolts, the ninth bolt being placed in the middle of the front portion of the frame.

DIMENSIONS OF FOUNDATION BOLTS.

B.H.P.	15.	25.	35.	50.	75.	100.	150.
Diameter of bolts, inches	1	$1\frac{1}{8}$	$1\frac{1}{4}$	$1\frac{3}{8}$	$1\frac{1}{2}$	$1\frac{1}{2}$	2
Number " "	6	6	6	6	8	9	9

The holding-down bolts should preferably pass through the whole depth of the frame so as to obtain the best bond with the foundation. This arrangement increases the rigidity of the frame and, for an equal resistance, permits lighter sections to be used. It is much to be

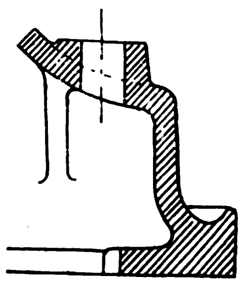


FIG. XII.—15. Boss for Foundation Bolt, with Stiffening Web.

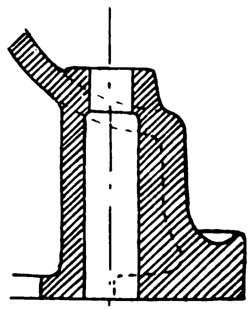


FIG. XII.—16. Boss for Foundation Bolt, with Solid Sheathing.

preferred to that in which the bolts are passed through bosses formed in the bottom of the frame casting.

The bolt holes should be strengthened at their upper part by a rib (Fig. XII.—15) or by a solid sheathing wider towards the base, as in Fig. XII.—16. The nut should rest upon a perfectly plane surface with a washer intervening.

It is a good plan to arrange the bolts in two parallel lines, and to make them flush at the same height so as to be all of the same length, a matter which simplifies the construction of the foundation. This condition can only be realised when the base is designed accordingly.

MAIN BEARINGS.

The frame of a gas engine comprises two main bearings designed to support the shaft. These bearings are arranged horizontally in the majority of cases. Bearings inclined towards the cylinder are only employed for the smaller engines. Those inclined away from the cylinder are wrongly designed and are no longer used.

The crank shaft bearings should be fitted with gun-metal or anti-friction metal brasses of a type which allows play to be easily taken

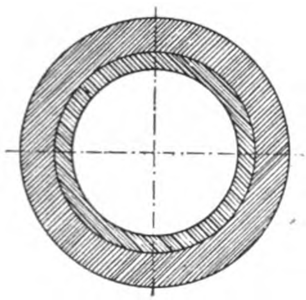


FIG. XII.—17. White-metal lined Bearing before cooling.

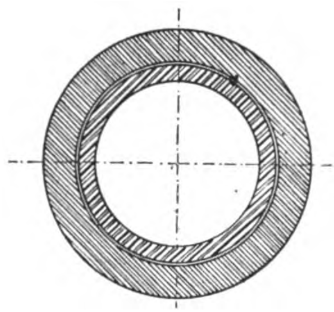


FIG. XII.—18. White-metal lined Bearing after cooling.

up. The brasses are fitted with a key or dowel peg to prevent their rotation.

Gun-metal or phosphor bronze is used for brasses of small dimensions. For others either shells of cast iron, steel, or delta metal are used, lined with white metal. The latter should project at the edges with respect to the shells in order to provide a side supporting surface for the clearances of the drip rings (Fig. XII.—23, p. 292).

The construction of these bearings necessitates certain precautions.

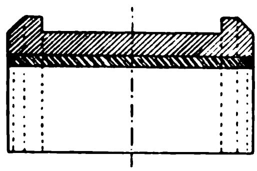


FIG. XII.—19. White-metal lined Bearing after cooling.

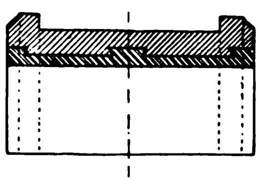


FIG. XII.—20. White-metal lined Bearing, with Dovetailed Grooves.

The coefficient of contraction of the white metal is two or three times greater than that of iron or steel, with the result that after cooling the surfaces do not remain in contact in the transverse direction as shown in Fig. XII.—17, but are slightly separated as is to be seen in Fig. XII.—18. In the longitudinal direction they similarly produce a shrinking at each end of the journal (Fig. XII.—19).

To avoid such shrinking, dovetailed notches are provided, as shown in Fig. XII.—20; but during cooling of the metal the back surfaces of these notches take a concave form (Fig. XII.—21). Little by little, the oil penetrates into the free space left, and as its coefficient of conductibility is about $\frac{1}{200}$ that of the metal, it may set up abnormal heating of the white metal and cause its fusion.

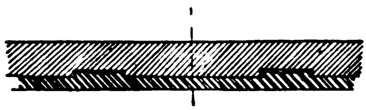


FIG. XII.—21. White-metal lined Bearing, showing effect of Contraction in Grooves.

The arrangement for taking up wear in single-acting engines is placed at the side of the cylinder so that it may not be directly submitted to the force of the explosions.

For engines exceeding 60 to 75 H.P. it is indispensable to ensure lubrication by means of revolving rings or chains, and in the larger engines the oil is forced into the bearings under pressure.

Fig. XII.—22 represents a prospective view of a ring bearing and shows the ring resting on the shaft itself, a notch *E* being cut away in the brass. This notch should only exist in the upper part of the top half brass, so that this may be all in one piece and not cut into two.

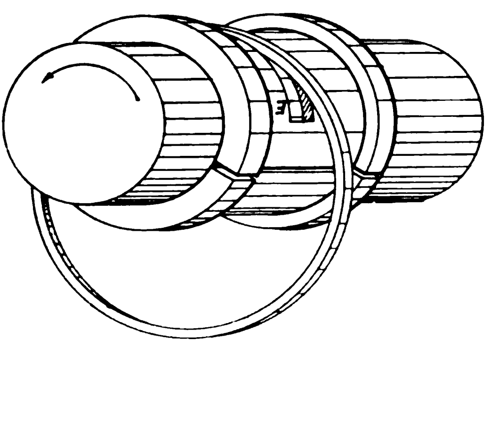


FIG. XII.—22. Shaft in Bearing, with Oiling Ring.

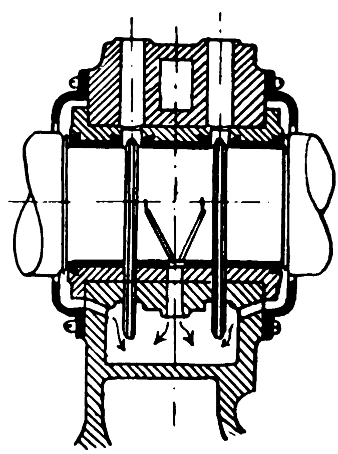


FIG. XII.—23. Section of main Bearing, with two Oiling Rings.

The ring should be of sufficient diameter to revolve freely under these conditions. The interior face of the ring in contact with the shaft should be in the form of a groove rather than a plane surface, as it will then take up more oil and is more likely to continue in rotation.

For engines of 75 to 100 H.P. two rings are employed. It is known that, unless precautions are taken by cutting grooves in the middle of

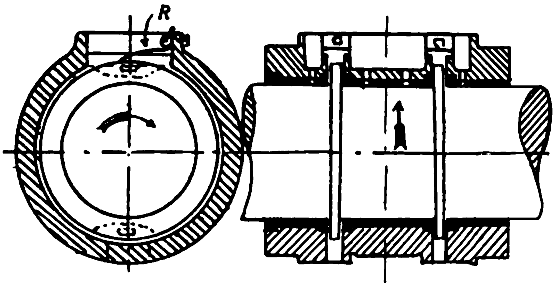


FIG. XII.—24. Bearing for Shaft with two fixed Oiling Rings.

the bottom brass and a vertical passage leading to the oil bath, the oil is likely to flow badly between the two rings (Fig. XII.—23).

Instead of a movable ring, which is likely to cease revolving, some constructors prefer to use fixed rings upon the engine shaft, bathing in the oil and rubbing against a spring *R*, in the upper part of the bearing, in the form of a spoon which scoops the oil from the ring

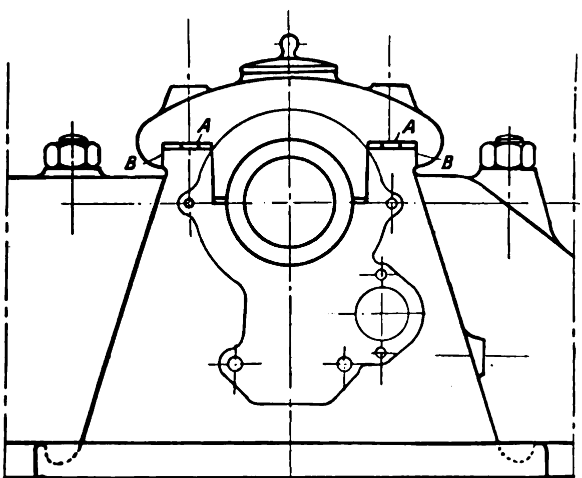


FIG. XII.—25. Bearing with Socket Cap.

and diverts it to a distributing trough (Fig. XII.—24). This fixed ring is formed of two pieces joined together.

The grooves in the brasses should be sufficiently large to permit lateral movement of the engine shaft. The play of each side should be from $\frac{1}{8}$ to $\frac{3}{16}$ inch. In this arrangement the brasses present the inconvenience of being in several pieces, and therefore it is a

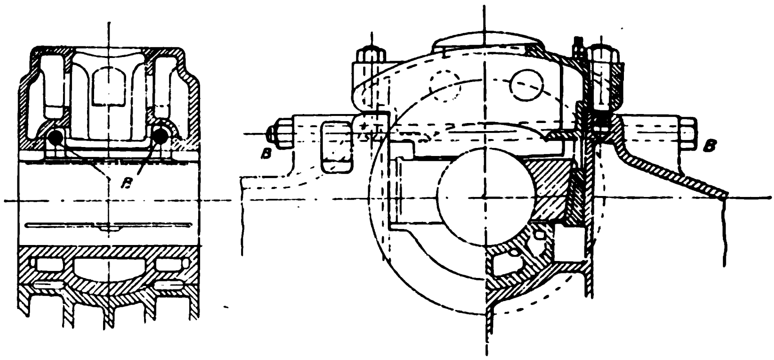


FIG. XII.—26. Transverse Section. FIG. XII.—27. Elevation.
Main Bearing Construction of 5,400 H.P. Engine (Snow Steam Pump Works).

matter of greater difficulty to rigidly maintain the different sections as a whole.

In Figs. XII.—23 and 28, the bearings are fitted with side oil catchers within which the drip rings projecting from the shaft revolve. These oil catchers are cast with the frame in small engines as in

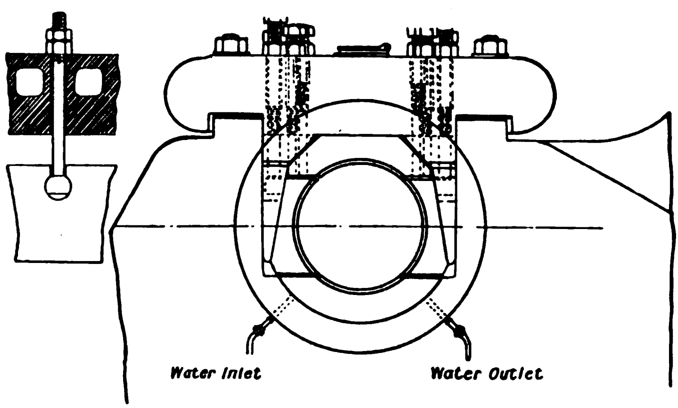


FIG. XII.—28. Four-part Bearing, with Wedge Adjustment (Snow Steam Pump Works).

Fig. XII.—28, and attached by screws as separate pieces in large engines (see Fig. XII.—23). In the latter case they do not involve the separation between the two halves of the bearing, which might allow oil to escape.

An opening must be provided in the external face of the bearings by means of which the internal oil bath can be cleaned out. On the cover of this aperture an oil gauge glass can be readily fitted to show the level of the oil, and a drain cock should also be furnished.

Some makers make use of the plain cap, which is simpler and requires least adjustment. In the author's opinion it is preferable to construct the bearings as shown in Fig. XII.—25, with a socket cap. The internal parts are fitted but not jointed, whilst the parts *B* that strengthen the bearing are adjusted to dimensions with a little overlap to prevent them becoming wedged during dismantling. This overlap is from 0·2 to 0·1 of the diameter of the shaft. The space *A* (Fig. XII.—25) permits the wear of bearings to be taken up.

If the weight of the bearing cap exceeds a hundredweight it is necessary to provide fixed lugs or screwed holes in which a hook can be inserted for easier handling.

The covers are secured by two or four bolts with shoulders, screwed into the frame and furnished with a turned washer and two lock-nuts. Sometimes the bolts used are tee-headed, fitting into a socket cast in the frame, but this arrangement is less mechanical than the preceding.

In their very large engines the Snow Steam Pump Co., of Buffalo, strengthen the upper part of the bearings by bolts *B* (Figs. XII.—26 and 27) arranged horizontally and passing through the cap and the bearing itself.

When ring bearings are used it is easier to fix the caps by four bolts instead of two, in view of being able to arrange the oil bath between the bolts.

DIMENSIONS OF CAP BOLTS.

B.H.P.	15	25	35	50	75	100	150
Number of bolts	2	2	4	4	4	4	4
Diameter in inches	$\frac{7}{8}$	1	1	1	$1\frac{1}{8}$	$1\frac{1}{4}$	$1\frac{1}{4}$

Kynoch, Ltd., of Birmingham, for their engines of more than 80 H.P. employ brasses in four pieces, adjustable in horizontal and vertical directions, and removable without lifting the shaft.

Scheben & Krudewig use long brasses in red brass. Dingler's brasses are interchangeable, in cast-steel lined with alloy. Lubrication of the main shaft bearings is by means of a chain; the side shaft bearings are fitted for ring oilers.

In the large engines of the Snow Steam Pump Works the bearing consists of four pieces of cast iron lined with white metal (Fig. XII.—28).

The lower part of the brass is let in a seat bored in the frame, whilst the two side pieces are adjustable by means of wedges and tightening screws. The lower shell may be water-cooled in case unforeseen heating is produced.

OUTER BEARINGS.

Modern gas engines generally comprise a third bearing supported by brickwork or by a cast-iron standard. It is important that such brickwork or concrete pier should be built as part and parcel of

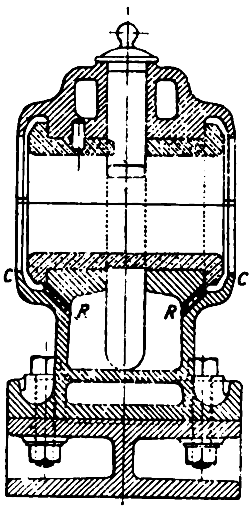


FIG. XII.—29. Outer Bearing and Sole Plate.


FIG. XII.—30. Hornsby Outer Bearing and special Oiling Device.

the engine foundation. The bearing should be fitted upon an independent base or soleplate with oil catchers and keyed to the brickwork with long foundation bolts, two for engines up to 50 to 75 H.P., and four for higher powers.

The bearing and the soleplate should be connected by bolts. In order to allow slight movement in all directions during erection, the holes arranged in the soleplate and in the bearing should be cast oval in two directions at right angles to each other. The dimensions of this bearing, pedestal and diameter, will be given in the paragraph relating to crank shafts. The form and arrangement will be the same as those for the main bearings, but the dimensions may be somewhat less, seeing that it will only support a part of the weight of the fly-wheel, of the pulley, and of the crank shaft as well as of the work transmitted.

Fig. XII.—29 represents an outer or outboard bearing with gun-metal brasses, suitable for engines of less than 50 H.P. The oil catchers are shown at *C*, being circular grooves cast with the pedestal itself. Two passages *R* permit the oil thus collected to return to the central trough.

R. Hornsby & Sons, of Grantham and Stockport, sometimes employ the arrangement shown in Fig. XII.—30 for the outer bearing. It comprises a set of adjusting screws to level and adjust the bearing upon the cast-iron soleplate. Continuous lubrication is effected by means of a small oil reservoir; the oil overflows in a side groove,

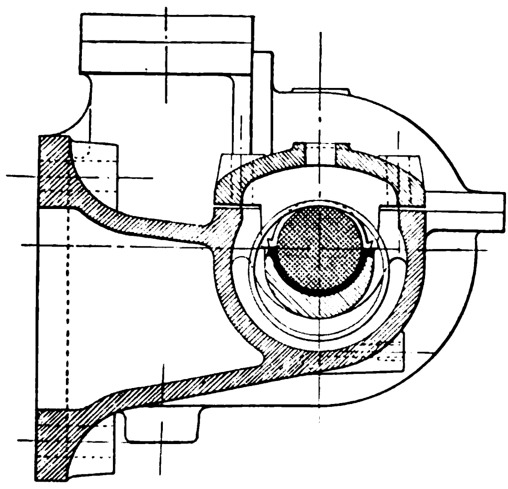


FIG. XII.—31. Cam-shaft Bracket and Ring Oiling Bearing.

and is returned to the reservoir by a small centrifugal pump driven by a chain from the crank shaft.

In large engines upon the shaft of which a dynamo is mounted, particular care is demanded in the construction of the outer bearings. Long brasses must be used having a spherical part, and the shells must be placed in the direction of the line of elasticity. This outer bearing should also be fitted some mils higher than the two engine bearings (1 mil = 0·001 inch).

SIDE SHAFT BEARINGS.

At least two bearings are fixed upon prepared facings, either on the frame or on the breech end, making a solid and firm support. The length of the bearings is from one-and-a-half to twice the diameter of the side shaft.

The first bearing is fixed as near as possible to the skew gear wheels so as to reduce the overhang of the shaft. It partly projects into the gear wheel box, which serves at the same time as an oil bath.

For engines over about 100 H.P. it is a good plan to support the side shaft at the centre by a third bearing. In very large engines it is even wise to place a bearing on each side of the gear wheels.

The second bearing is placed as close as possible to the cams or eccentrics operating the valve gear to reduce the bending stresses upon the shaft. For simplicity it is advantageous to cast the governor bracket and gear wheel case with this bearing.

English makers usually make these bearings solid without gun-metal liners, even though the length of bearing is necessarily increased. Steel and iron answer very well as regards friction of surfaces. Lubrication is effected by means of a small oil cup or wick.

Other makers use gun-metal brasses in two pieces to allow for taking up wear. The author advocates the type of ring bearing shown in Fig. XII.—31 in cast iron for 50 H.P. and under, in white metal or bronze for higher powers.

Frictional Areas.—The frictional areas of the brasses of the different bearings should be calculated to prevent all kinds of heating or undue wear. The loads per unit of surface given below may be applied as limit bases, reckoning upon an initial explosion pressure of 430 lbs. per square inch upon the piston area.

	Lbs. per square inch.
For the piston pin	1,700 to 1,850
crank shaft bearings . . .	500 to 550
crank pin	1,700
pins for valve gear levers and rollers	550 to 650

These loads are on the basis of the projected areas (diameter × length).

Cylinders—Single-acting Engines.

General Form.—It is generally accepted that a gas engine cylinder should be formed of a separate liner made of specially hard cast iron of great strength and tenacity, and a water-jacket independent of the frame or forming one piece with it. There are very few exceptions to this rule in modern horizontal gas engines.

Some constructors have provided circular ribs at the external surface of the cylinder liner with the object of quickening the cooling. The author's opinion is that this provision is irrational, seeing that, though it is necessary to encourage ample water circulation to the combustion chamber to guard against the explosion temperature acting prejudicially to its strength, it is incumbent to cool that portion of the cylinder within which expansion occurs very slightly indeed to obtain proper working conditions.

A sufficient space should be left between the liner and the cylinder casing to diminish the influence of incrustations and to allow for cleaning. This space is usually equal to twice the thickness of the cast-iron casing.

In addition to the effects of expansion due to the temperature variations, either in the interior of the liner or in the interior of the casing, stresses are set up in the liner itself resulting from the internal wall in contact with the hot gases being at a much higher temperature than the water-cooled external wall. This causes the internal fibres to be compressed and the external fibres to be in tension, a condition which, for great temperature differences, might be serious. Apart from this, tensile stresses exist in the horizontal direction resulting from the force of the explosion acting against the back of the cylinder and reacting upon the liner itself. Finally, in some cases, the separate portions which constitute the cylinder are tied together by means of bolts, the tightening of which sets up considerable tension.

Experience has shown that it is advisable, whenever possible, to provide large fillets to couple the pieces so as to permit free movement and to avoid making ribs which interfere with expansion without preventing it, and which, in consequence, leads to the formation of cracks.

Dimensions.

Thickness of Liner.—The following formula is applied :—

$$e = \frac{PD}{2R}$$

in which

e = Thickness of liner in inches.
P = Pressure per square inch = 285 lbs. per square inch.
D = Internal diameter of liner in inches.
R = Practical coefficient of resistance of cast-iron to tension = 2,850 lbs. per square inch.

This gives—

$$e = \frac{D}{20}$$

For P, 285 lbs. per square inch is taken instead of 430 lbs. per square inch, because the latter, when acting upon the liner, has already been appreciably reduced by expansion.

In this formula fracture in the direction of the force is considered, because in the direction at right angles the work on the metal is only one-half as much.

To provide for re-boring the liner at some future time the formula should be:—

$$e = 1{\cdot}2 \times \frac{D}{20}$$

This formula does not take into account the bending that may occur in the liner by the vertical reaction due to the obliquity of the connecting-rod at about half-stroke of the piston. Also for engines beyond 50 to 75 H.P., and particularly for long-stroke engines, it is a good plan to cast a semi-circular support in the frame to bear against the cylinder towards the middle of the water-jacket.

Similarly it should be noted that for engines with cylinders less than about 12 inches internal diameter the liner should not be less than $\frac{9}{16}$ inch thick to ensure good working.

Thickness of Casing.—The thickness of casing $e_1 = 0{\cdot}6$ to $0{\cdot}8e$, e being the thickness of the liner.

Bolts or Studs of the Cylinder Joint with the Breech End or with the Cylinder Cover.—Up to 200 H.P. sizes the number of bolts is determined by the equation :—

$$n = 0{\cdot}5 \text{ to } 0{\cdot}625\,D.$$

Some constructors take $n = 0{\cdot}75\,D$, in which

n = number of bolts.
D = diameter of cylinder in inches.

For engines upwards of 15 to 20 H.P. the pitch of the bolts should not exceed 5·5 times their diameter.

The effective area is taken as that of the bottom of the thread This is equal to about 0·7 of that of the bolt, implying that the diameter of the two sections is in the ratio 0·84 to 1.

To calculate the diameters the following equation is used :—

$$d = D \sqrt{\frac{430}{nR}}$$

in which—

d = effective diameter of bolt expressed in inches.
D = diameter of cylinder expressed in inches.
n = number of bolts.
R = practical coefficient of resistance of steel to tension.
= 7,200 for *small* bolts (to take into account the additional tension due to tightening).
= 7,900 for *medium* bolts.
= 8,600 for *large* bolts
430 = initial explosion pressure in lbs. per square inch.

d = outer diameter of bolt $= \dfrac{d}{0{\cdot}84}$

The following table has been constructed from the above formula:—

DIMENSIONS OF CYLINDER BOLTS.

Number.	Diameter.	
	Effective.	External.
3	0·141 D	0·180 D
4	0·122 D	0·145 D
5	0·104 D	0·124 D
6	0·099 D	0·118 D
8	0·082 D	0·098 D
10	0·071 D	0·084 D
12	0·064 D	0·076 D
14	0·060 D	0·071 D
16	0·056 D	0·067 D

Many constructors adopt stronger dimensions than those given, but it must be taken into account that the above are based upon the hypothesis that the breech ends are entirely independent of the valve-operating mechanism, and, consequently, are only called upon to resist minimum side strains.

Flanges or Collars.—The bearing surface of the nuts should be carefully prepared. The breadth of the flanges should be 2·5 times the diameter of the bolts. Their thickness, if solid, should be 1·5 to 2·0 times the diameter of the bolts.

The bolts and studs should be made from mild steel of the best quality. The threaded portion of the stud screwed into the metal should have a length at least 1·5 times the external diameter.

Chamfers.—At each of the two ends of the liner, a chamfer or counterbore should be turned, and the piston rings should be arranged in such a manner that, when travelling backwards and forwards, the front ring and the last of the series of rings at the rear overlap the commencement of these chamfers at each extremity of the stroke.

This rule is particularly observed in the Tangye and Winterthur engines, as well as those of other makers. Some, on the contrary, neglect this detail, and the friction of the rings wear the liner, and after a time the latter becomes rutted, as shown in exaggeration in Fig. XII.—32.

The piston ring grooves become deformed at the sides, and dirt, sooner or later, enters under the rings. The latter are then prevented

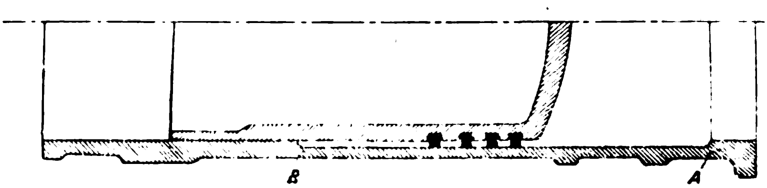


FIG. XII.—32. Wear of Liner due to Piston Movement.

from closing in, with the result that it is impossible to remove the piston from the cylinder. On the other hand, supposing that this accident should be avoided, the renewal of the rings could only be done effectually by re-turning and consequently enlarging the grooves. This involves fitting wider rings, which butt against the portions *A* and *B* (Fig. XII.—32), causing a very disagreeable noise and probably eventual breakage.

It is to avoid these inconveniences that a ring in the front of the piston is employed and that counterbores are turned at the ends of the liner. These chamfers should similarly be provided in the case of double-acting engines.

Joints.—The arrangement of cylinder most widely adopted in European practice for single-acting engines is represented by Fig. XII.—5. It consists of an internal liner, which is inserted into the casing from the rear, bearing in front in a cylindrically-turned portion, and, at the back, against a fillet in the casing, terminating a little behind the vertical face of the latter, so as to ensure the tightness of the whole upon the erection of the breech end. At the lower back part of the liner an oil blow-off valve is fitted, as explained in connection with Figs. X.—36 and X.—43.

This is the type of cylinder that the author advocates as the best.

Fig. XII.—33 shows the manner in which a good joint is made between the liner *C* and the water-jacket casing *E* for engines fitted with a separate breech end. The cylindrically-turned part, with the shoulder *A* which forms the joint between the jacket and the liner, are made gastight by a simple coating of red lead, whilst the joint *D*, which resists the internal pressures of the cylinder, is made with asbestos or annealed copper.

The coupling of the breech end with the water-jacket *E*, forming

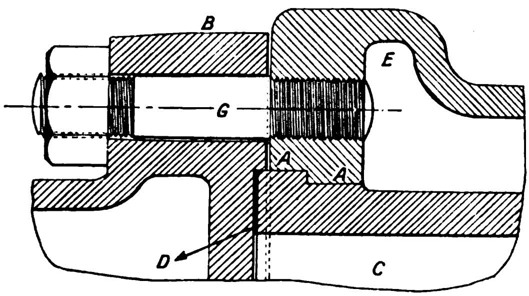


FIG. XII.—33. Detail of Joint between Liner, Jacket, and Breech Casting.

part of the frame, is effected by means of a series of studs *G*, arranged around the circumference of the flange *B*. It should be observed that all the pressure at this coupling acts against the joint *D*; therefore, to reduce torsional strains, the coupling studs should be placed as closely as possible to this point.

The arrangement in which the liner is fitted with a large flange which is flush with and is coupled between those of the jacket and breech end is not advised, because of the extra thickness of the metal making the efficient cooling by water difficult.

Fig. XII.—34 represents the method of fixing the cylinder liner and separate breech end, for small engines in which the water-jacket *M* forms part of the base.

The liner is first of all fixed to the breech end *A* with the joint *a* of asbestos or annealed copper. The tightening is by means of bolts *B*, of which the nuts are placed outside the water-jacket of the combustion chamber. This done, the liner is inserted within the water casing from the back. An indiarubber joint ring *D* sufficiently ensures tightness of the casing and allows free expansion of the liner in the horizontal direction. The whole is fixed to the frame by the nuts E

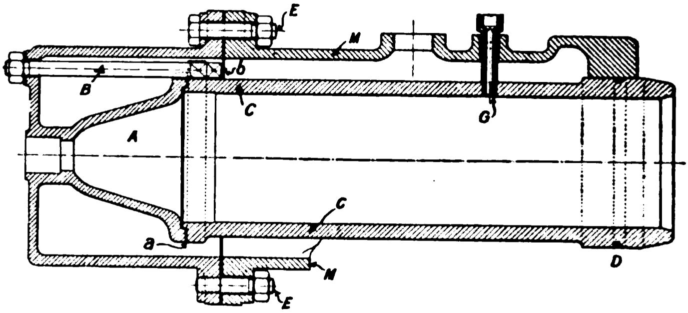


FIG. XII.—34. Detail of Liner Joint, with Bolts through Breech Casting.

on the studs and bolts of the breech end. The tightness of the outer joint is ensured by means of an asbestos ring.

Examples.—Other arrangements applied in France, America, and in England are very numerous, but only some of the better known will be referred to.

The Charon engine, which was at one time a very well known

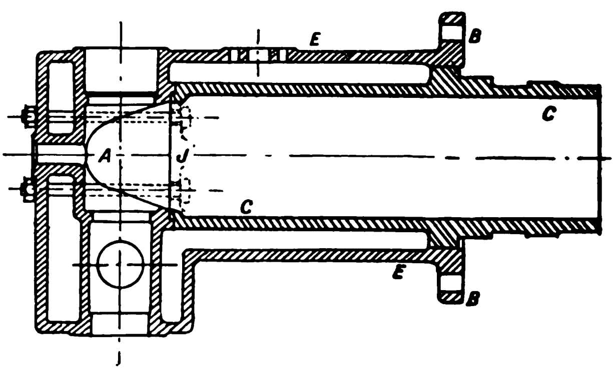


FIG. XII.—35. Liner in combined Combustion-chamber and Water-jacket.

French type and which gave very favourable results under test, was cast with its cylinder—internal liner and casing—in one piece. The whole was coupled up and overhung the frame by means of a flange and bolts. The cylinder cover was formed by a double wall, forming a water chamber in communication with the cylinder jacket. Tightness was obtained, after a fashion, by means of asbestos joints. The inlet valve was placed on the cylinder end, whilst the exhaust valve was arranged in a separate box, supported from the side of the cylinder.

One of the forms that has been in use for several years by a number

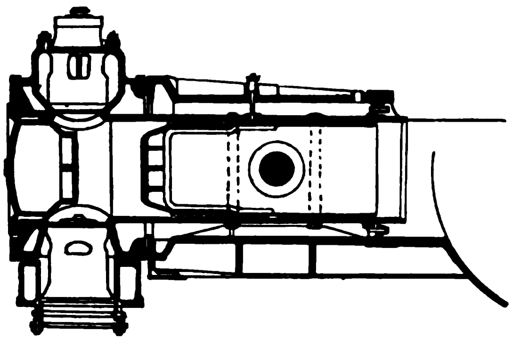


FIG. XII.—36. Reichenbach Engine Cylinder.

of English makers, and is still in use to-day, consists of a cylinder casing E (Fig. XII.—35) independent of the liner C, but of one piece with the combustion chamber A, in which the inlet and exhaust valves are placed, one above the other. This casing overhangs the frame, to which it is attached by bolts. In this arrangement, the joint J, between the liner and the separate water-jacket, is inaccessible and difficult to keep tight. Other makers employ the same arrangement,

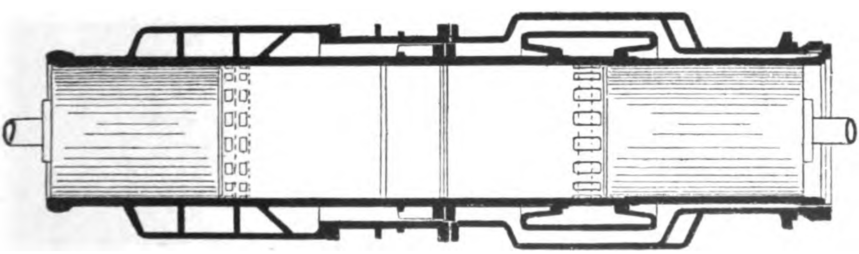


FIG. XII.—37. Section of Cylinder, Oechelhäuser Engine.

but with the admission valve placed on the horizontal axis, at the side of the cylinder. The frame casting forms the air chamber.

An old type of Tangye engine, which had the merit of a rationally formed combustion chamber, had both admission and exhaust valves vertical, the former in the back of the combustion chamber and the latter on one side.

The latest type of Hornsby-Stockport engines employ a cylinder liner and breech end independent of the frame, the latter being one with the water-jacket. The valves are vertical, one above the other.

As an interesting arrangement, reference may be directed to Fig. XII.—8, in which the casing is supported upon two longitudinal

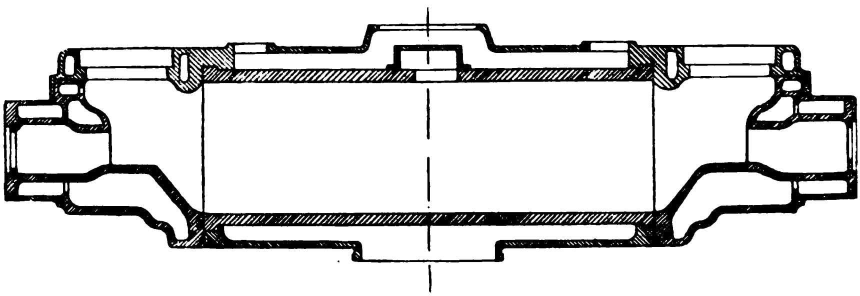


FIG. XII.—38. Section of Cylinder, Koerting Engine.

bearers from the frame, in order to avoid an overhanging cylinder, the whole being joined together by means of studs. Many American firms have adopted this type of engine, the liner and casing being one casting, with the valves at either side.

The special form of the Premier gas engine is alluded to in Chap. IV., p. 45. The single-acting Reichenbach engine cylinder

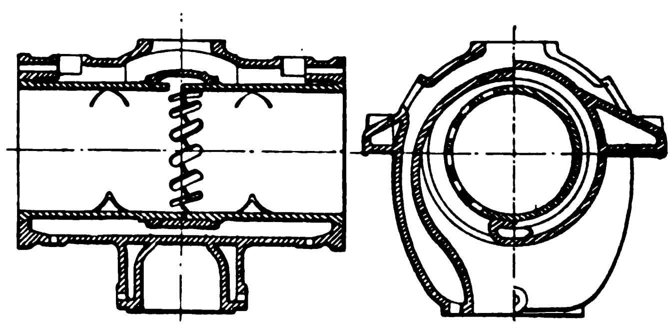


FIG. XII.—39. Section of Cylinder, Koerting type, Gutehoffnungshütte.

is illustrated in Fig. XII.—36, and is similar to the form of double-acting cylinder without a separate combustion chamber and with a water-jacketed cylinder cover.

Whatever the form adopted for the cylinders and water-jackets may be, a ½-inch hole for the reception of an indicator should be arranged in an accessible position, and tapped with a ¾-inch Whitworth thread.

This hole should be filled flush up to the interior wall of the combustion chamber by a plug or prolongation, so as to prevent

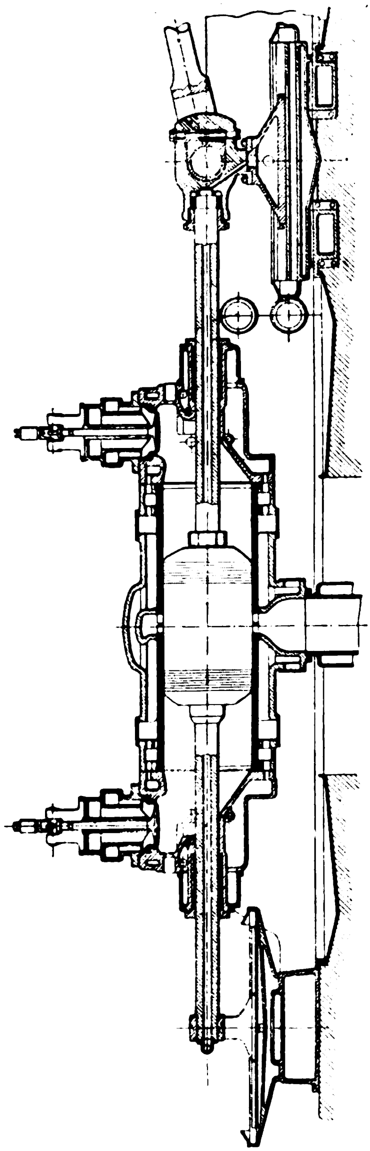


FIG. XII.—40. Section of Cylinder, Koerting type, Siegener Maschinenbau.

accumulation of oil or dirt in the gas passage when the aperture is not in use.

Cylinders—Double-acting Engines.—In double-acting engines, where the symmetry of arrangement of organs is of most importance, and in which the breech end is replaced by cylinder covers, the amount of expansion, due to the size of the pieces, should be particularly considered.

The effects of such expansion have been skilfully contended with in the Koerting and Oechelhäuser types of two-cycle engines.

In the modern Oechelhäuser engine (Fig. XII.—37), the internal cylinder liner is formed of two trunks tied at one of their extremities, whilst the other is free. These trunks are, moreover, independent of the casing, with which, as regards free expansion, they compare with the working conditions of single-acting engines.

In the Koerting, a similar arrangement is adopted (Fig. XII.—38). Fig. XII.—39 represents the transverse and longitudinal sections of a modern Koerting engine constructed by the Gutehoffnungshütte of Oberhausen. The cylinder liner is independent of the casing. It is

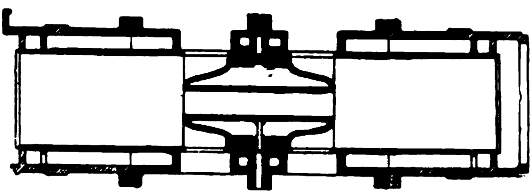


FIG. XII.—41. Section of Cylinders, Dingler.

formed of two parts joined in the middle of the exhaust ports in such a way that the expansion and contraction are entirely unfettered. In the lower part of the wall, upon which the piston bears, there are no exhaust ports, in order to ensure a large frictional area. The reactions due to the movement of the piston are normally absorbed by some intermediate cylindrical pieces and stays.

Fig. XII.—40 shows a horizontal section of a cylinder of a Koerting two-cycle engine made by Siegener Maschinenbau A. G.

The Dingler engine, four-cycle, with double open-ended cylinders, is of the same arrangement (Fig. XII.—41). The combustion chambers are placed back to back and bolted together, a long water-cooled box intervening within which the rod passes that couples the two pistons together.

In the other double-acting engines with closed cylinders, the free expansion of the internal liner and of the casing, the shape of which is made very complex by the valve orifices, is provided for in various ways. Certainly the most simple consists of casting the liner and

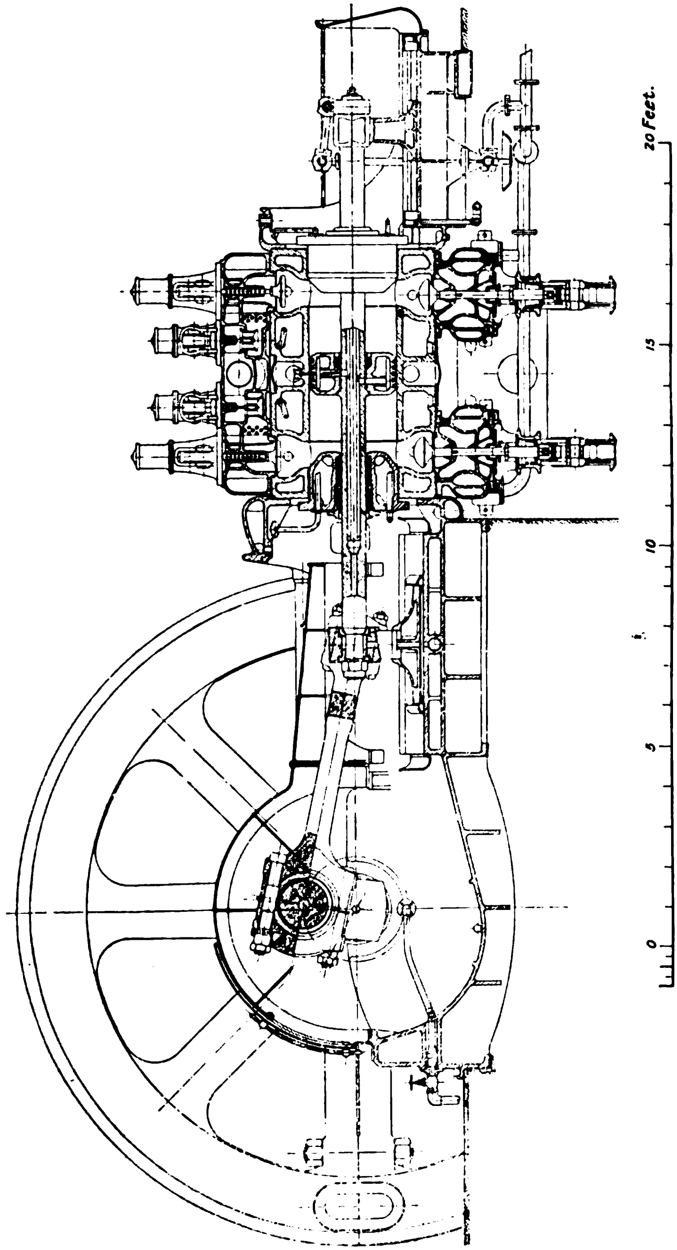


FIG. XII.—42. Section through Cylinder, Nürnberg double-acting Engine.

casing in one piece, spacing them sufficiently clear one from the other that the curves of the junction pieces at the extremities yield to slight bending by the elasticity of the material. This is done in the Maschinenfabrik Augsburg Nürnberg engines (Fig. XII.—42), who have experienced no trouble and have not been forced to run their cylinders colder than those of their competitors. The external portion of the

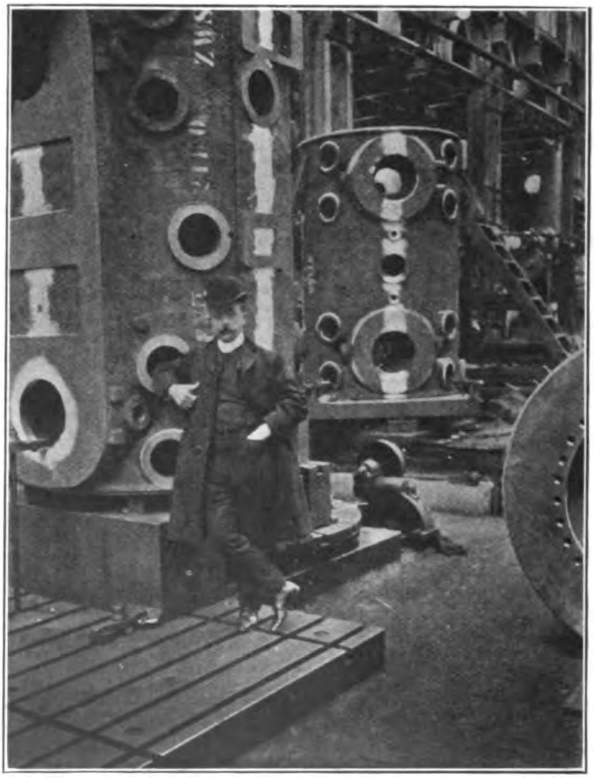


FIG. XII.—43. Cylinder casting, Nürnberg.

casing is stopped off, the gap being closed by a flexible circular band. Fig. XII.—43 represents one of the cylinders of the Nürnberg engines installed at the Rombach Steelworks.

The Ehrhardt & Sehmer cylinder is similarly cast solid with its casing (Fig. XII.—44), but this, owing to the great interval provided between the two parts, can expand freely, having regard to the elasticity of the cast iron at the points where the two parts are connected. These constructors have also striven to realise some symmetrical shapes to ensure equality of tension throughout. It will be seen that large cleaning holes are everywhere provided to give ready access to the casing.

In designing the form of the casting to accommodate the valve

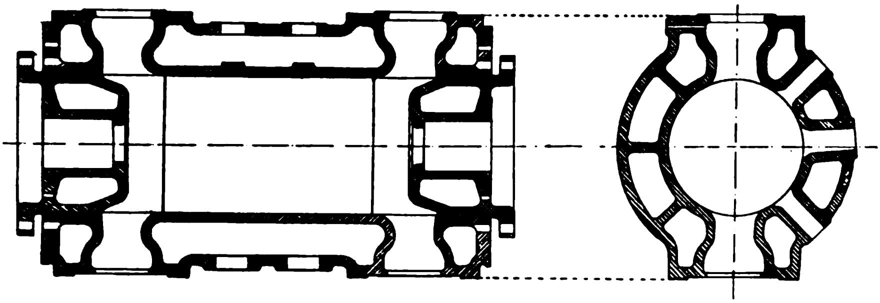


FIG. XII.—44. Section of Cylinder, Ehrhardt & Sehmer.

boxes, Ehrhardt & Sehmer have followed the arrangements adopted in high pressure pump construction which, but for temperature, work under similar conditions to those of gas engines. The cylinder is bolted in front to the base and bears freely at the rear upon a guided

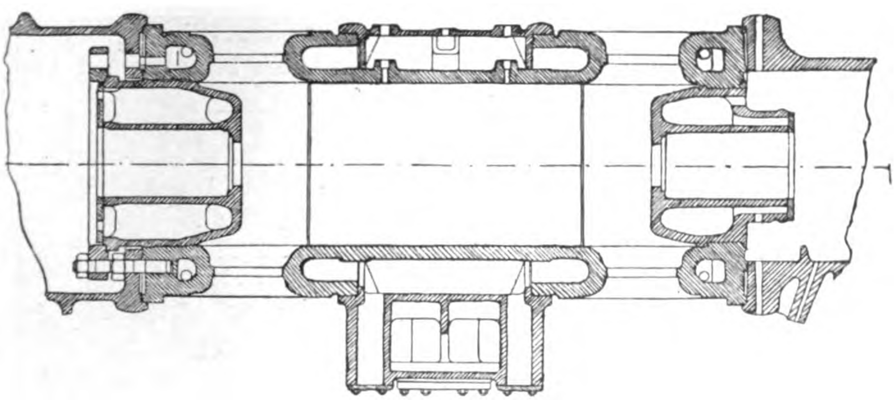


FIG. XII.—45. Section of Cylinder, Gasmotoren Fabrik Deutz.

slide which permits free expansion along the horizontal axis of the engine.

The Gasmotoren Fabrik Deutz, on the contrary, make some double-acting cylinders in two pieces (Fig. XII.—45). One is formed by the liner itself, in which the admission and exhaust apertures are cast, and the other by a sort of central sheath connected to the first by an expansion joint which permits the extremities to slide one in the other. The easy dismantling of the central portion enables the casing to be thoroughly cleaned.

The Gasmotoren Fabrik Deutz in their 2,000 H.P. engine have even

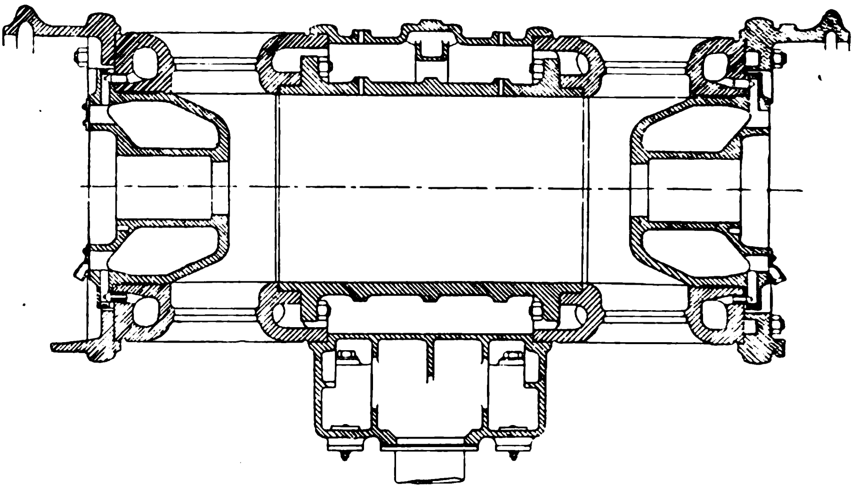


FIG. XII.—46. Section of Cylinder, Gasmotoren Fabrik Deutz.

deemed it advisable to make the cylinder in five pieces (Fig. XII.—46), the two extremities for receiving the valve boxes being formed by

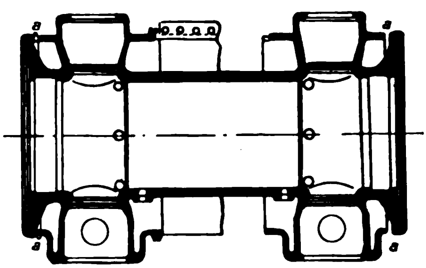


FIG. XII.—47. Section of Cylinder, Reichenbach.

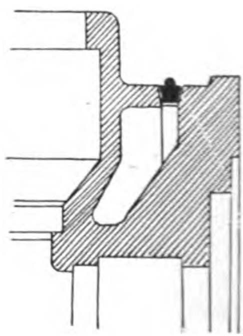


FIG. XII.—48. Detail of Cylinder joint, Reichenbach.

annular portions independent of the liner and of the external expansion sleeves. The valves are arranged in the same vertical axis, the air valve above and exhaust valve below, so that excess of oil or moisture may be expelled naturally.

The double-acting Reichenbach cylinder, represented in Fig. XII.—47, is entirely symmetrical to avoid contractions of the cast iron and to allow longitudinal expansion. The circular bearers are arranged close by the side of the valve chambers, and serve to support the wrought iron cylinder which forms the external wall of the water-jacket. The conical-shaped end flanges are separated from the adjacent cast-iron external casing by circular grooves which are filled by an india-rubber ring held by a cord (Fig. XII.—48).

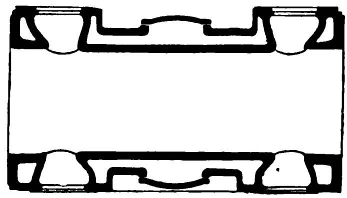


FIG. XII.—49. Section of Cylinder, Winterthur.

By the shape given to the cylinder covers and the piston itself, the compression chamber forms an annular space without recesses or projections, which is one of the characteristic features of the Reichenbach engine.

The Winterthur Co. similarly adopt a symmetrical arrangement (Fig. XII.—49). The whole is cast in one piece, but the external casing is stopped off in the centre to allow for expansion. The opening is closed by two cast-iron semi-circular pieces to form the water chamber.

As shown in Fig. XII.—50, the cylinder of the Elsassische

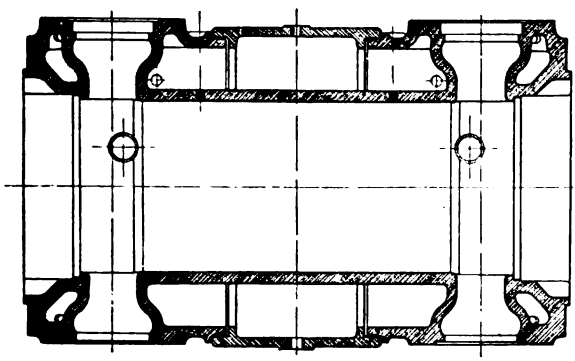


FIG. XII.—50. Section of Cylinder, Elsassische.

Maschinenbau engines is made on the same principle. The internal liner and the valve box seats are cast in one piece, and the central water chamber is formed by two semicircular pieces carried and bolted as in the Nürnberg cylinder, near the valve box seats, by solid stays passing through the water-jacket to strengthen the liner.

The cylinder of the Markische Maschinenbau Anstalt engine is in three pieces (Fig. XII.—51). One of these forms the central part of the cylinder, liner and jacket, and at the same time is part of the supporting beam which runs the length of the engine and serves to connect the tandem cylinders as in the arrangement of the Cockerill engine (Fig. XII.—52). The covers are tied together by long bolts passing through the water chamber.

In the cylinder (Fig. XII.—51) this central portion is free to expand. The two end parts which receive the valves are shorter, expansion

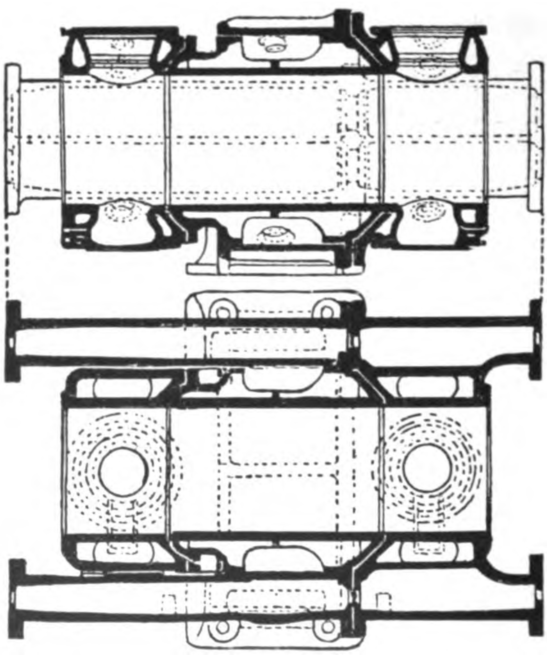


FIG. XII.—51. Section of Cylinder, Markische.

has less effect upon them, and they form annular pieces with a closed casing. One of these end pieces is independent of the frame, whilst the other is part of the central portion.

From the point of view of stability as much as of extension by expansion, multiple-cylinder, double-acting engines should be specially designed. The Cockerill cylinders, by the method of connection with screw and keys, are independent of the frame on which they rest. They are cast with their outer casings and their tubular valve box seats, inlet above and exhaust below. Their construction is absolutely symmetrical, to encourage the resistance of the metal to the tensile stresses produced.

Dingler employs cylinders in the form of simple tubes, interchangeable

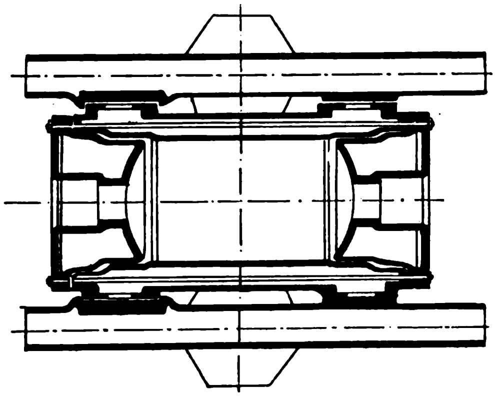


Fig. XII.—52. Section of Cylinder, Cockerill.

and without projections, entirely free to expand at one end. The connection is made by flanges arranged outside the water chamber (Fig. XII.—53).

In the Snow Steam Pump Co.'s 5,400 H.P. engines the cylinder is in

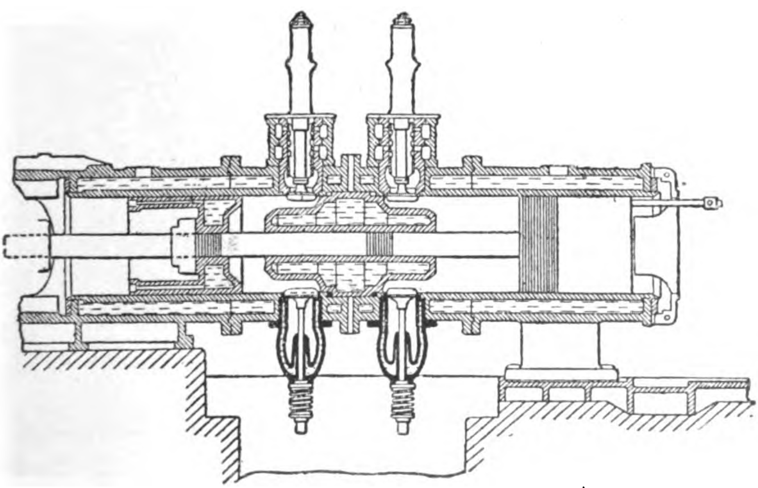


Fig. XII.—53. Section of Cylinder, Dingler.

two pieces with a joint in the middle (Fig. XII.—54). Each of the two pieces is cast vertically, the flange upwards, with a big waste end, so as to obtain soft and sound iron in view of the turning of this flange, and hard iron at the outer end which receives the cylinder cover. The flanges are let in within the water chamber, and the casing is

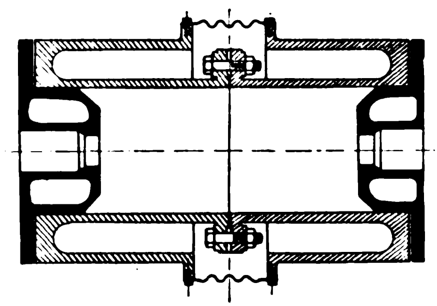


FIG. XII.—54. Section of Cylinder, Snow Steam Pump Works.

closed by an annular joint. The cylinders are supported on the frame with slides intervening to allow expansion.

Fig. XII.—55 represents the cylinder of a double acting tandem engine made by the Allis Chalmers Co., having a diameter of 42 inches for a piston stroke of 54 inches.

The Duisburg M. A. G. (formerly Bechem & Keatman) in their

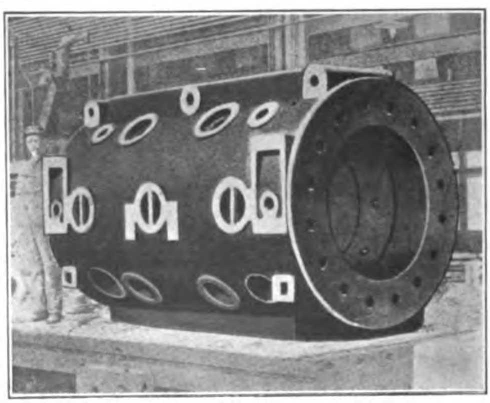


FIG. XII.—55. Cylinder Casting, Allis Chalmers Co.

double-acting engines, place the inlet and exhaust valves one above the other, but on a vertical axis out of centre with the cylinder as shown in Figs. XII.—56 and 57. This arrangement constitutes a combustion chamber having an annular form which, upon ignition of the mixture, tends to set up a rotatory movement of the gas favourable to the rapid propagation of flame.

The internal liner of the cylinder is made in two parts joined in the

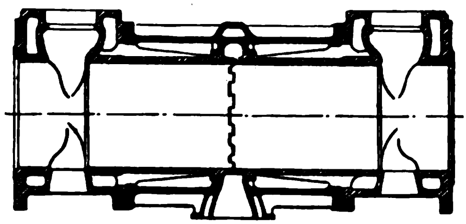


FIG. XII.—56. Section of Cylinder, Duisburg.

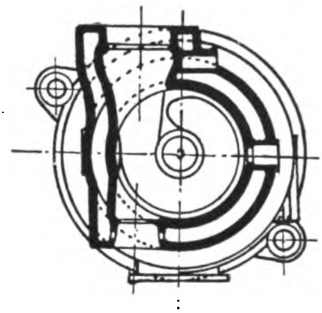


FIG. XII.—57. Section of Cylinder Duisburg.

centre at the circular exhaust ports, the latter being arranged along the line of juncture (Fig. XII.—56). At the end of the piston stroke these ports are uncovered and the hot gases, still at a pressure of 20 to 30 lbs. per square inch, escape freely. The exhaust valves proper, being called upon merely to pass the gas completely expanded, may have a much

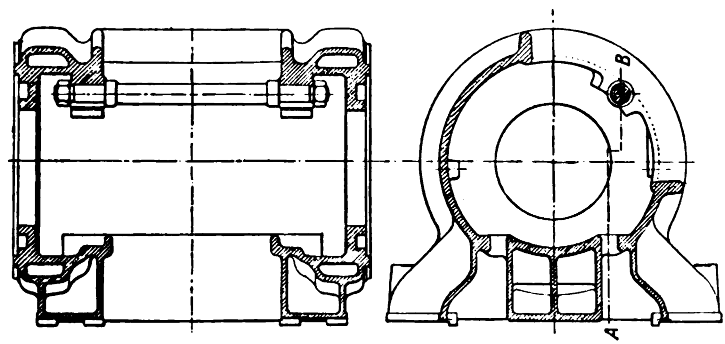


FIG. XII.—58. Intermediate casting or distance piece, Koerting.

less diameter than in other four-cycle engines. Moreover, water cooling is unnecessary.

Fig. XII.—58 shows an arrangement adapted by Koerting Brothers, to tie together two cylinders in tandem. The connecting piece is a hollow casting giving access to the crosshead, piston rod, cylinder covers, and stuffing-boxes. To avoid all risk of fracture a steel tie rod is provided towards the middle of the opening.

The great diversity of the forms of cylinders successively employed by the constructors gives an idea of the difficulties which have been encountered in making the pieces in such a manner that no cracks develop sooner or later during work. Nearly all the makers of engines of considerable size have been troubled with fractures of this kind, at least in the first machines they constructed. Generally the external walls are the first to crack and then fissures appear either in the connecting flanges or in one or other of the corners of one of the openings giving access to the water chamber. These fissures gradually extend without interrupting the working of the engine whilst they do not affect the inner walls of the cylinder. Fissures in the inner walls are manifested by the irregular and jerky flow of cooling water from the jacket outlet.

With regard to the wear of cylinders, this occurs to a greater extent at the point at which the piston bears at the moment of explosion, and it is not the same in the horizontal as in the vertical direction. In a paper presented by Mr. C. St. George Moore to the Society of Engineers (London) in July, 1907, the following figures were given relative to the different cylinder diameters of a two-cycle Koerting engine made in England, after erection and also after eight months' wear:—

	5th July, 1905.	16th March, 1906.	Difference.
	Inches.	Inches.	Inches.
Front end, vertical . . .	29·925	29·990	0·065
,, ,, horizontal . .	29·911	29·956	0·045
Centre of ports, vertical . .	29·995	30·025	0·030
,, ,, horizontal .	29·975	30·000	0·025
Back end, vertical . . .	29·911	29·975	0·064
,, ,, horizontal . .	29·890	29·963	0·073

This amount of wear appears excessive, and in the author's opinion is only due to abnormal working conditions.

Dimensions.

Cylinder Cover Bolts.—For large engines made after the style of the Nürnberg (Fig. XII.—42) and Ehrhardt (Fig. XII.—44), the connecting bolts are arranged in two concentric rings, each being placed in the extension of the inner and outer cylindrical portions. In this way the tension is more uniformly distributed, and a better joint is obtained.

Thickness of the Liner.—For small engines, as mentioned on p. 300, the equation used is

$$e = \frac{D}{20}$$

For double-acting engines in which the liner supports considerable strains, particularly owing to the cleavage or separation of the water-jacket, the following equation serves :—

$$e = \frac{D}{12 \text{ to } 14}$$

Cylinder Covers.

The cylinder covers are the simple cover-plates, generally water-cooled, which close the cylinders either at front or back, similar to those used in steam engine practice.

If the cylinder covers must encircle a piston rod, they are provided with a central stuffing-box. When these are used the valves are fitted to the cylinder itself.

Some makers, such as Charon, Foos, etc., have adopted cylinder covers for small single-acting engines. They are usually reserved for double-acting types, or for tandem engines for which an ordinary breech end is not suitable.

Dingler uses cylinder covers instead of breech ends in a similar way.

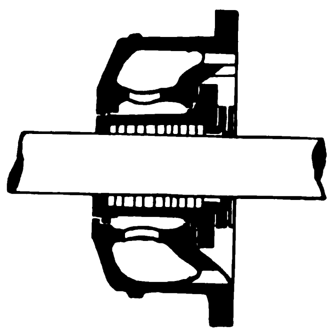


Fig. XII.—59. Cylinder cover, Winterthur.

With the type of engines without breech ends it is possible, by adding a tandem cylinder, to economically double the power. The progress in double-acting engine construction, moreover, at the present time permits the cylinder cover arrangement to be considered equally as practical and dependable for working as that of the standard breech end.

Water circulation is provided, apart from the cylinder jacket, for the cylinder covers, stuffing-boxes, exhaust valve boxes, pistons and rods, and sometimes even the crank shaft bearings and crosshead slippers and guides.

The cylinder covers of the Gasmotoren Fabrik Deutz are held by bolts with springs, permitting slight movement set up by expansion.

Fig. XII.—59 represents the cylinder cover of a double-acting

Winterthur engine. It will be seen that the stuffing-box is entirely surrounded by water, whilst in the majority of other designs it is

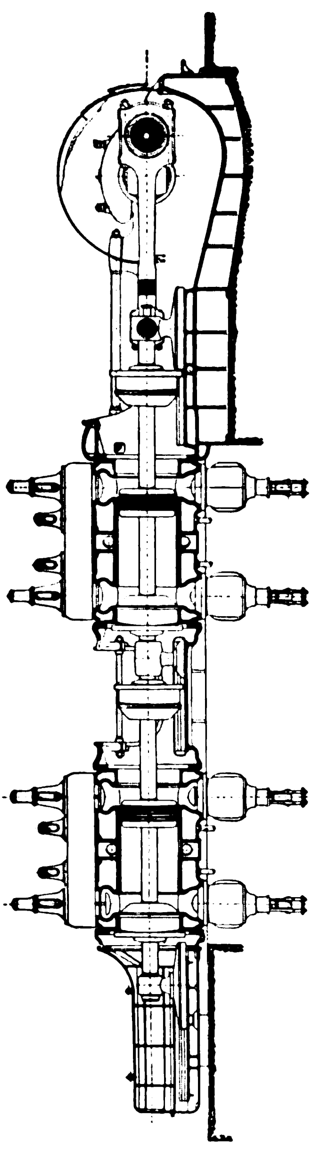


Fig. XII.—60. Nürnberg Engine with front Cylinder covers removed.

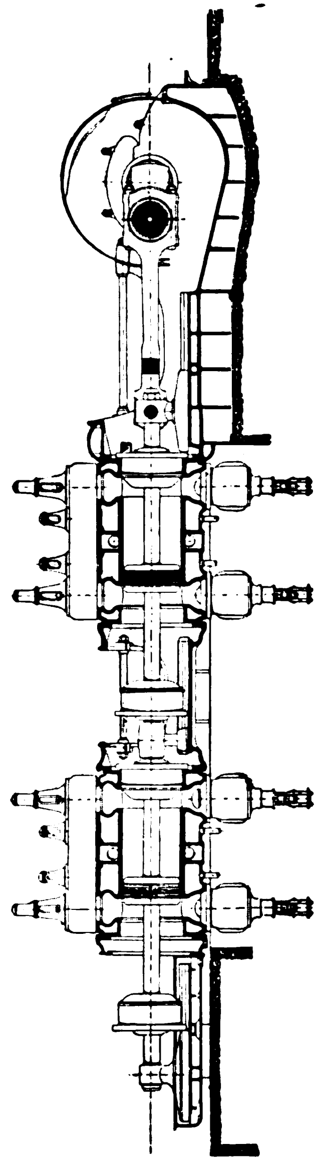


Fig. XII.—61. Nürnberg Engine with back Cylinder covers removed.

placed wholly within the cast-iron walls and is cooled in consequence, only indirectly.

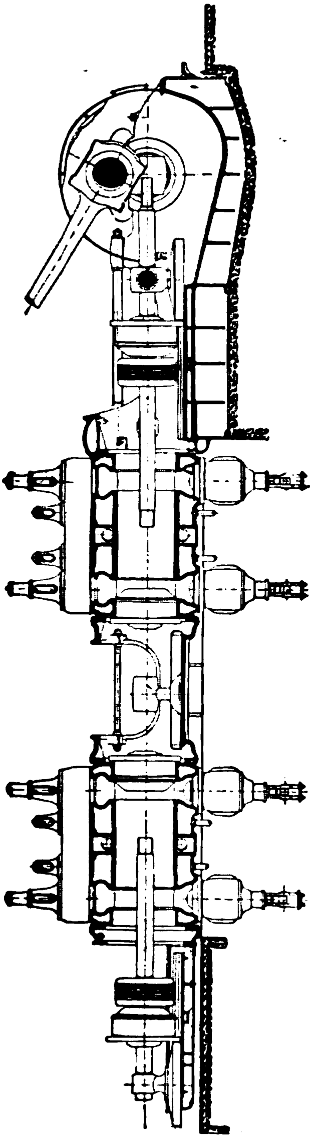


FIG. XII.—62. Nürnberg Engine with Pistons withdrawn.

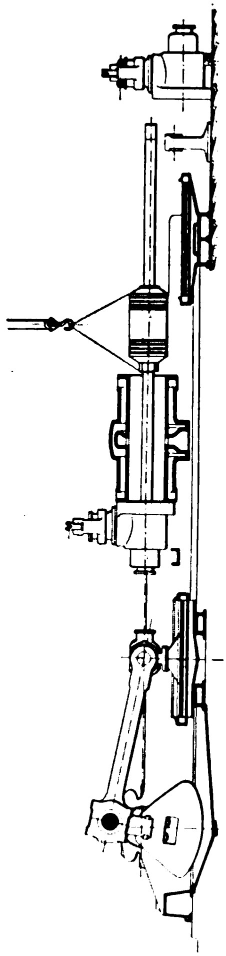


FIG. XII.—63. Removal of Piston from back end of ordinary Engine, Siegener.

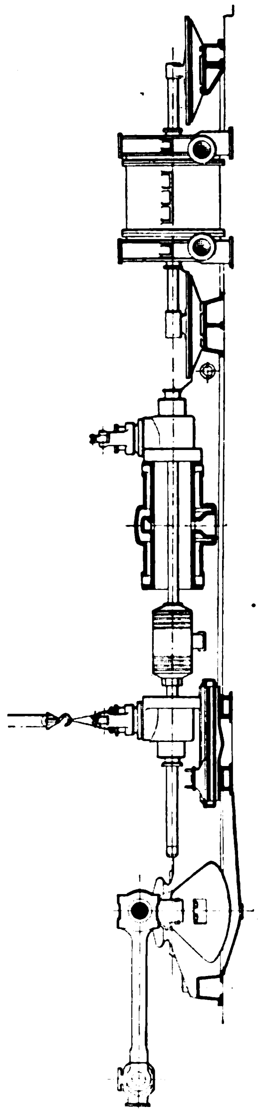


FIG. XII.—64. Removal of Piston from front end of Blowing Engine, Siegener.

Figs. XII.—60 and 61 show how, in a Nürnberg tandem engine, the front and back covers respectively are displaced to give access to the corresponding valves.

The pistons are removed by disconnecting the connecting rod, as in

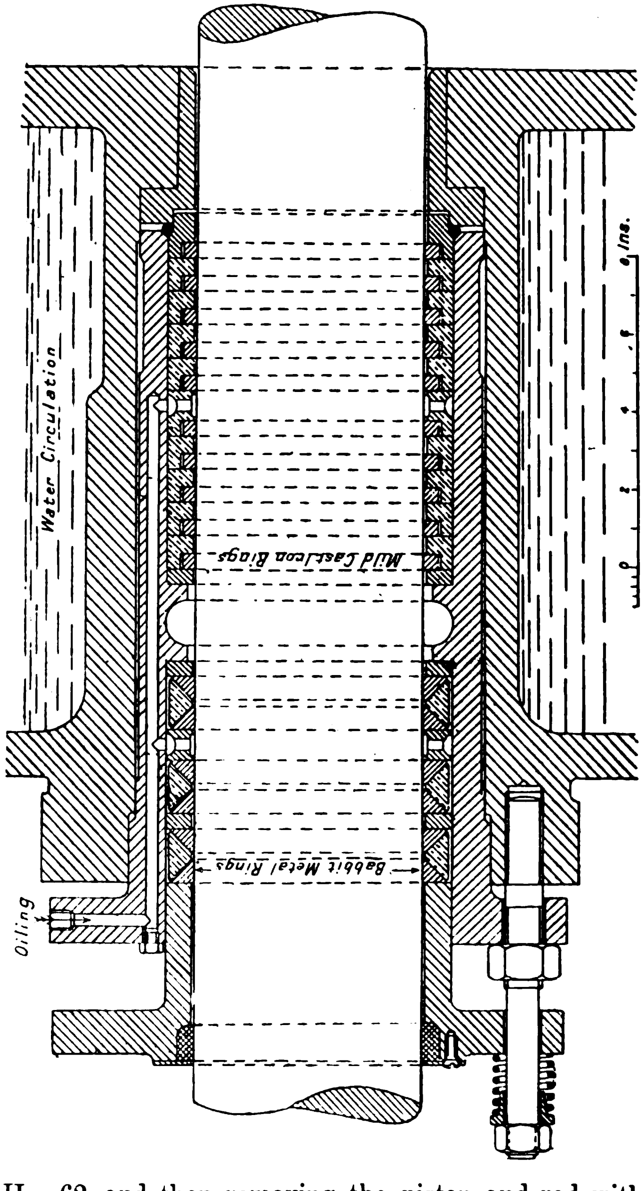


FIG. XII.—63. Stuffing-box, Gasmotoren Fabrik Deutz.

Fig. XII.—62, and then removing the piston and rod with the front cover, the piston rods being disconnected at the middle to free the back piston.

The Ehrhardt & Sehmer engine is similarly constructed with easily removable cylinder ends. In Fig. V.—10, p. 77, particulars have been given of the method adopted when the pistons are dismantled in a Winterthur tandem engine. Figs. XII.—63 and 64 illustrate the same operation in connection with the two-cycle engine of the Siegener M. A. G.

Stuffing-Boxes.

The problems connected with the construction of piston rod stuffing-boxes, so intimately bound up with the good working of double-acting engines, appear now to have been solved, but they were frequently a check upon the makers' progress in the early days.

Metallic packing rings are usually adopted in elastic segments

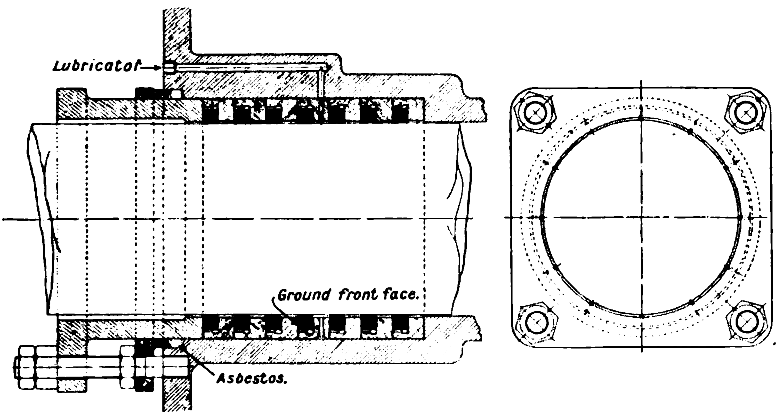


Fig. XII.—66. Stuffing-box, Cockerill.

capable of a certain amount of movement at right angles to the axis, with chambers permitting a free circulation of oil and giving protection from becoming overheated.

The oil tube should be fitted with a retaining valve, and it is important that the weight of the rod and piston be not carried by the stuffing-boxes, but by the crossheads.

The packing rings are frequently placed in two compartments in the same box, being separated by a chamber in connection with the exhaust pipe. The boxes are cooled either directly—as shown in Fig. XII.—59—or indirectly, by water circulation.

Koerting Brothers were the first to solve the problems involved by employing spring rings round the piston rod and friction segments in white metal cooled by water circulation.

The Otto-Deutz stuffing-box (Fig. XII.—65) consists of a movable sleeve fixed to the cylinder by an external flange. The sleeve is provided with a longitudinal lubricating passage. At the end towards the cylinder, a joint ring is placed. In the first compartment is a series of carefully fitted rings, and in the grooves between them the elastic segments are placed to bear upon the rod, the segments being made in one piece of special cast iron.

In the second compartment, towards the outer end, white anti-friction metal rings of triangular section are placed alternately with

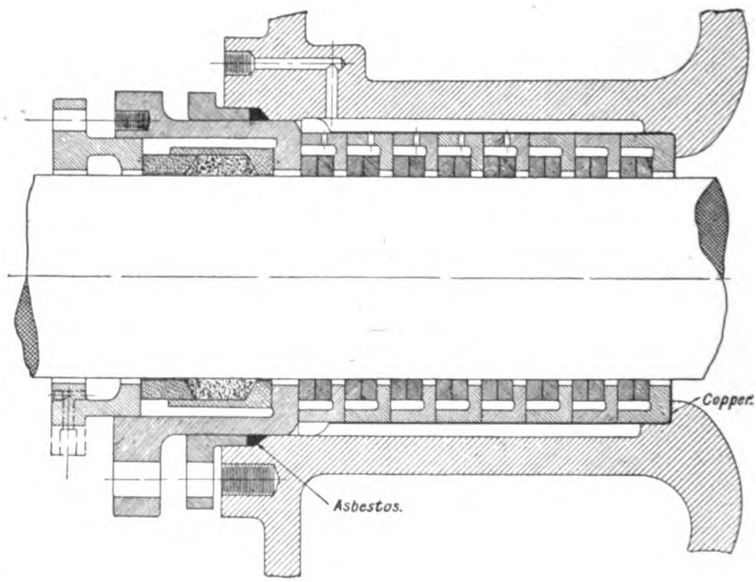


FIG. XII.—67. Stuffing-box, Elsassische.

bevel shaped steel rings to effect the necessary tightening by means of an outer flanged gland forming the stuffing-box.

In the arrangement adopted by the Cockerill Co. (Fig. XII.—66), the packing rings have their inner faces ground and are placed in the grooves at the enclosing rings that form the box with a play of about 4 mils (0·004 inch). An asbestos joint is fitted to the front portion of the stuffing-box. Lubrication is effected under pressure from the third packing ring, as shown in the figure.

Fig. XII.—67 represents the Elsassische arrangement. Six solid cast-iron segments are placed in pairs in the rings which form the rear portion of the box. These segments are made similar to piston rings. The grooves for the first five pairs of segments are placed in communication with an oil chamber. In the front a second stuffing-box for elastic material, asbestos for instance, is arranged to permit tightening during work if necessary.

Fig. XII.—68 shows the Schwabe water-cooled stuffing-box. At the

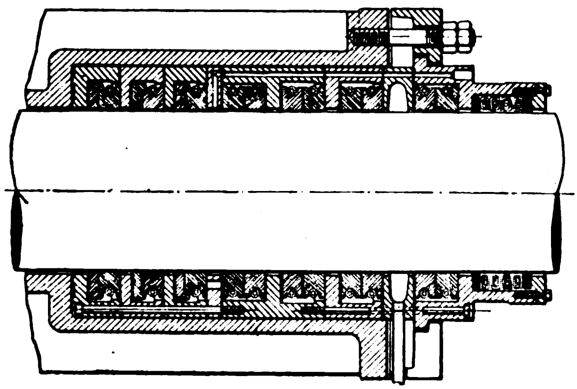


FIG. XII.—68. Stuffing-box, Schwabe.

back the segments forming packing rings are of cast iron, in three pieces, pressed inwards by springs. The front box is similar to that of steam engines.

Fig. XII.—69 represents one of the Nürnberg stuffing-boxes made

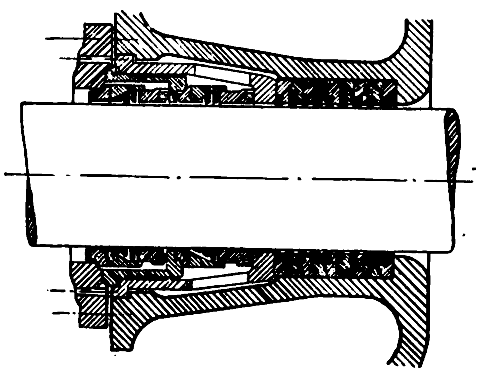


FIG. XII.—69. Stuffing-box, Nürnberg.

in three parts. At the cylinder end solid soft cast-iron segments are placed, held by a series of outer rings. At the front two series of triangular antifriction packing rings are fitted. The chamber separating the two compartments is in communication with the exhaust. Lubrication is effected under pressure.

Fig. XII.—70 shows the Haniel & Lueg stuffing-box. The tightening rings are bevelled and the wedge-shaped packing rings are arranged in pairs.

All types of stuffing-boxes should be cleaned out from time to time to maintain free movement of the segments which little by little become coated with burnt oil. To facilitate this cleaning it is necessary to ensure that they can be readily taken to pieces and that all parts are quite accessible.

Fixing of Cylinders.—The method of fixing the cylinders of large gas engines to their foundations is an important matter, which has necessitated much attention on the part of the makers. Those

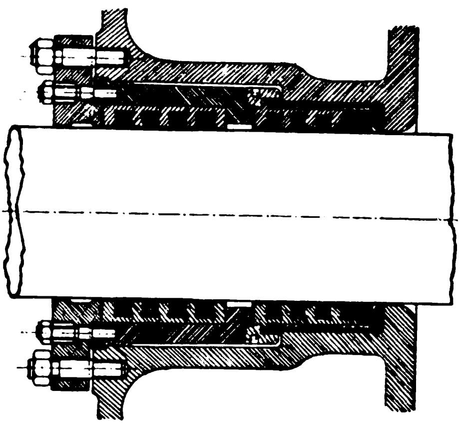


FIG. XII.—70. Stuffing-box, Haniel & Lueg.

who have had practical experience with large steam engines, and were thus familiar with the details, have successfully solved the problem.

It has already been explained how, in the latest Cockerill engine (Fig. XII.—52), the cylinders are fixed laterally to a system of cast-iron beams running the entire length of the engine. The Markische engine is similarly fixed (Fig. XII.—51), the cylinders being bolted or keyed to the beams by means of large bearers.

In the Nürnberg, Otto-Deutz, Krupp, and other engines the casing is cast with lateral feet at the middle of the cylinder. In this way the ends are left clear and the exhaust valves are easy of access.

During erection it is indispensable to equally, systematically, and finally tighten the holding-down bolts before commencing to put the working parts in position.

Breech Ends.—The influence of the shape of explosion chamber upon the efficiency of an engine is manifest. The enclosure within which combustion at high temperatures occurs, from a thermal standpoint, should present a minimum of cooling surface and be free from projections or recesses. Projections constitute obstacles to the movements of the exploding gas, while recesses are likely to retain a portion of the fresh mixture and to permit it to escape finally unburnt or to retain the burnt gases which pollute the following charges.

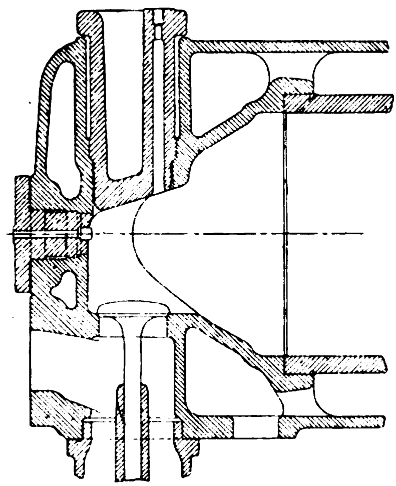


FIG. XII.—71. Form of Combustion Chamber, Tangye (old type).

Explosion chambers short and regularly shaped, arranged as an extension of the cylinder, are to be preferred. An example is given (Fig. XII.—71) of the old type of Tangye single-acting engine.

The ideal form would be either a spherical or a parabolic cylinder head, with the ignition device at the back or lowest part, the valves

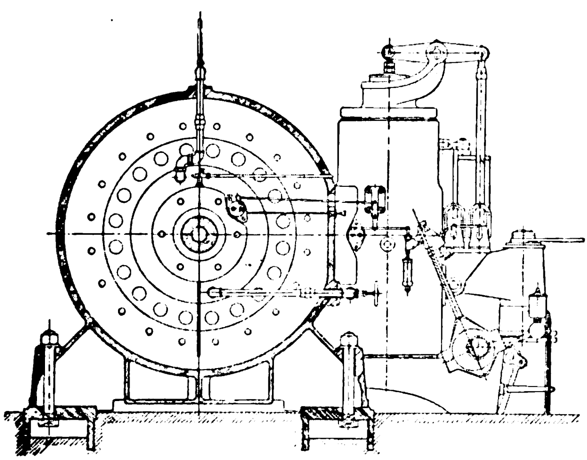


FIG. XII.—72. Koerting four-cycle Engine with side Valve Boxes.

being placed to open into the cylinder axis without forming projections, hollows, or recesses. Unfortunately the ideal cannot be obtained in practice as much from constructional as from functional points of view.

Compression pressures of from 140 to 170 lbs. per square inch can

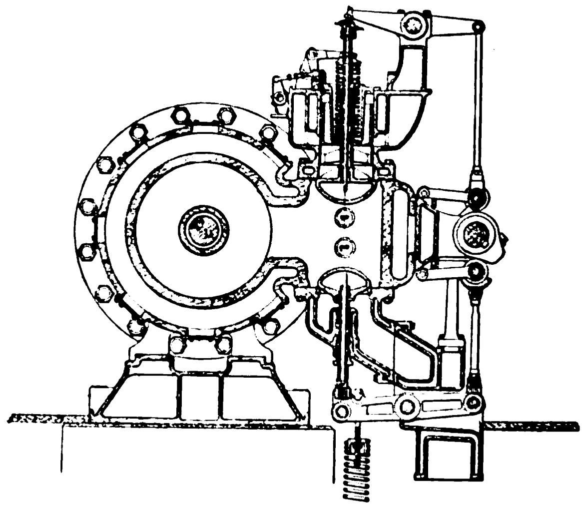


FIG. XII.—73. Cylinder with side Valve Boxes. Snow Steam Pump Works.

only be obtained when the volume of the compression space is from 18 to 15 per cent. of the total cylinder volume, and this involves either long piston travel or deeply domed pistons. Moreover, for ease of construction and rational disposition of valve mechanism,

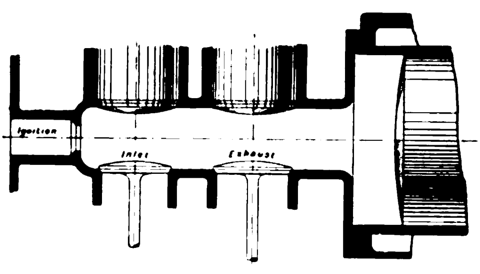


FIG. XII.—74. Faultily designed Valve Chamber.

the valves, as far as possible, should be arranged vertically and in the same axis.

The excess of lubricating oil plays an important part in the working of gas engines, and should be blown out naturally, as formed, at each exhaust stroke as in the engines which have the discharge valves situated in the lowest part of the cylinders. These valves, however, demand the most care and attention as far as inspection and maintenance are concerned, and, therefore, it should be recognised that, thus placed, they are very badly arranged, because under the cylinder they are very inaccessible.

Moreover, in powerful engines and tandem types the exhaust valves below the cylinders and below ground level involve separation of the foundations affecting their strength, unless they are excessively massive.

All these practical considerations, the most important of which are simplicity, ease of construction, and access, have led Koerting Brothers to prefer separate explosion chambers for their large four-cycle engines,

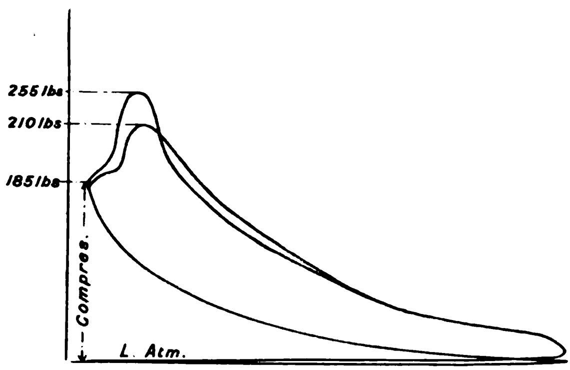


FIG. XII.—75. Diagram from Engine with Valve Chamber as Fig. XII.—74.

supported laterally by the cylinders, notwithstanding the theoretical considerations. In these engines (Figs. XII.—72 and XI.—45) the evacuation of oil is effected by a blow-off valve specially fitted at the point at which, in other engines, the exhaust valve is placed.

The lateral chamber was also adopted in the double-acting engine made by the late Blaisdell Machinery Co., at Bradford, Pa., U.S.A., as well as in the old type horizontal Westinghouse engine. The Snow Steam Pump Works use the same arrangement, as seen in Fig. XII.—73, for their 5,400 H.P. engine, to which reference has been made previously.

As shown in Fig. XI.—60, Schuchtermann & Kremer, of Dortmund, arrange the exhaust valve in a side chamber, whilst the inlet valve is placed above the cylinder.

When the side chamber contains the two valves, a long passage is formed which may prejudice the rapid propagation of flame. It is then necessary to fit two ignition devices within the ignition chamber to avoid working with a considerable amount of lead or advanced ignition. It is very difficult to design a very short explosion chamber with lateral valve gear, and therefore the bad points must be taken with the good.

In some breech ends which are a prolongation of the cylinder, and in which the sparking points of the magneto ignition plug are placed at the side of the explosion chamber and near the cylinder liner, the author has frequently obtained the ignition at the exact dead centre, even when the trip gear had operated when the crank shaft had from

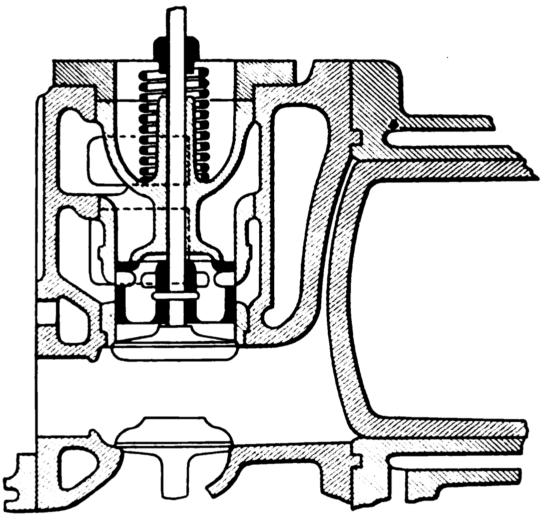


FIG. XII.—76. Combustion Chamber of Nürnberg Engine.

3° to 5° of inclination before the dead centre, whilst with a certain old type of German engine it necessitated a lead of 50° to 60°.

In one similar engine of 75 H.P. tested by the author, the breech end had a very elongated form, as shown diagrammatically in Fig. XII.—74, and contained both air and exhaust valves on the same level. The ignition device was at the back.

The characteristic diagram taken from this engine (Fig. XII.—75) proved that the explosion had two distinct phases. The first, from the mixture within the combustion chamber, reached 185 lbs. per square inch, while the second, resulting from the ignition of the mixture in the immediate vicinity of the piston, was from 210 to 255 lbs. per square inch. The breech end, abundantly water-cooled, must undoubtedly act prejudicially to the rapid propagation of flame, but, as soon as the latter reaches the gas close behind the piston within the very hot cylinder, a more powerful burning increases the explosion pressure. The mean linear piston speed of this engine was 785 feet per minute.

As for the evacuation of excess oil by the exhaust, the Maschinen-

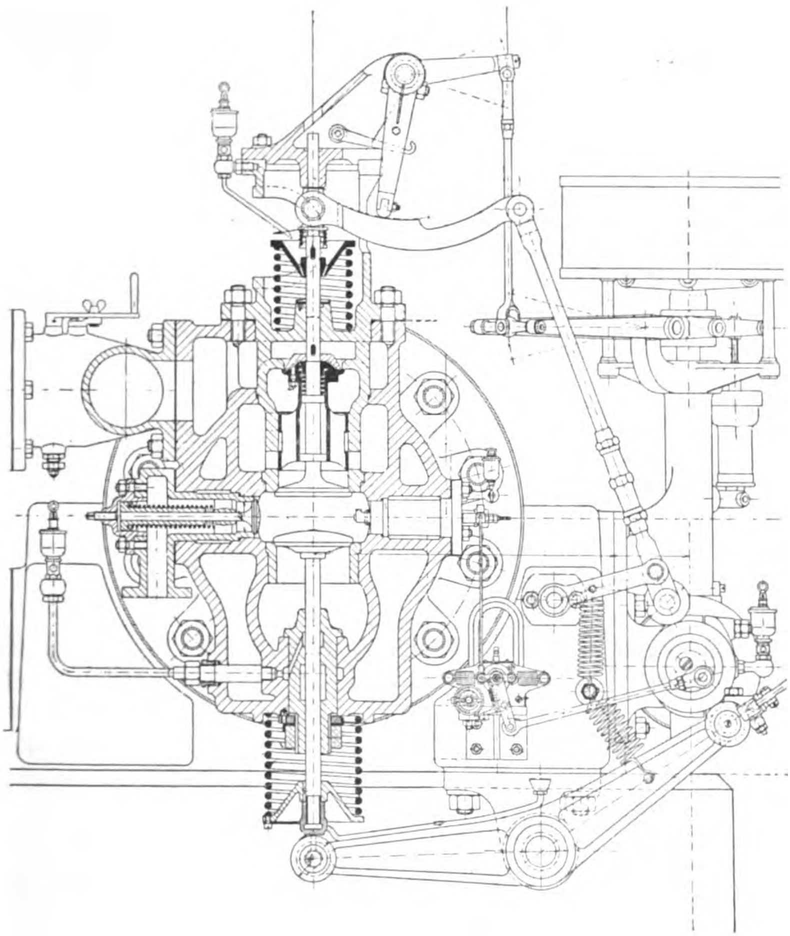


FIG. XII.—77. Section of Combustion Chamber, Gasmotoren Fabrik Deutz.

fabrik Augsburg Nürnberg attach so much importance to it, that in their single-acting engines they have lowered the explosion chamber so that the exhaust valve seat is sufficiently low to permit the oil driven back by the piston to be got rid of automatically (Fig. XII.—76). From this point of view it is an interesting arrangement, in spite of the peculiar shape it gives to the combustion chamber; but it is to be feared that during the explosion the thin film of gas between the back of the piston and that of the cylinder in the upper part of the clearance space will be retarded in its combustion. Diagrams, taken simultaneously at this point and in the back of the breech end itself, confirm this opinion. Soest used the same arrangement in his 350 H.P. engine exhibited at Düsseldorf in 1902.

The author does not recommend the automatic discharge of oil by the exhaust valve when the latter is not water-cooled, because the high temperature of the valve may give rise to the deposit of hard carbon and dirt, which would tend to cause pre-ignition.

Symmetrical explosion chambers, forming a vertically narrow exten-

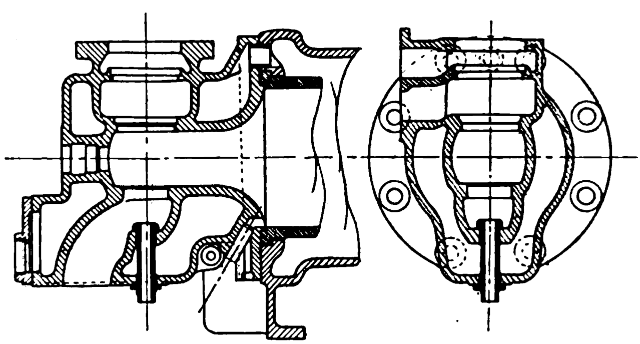


FIG. XII.—78. Longitudinal section. FIG. XII.—79. Transverse section.
Form of Combustion Chamber.

sion of the cylinder with the exhaust passage in the longitudinal centre line of the engine, are now adopted by a large number of makers.

The expansion of metal exposed to high temperatures produces unequally distributed strains. Fractures due to unequal expansion have therefore become very infrequent, and are more often caused by stresses in the cast iron, originally due to unequal thicknesses of the walls, from displacement of cores during casting, or to defective connections between the explosion chamber and the walls of the water-jacket. These connections consist of the inlet, exhaust, ignition, and starter valve casings, which present difficulties in moulding that only long experience can successfully overcome.

Fig. XII.—77 is taken from the *Gas and Oil Engine Record*, and is a transverse section through the centres of the valves, breech end, and valve gear of a 110 H.P. Otto-Deutz engine of modern type, with 520 mm. cylinder diameter and 620 mm. piston stroke (20·5 × 24·4 inches).

The two sections in Fig. XII.—78 and 79 show the general shape and the arrangement advocated by the author, and several large

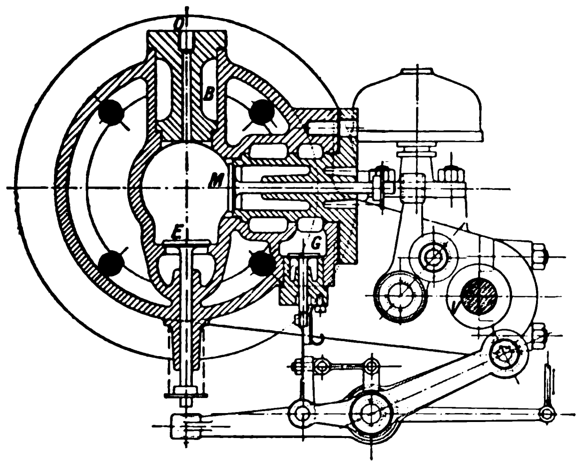


FIG. XII.—80. Breech end for small Engines.

English and American firms have adopted the principle. As shown in the drawing, the water space is very spacious and uniformly

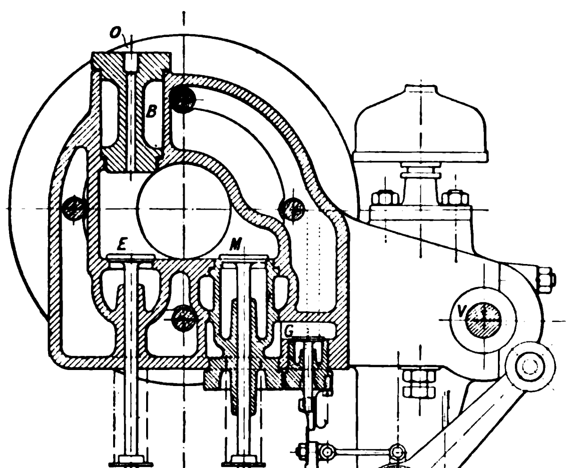


FIG. XII.—81. Breech end for small Engines.

surrounds all parts of the exhaust valve seat. The exhaust valve spindle is guided in a well cooled sleeve. Large openings give access to the interior of the water chamber. The oil scavenging valve and passage is arranged at the lowest part.

Figs. XII.—80 and 81 illustrate two breech ends suitable for small engines. In Fig. XII—80, the inlet valve *M* is horizontal, and carried in a removable box. The exhaust valve *E* is vertical and placed on the centre line, its seat being part of the main casting. It is put into place through an upper opening closed by a plug, the latter being cast with an aperture to serve as an indicator connection. The gas valve *G* is similarly vertical. Each valve is operated by a special lever.

In Fig. XII.—81 the three valves are vertical. The exhaust valve *E*

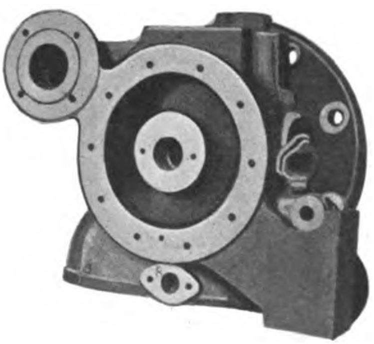


FIG. XII.—82. Breech end Casting of Schmitz Engine.

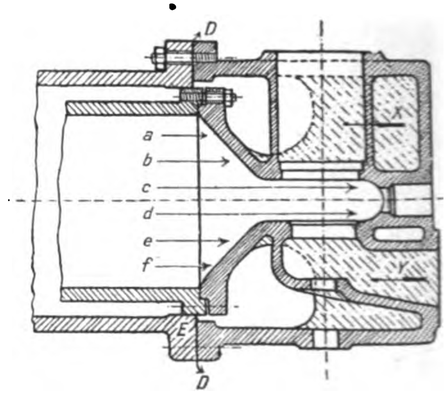


FIG. XII.—83. Bollinskx Combustion Chamber.

is placed at the side. The two operating levers are fitted upon the same fulcrum pin *A*.

Fig. XII.—82 is reproduced from a photograph of the breech end of a Schmitz engine. For their new type small engines, this firm have a special method of ensuring free expansion of the internal walls of the combustion chamber and the outer walls. As seen from Fig. XII.—34, all direct connection between the inner and outer walls of the combustion chamber is suppressed near the portion which is attached to the frame. The external wall is bolted to the cylinder water-jacket, and the internal wall into that in which the piston moves. The latter bolts are lengthened to reach the rear end of the breech casting.

Fig. XII.—83 shows a similar arrangement made by Bollinckx with the bolts let in the water chamber. In the Foos engine (Fig. XII.—84), the breech end is replaced by a simple water-cooled back cover; the valves are placed on either side of the cylinder, the liner and jacket are cast together.

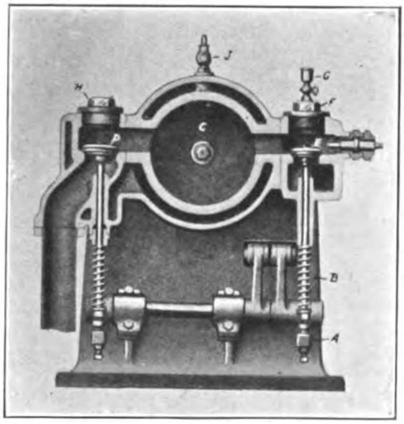


FIG. XII.—84. Foos Engine.

In the old Otto type both the valves were below the cylinder, the inlet valve being on the centre line and the exhaust valve on one side. The 600 H.P. Simplex engine breech end made by Cockerill also had the two valves underneath.

The Crossley, Dudbridge, &c., combustion chambers, which are either separate castings or else cast in one with the cylinder water-jacket, have the exhaust valve vertical and the inlet valve at the side, the exhaust valve being removed through an upper aperture closed

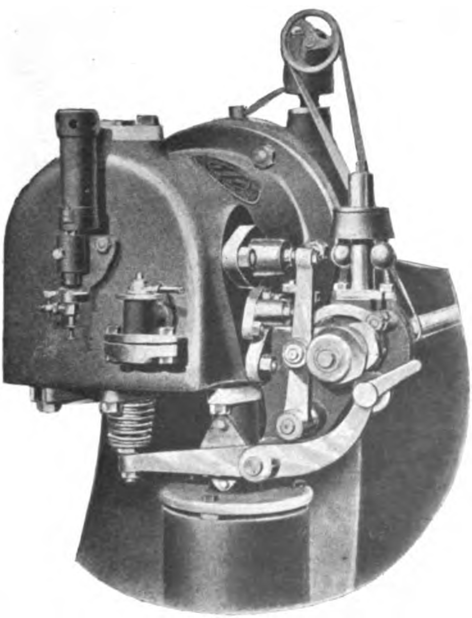


FIG. XII.—85. Breech end of Dudbridge Engine.

by a hollow plug. The gas and mixture inlet valves are in removable boxes which form their seats.

Fig. XII.—85 shows a back view of the Dudbridge combustion chamber end. This arrangement has been fitted to town gas engines and oil engines. For producer gas engines new types have recently been built as described elsewhere.

Fig. XII.—86 shows, in longitudinal and transverse section, a breech end designed by Mr. R. Bellamy, who for many years has skilfully directed the construction of the Hornsby-Stockport gas engines.

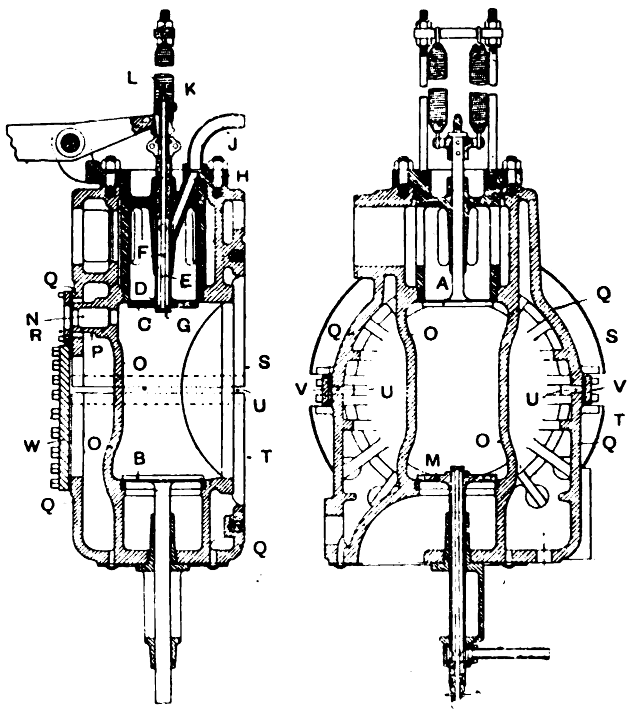


FIG. XII.—86. Combustion Chamber, Hornsby-Stockport.

As will be seen, the height of the combustion chamber is equal to that of the cylinder, but laterally it is narrower. A large water chamber is arranged all round the inner walls, and a cleaning door of ample size is fitted at the back. The ignition gear is placed either at the upper part in the immediate vicinity of the inlet valve, or upon the side close to the piston. The excess of oil is blown out by the exhaust.

It will be seen that at each side of the outer wall a gap is left in the outer casing to permit expansion. These gaps are filled by cover plates bolted on making a flat joint to retain the water in the jacket.

Compression Chambers.—On p. 165 mention has been made of the fact that compression pressures from 140 lbs. to 170 lbs. are the limits which it is not prudent to exceed in practice for producer-gas engines. From a given compression pressure, the volume of the compression chamber is determined according to the volume of the engine cylinder.

Compression of gas in an enclosing vessel which is assumed to be impermeable to heat, gives rise to an increase of temperature and pressure, consequent upon the internal heat of compression and of the external work which produces the compression.

Upon expansion of this gas there is a diminution of the internal heat, or, in other words, the heat accumulated by the work of compression is transformed into work. This naturally would bring about a diminution of temperature within the cylinder. To this diminution of temperature there is a corresponding contraction—that is to say, a diminution of volume, and a fall of pressure. This change of state is called "adiabatic" when it is assumed that no heat is either gained or lost.

The equation of the adiabatic curve is:—

$$P\,V^n = \text{constant}$$

$$\text{where } n = \frac{Cp}{Cv} = \frac{\text{Specific heat at constant pressure.}}{\text{Specific heat at constant volume.}}$$

The constant is determined by basing it upon the pressure and volume corresponding to a given state $p_o v_o$. The problem then is to find the volume that the mixture should occupy to have the pressure required.

The following table gives the values of the ratio $\frac{Cp}{Cv}$ for different gases:—

Air	= 1·41
Oxygen	= 1·4026
Nitrogen	= 1·4114
Hydrogen	= 1·4132
Carbon monoxide	= 1·4104
Carbon dioxide	= 1·4090

and for the following mixtures of illuminating gas and air, according to Güldner :—

$$\frac{\text{Gas}\ 1}{\text{Air}\ 6} = 1\cdot3560$$

$$\frac{\text{Gas}\ 1}{\text{Air}\ 9} = 1\cdot3700$$

$$\frac{\text{Gas}\ 1}{\text{Air}\ 12} = 1\cdot3800$$

It would be rational to calculate the volume of compression chambers upon the basis of $PV^{1\cdot38}$, but practice has shown that, usually, the compressions obtained are a little lower than those found by calculation, and it is therefore better to use the expression $PV^{1\cdot3}$ which gives a smaller compression chamber.

By applying the formulæ :—

$$PV^{1\cdot38} = \text{constant}$$
$$PV^{1\cdot3} = \text{constant}$$

the following figures are obtained :—

Terminal Compression Pressure in Atmospheres (1 atm. = 14·23 lbs. per square inch).	Ratio between the Clearance and Piston Displacement Volumes.		Ratio between the Clearance Volume and the Total Volume of Piston Displacement plus Clearance.	
	n = 1·38	n = 1·3	n = 1·38	n = 1·3
2	1·545	1·420	0·606	0·587
3	0·820	0·753	0·450	0·430
4	0·575	0·525	0·365	0·344
5	0·452	0·407	0·312	0·290
6	0·376	0·337	0·278	0·252
7	0·323	0·288	0·244	0·224
8	0·285	0·253	0·222	0·202
9	0·256	0·226	0·204	0·185
10	0·233	0·205	0·189	0·170
11	0·214	0·188	0·176	0·158
12	0·198	0·174	0·165	0·148
13	0·185	0·162	0·156	0·139
14	0·173	0·153	0·148	0·131
15	0·163	0·142	0·140	0·125

Joints.—The joints upon which the tightness of the cylinder depend should be obtained by the direct contact of metal to metal, the surfaces, whenever possible, being ground.

All "ground" joints should be so arranged that the surfaces may be bedded down to each other by rotation of one upon the other after erection of the engine. The rear joint of the liner, which is one of the most important, should prevent all leakage of gas even at the instant of explosion, when the pressure may reach 550 lbs. per square inch, and should be made as explained on p. 303, Fig. XII.—33. The author is of the opinion that except for extensive contact areas, dry joints, metal to metal, with well ground surfaces, are the best.

The joint in front between the exterior of the liner and the internal portion of the jacket should be sufficiently tight to prevent all leakages of the circulating water. It is made either by means of an indiarubber ring, as in Fig. XII.—34, kept in place by a groove between the liner and the jacket, or, for engines of larger dimensions, by an external bolted ring (Fig. XII.—87).

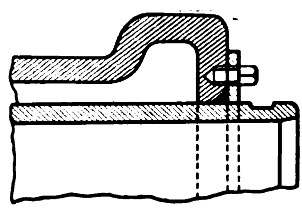


FIG. XII.—87. Front end Liner Joint.

The ignition plugs and the mixture inlet, exhaust, and compressed air valve boxes are generally given an inclination of 30° at least or 45° at most, at the jointing surface. The width of the joint varies from $\frac{1}{4}$ inch at least to $\frac{1}{2}$ inch at most, according to the diameter of the box or plug.

Water Circulation.—The breech ends should be designed to permit ample cooling, and particularly round the exhaust valve. Cleaning doors should be provided, giving access to all parts of the chamber, similar to those arranged on the cylinder water-jackets (Figs. XII.—78, 82, and 86).

Communication between the water-jacket and breech end should be incorporated in the flanges as indicated in Fig. XII.—78. The employment of an external connecting pipe for this purpose should be avoided, because it permits air or vapour to collect in it, and thereby to cause an obstacle to the free circulation. As a remedy, however, a small pet cock can be arranged at the upper part of the bend.

The water outlet should be placed, preferably, near the rear flanges so as to avoid rapid circulation in the front portion of the jacket, which, without inconvenience, can remain stagnant, and be allowed to reach a higher temperature. It should be remembered that in this part, heat is absorbed by the mechanical work of expansion, and therefore all losses to external cooling should be avoided as much as possible.

For engines of 50 to 75 H.P., with compressions of 140 to 170 lbs. per square inch, the jacket of the breech end should be furnished with separate water circulating arrangements to those for the cylinder itself so as to permit control of the temperature for each part.

The exhaust pipe, in the portion contiguous to the breech end, should also be fitted with a water-jacket for engines of over 75 H.P., in order that abnormal temperatures within the engine-room may be avoided, and that the noise of exhaust may be deadened by the diminution of pressure due to such cooling.

Dimensions of Orifices.—The sizes of water circulation pipes are determined by the diameter of the tubes usually stocked by makers. They should be such that, in each section, the velocity of the water in circulation should not be less than 2 feet per second in the case of forced circulation. This speed is taken as a basis; (1) to take into account the throttling of cocks for controlling the flow, (2) to prevent the pressure within the jackets becoming too high which might cause leaky joints, and (3) to allow for eventual reduction of areas through incrustation.

The figures given in the following table are based upon these speeds and for an output of 5·5 gallons per H.P. hour:—

B.H.P.	15	25	35	50	75	100	150
Maximum quantity, gallons per second	·023	·036	·053	·077	·114	·153	·23
Thermo-syphon system inlet bore in inches	2	$2\frac{1}{4}$	$2\frac{1}{2}$	(Separate service for jacket and breech end for these sizes.)			
Thermo-syphon system outlet bore in inches	2	$2\frac{1}{4}$	3				
Forced circulation at 7 lbs. per square inch inlet bore in inches	$\frac{3}{4}$	$\frac{7}{8}$	1	$1\frac{1}{4}$	$1\frac{1}{2}$	$1\frac{1}{2}$	$1\frac{5}{8}$
Forced circulation at 7 lbs. per square inch outlet bore in inches	$\frac{3}{4}$	$\frac{7}{8}$	1	$1\frac{1}{4}$	$1\frac{1}{2}$	$1\frac{1}{2}$	$1\frac{5}{8}$

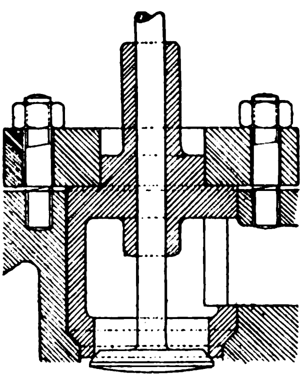


FIG. XII.—88. Removable Valve Seat with separate flange.

Valve Boxes.—In single-acting engines the exhaust valve seats are usually part of the breech end casting, an arrangement which presents many inconveniences, such as the impossibility of making the cylinder gastight without renewing the entire casting if the seat of this valve has a flaw or is eroded, unless the thickness of metal is sufficient to support a separate seat. For engines of over 15 to 20 H.P. the spindle of the exhaust valve is guided by a sleeve passing through the water-jacket to obtain more efficient cooling, and also to permit lubrication without risk of burning the oil.

The inlet valve is fitted with a movable box making a ground joint with the breech casting. In small engines it is a good plan to make provision for the joints to be re-ground after the studs have been placed in position, by making the valve boxes with a separate flange, as shown in Fig. XII.—88. In large engines, the box is similarly made in two parts, one of which is flanged while the other and lower portion can be rotated and thus ground to its seat.

CHAPTER XIII

MOVING PARTS

Fly-wheels and Pulleys.—Fly-wheels should be truly turned and faced on the sides of the rim and boss. For "electric lighting" engines the periphery should be slightly rounded to receive the driving belt. The circumferential speed in this case should not exceed 90 feet per second; usually the speed adopted is from 65·0 to 82·5 feet per second.

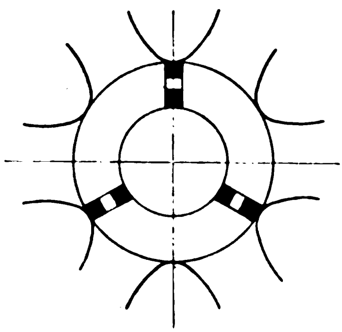


FIG. XIII.—1. Flywheel Boss Cast with Clefts.

The projecting ends of the crank shaft should be safeguarded by the provision of sleeves of sufficient diameter and length to cover the overhanging fly-wheel keys.

For industrial engines of less than 30 H.P., the crank shaft is sometimes fitted with one or two wheels overhanging the main bearings. This arrangement is not so expensive as that with three bearings, but it is less to be commended.

In large engines, as well as those directly coupled to a dynamo, a single heavy fly-wheel supported by a third outside bearing should be fitted. In this case, should the installation comprise a driving pulley this would be placed between the fly-wheel and outer bearing. The pulley should always be in halves, so that it can be removed without lifting the shaft or wheel.

Small fly-wheels are cast in a single piece with curved arms. In some cases to prevent fractures due to contraction after casting, and to facilitate erection, fly-wheels with straight arms are furnished with bosses with one or more clefts, according to the size of the wheel (Fig. XIII.—1). In erection keys are introduced in the clefts and removed before tightening the bolts or putting on the tightening bands.

When the weight exceeds 2½ to 3 tons, a system of barring gear should be employed to permit one man to place the engine in the starting position without pulling on the spokes.

In engines with only one fly-wheel, the latter is generally fitted on the left hand side of the engine when looking towards the crank from the cylinder, the half-speed shaft being on the right. The usual direction of rotation is such that the top of the fly-wheel runs away from the cylinder when viewed from the same position.

Calculations of resistance of material assign the figure of 115 to 130 feet per second as the limit of linear velocity for cast-iron wheels, but in practice it is prudent not to exceed 110 feet.

Fly-wheels of less than 10 feet diameter usually have six arms, above this size they are constructed with eight arms.

The finished fly-wheels weigh 90 to 93 per cent. of their foundry weight.

Coefficient or Degree of Irregularity.—With engines of the electric lighting type driving dynamos, the light from incandescent lamps should be perfectly steady, whether the engines be working at full load or one-third load, without the intervention of a battery. To obtain this result the engine should have a cyclic regularity, such that the variations of amperes and volts as shown by aperiodic instruments do not exceed 2 per cent.

The method of governing by admission at each cycle permits the use of less heavy fly-wheels than in the case of "hit-and-miss" governing. Double-acting or multiple-cylinder engines are, for the same reason, preferable to the single-acting engines.

The coefficient or degree of irregularity, a, is chosen in accordance with the use to which the motor is to be put, and represents maximum velocity — minimum velocity ÷ mean velocity.

The values of the coefficient, a, adopted for various applications are as follows:—

$\frac{1}{30}$ to $\frac{1}{40}$, for industrial engines, constructional workshops, wood-working machinery, pump driving, &c.

$\frac{1}{60}$ for continuous current dynamos driven by intermediate gearing.

$\frac{1}{80}$ for belt-driven dynamos, working in series.

$\frac{1}{80}$ for alternators in series, driven by intermediate gearing.

$\frac{1}{90}$ for continuous current dynamos, working in parallel, driven by intermediate gearing.

$\frac{1}{100}$ for belt-driven alternators in parallel.

$\frac{1}{120}$ for belt-driven, continuous current dynamos, working in parallel.

$\frac{1}{120}$ for alternators coupled in series, driven by intermediate gearing.

$\frac{1}{120}$ for continuous current dynamos, working in series, and direct coupled to the engine shaft.

$\frac{1}{180}$ for belt-driven alternators in parallel.

$\frac{1}{200}$ for continuous current dynamos, working in parallel, and direct coupled to engine shaft.

$\frac{1}{200}$ for alternators, working in series, direct coupled to engine shaft.

$\frac{1}{200}$ for cotton-spinning and similar industries.

$\frac{1}{250}$ for alternators, working in parallel, and direct coupled to engine shaft.

As will be noted from the following equation, the formula upon which fly-wheel calculations are based, includes an empirical coefficient K, which varies according to the type of engine.

The values of K for the different conditions are :—

For four-cycle engines,			single-cylinder, single-acting	$K = 475{,}000$
„	„	„	two cylinders, single-acting, *vis-à-vis*, upon the same crank pin	$K = 300{,}000$
„	„	„	one cylinder, double-acting, Fig. V.—4, charts II., III. and IV.	
„	„	„	two cylinders, single-acting, tandem, twin, or opposed, with cranks at 180°, Fig. V.—4, charts V., VI. and VII.	$K = 225{,}000$
„	„	„	four cylinders, twin, opposed or tandem, single acting	$K = 75{,}000$
„	„	„	two cylinders, double-acting, tandem or twin, Fig. V.—5, charts VIII, IX., X. and XI.	

Fly-wheel Calculations.—*Weight.*—The following formula may be used to determine the dimensions of the fly-wheels for the different types of engines :—

$$W.D^2 = K\frac{B.h.p.}{a.n^3}$$

from which :—

$$W = K\frac{B.h.p.}{D^2.a.n^3}$$

W = Weight of the rim (without arms or boss) in tons.
D = Diameter of the centre of gravity of the rim in feet.
a = Coefficient or degree of cyclic irregularity.
n = Revolutions per minute.
$B.h.p.$ = Brake horse power.
K = Coefficient varying with the type of engine.

The *total weight of the fly-wheel* would be about

$$W_1 = 1{\cdot}4W.$$

As examples, calculations will be made and tabulated for a series of single-acting engines, assuming $K = 475{,}000$

„ $a = \frac{1}{120}$ corresponding to the practical coefficient of irregularity for continuous current dynamos, working in parallel, driven by a belt from the fly-wheel or pulley.

$a = \frac{1}{80}$ for electric lighting engines, driving belt-driven dynamos coupled in series.

$a = \frac{1}{40}$ for ordinary industrial engines.

The *outside diameter of the fly-wheel rim* is obtained from the following equation :—

$$D_1 = 19{\cdot}1\,\frac{v}{n}$$

in which v = circumferential velocity of the fly-wheel in feet per second.

v = not exceeding 82·5 feet per second if used for belt-driving.

v = „ „ 105·0 feet „ „ if belt is driven by a separate pulley.

v = ranging from 82·5 feet „ „ for 15 to 50 H.P. engines.
to 100·0 feet „ „ „ 75 to 150 „ „

$D_1 = D \times 1{\cdot}1$ approximately, varying according to the rim section.

In the following table the *minimum width of belt* is noted for the different size fly-wheels and powers, as deduced from the equation.

$$B = 19{\cdot}3\,\frac{B.h.p.}{v} = \text{width of single belt about } \tfrac{1}{4} \text{ inch thick.}$$

$$B = 9{\cdot}0\,\frac{B.h.p.}{v} = \text{width of double belt about } \tfrac{3}{8} \text{ to } \tfrac{1}{2} \text{ inch thick.}$$

in which—

$B.h.p.$ = Brake horse power transmitted.
v = Circumferential velocity of wheel in feet per second.
B = Width of belt in inches.

The *section of the rim in square inches* is given as a function of the weight by the equation :—

$$S = 228{\cdot}5\,\frac{W}{D}$$

in which—

W = Weight of rim in tons.
D = Diameter of centre of gravity of rim in feet.

The shapes given to this section are very variable and are determined as fancy dictates. No general rule is applicable, and the depth

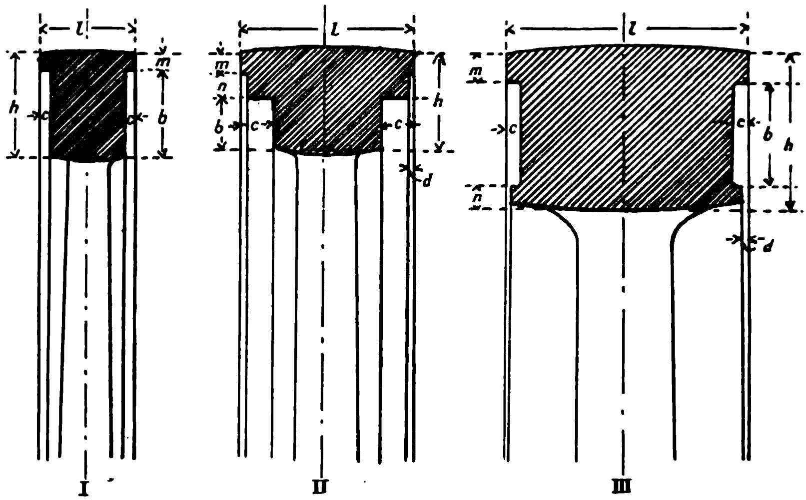


FIG. XIII.—2. Fly-wheel Rim Sections.

and breadth are necessarily a function of the selected shape. For the purpose of calculation three simple forms have been adopted, as shown

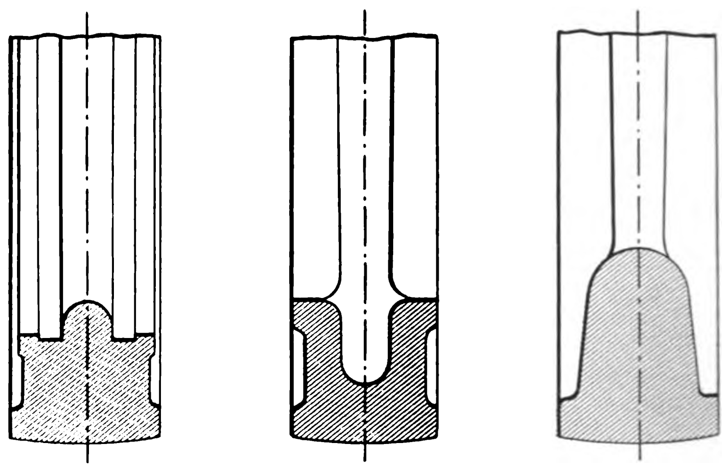


FIG. XIII.—3. Fly-wheel Rim Sections.

in Fig. XIII.—2, and, for further information, three other forms adopted by different makers (Fig. XIII.—3) have also been illustrated.

The tables give the dimensions of the rim for Sections I., II., III., of Fig. XIII.—2.

FLY-WHEELS.

Weight, Rim Speeds, Section of Rims.

B.H.P. Brake Horse Power.	15	25	35	50	75	100	150
n. Revolutions per minute	230	220	210	200	190	180	160
v. Circumferential speed feet per second	78·3	80·6	82·5	81·0	99·5	101·3	100·5
D_1. External diameter $= 19{\cdot}1 \frac{v}{n}$ feet	6·5	7·0	7·5	7·75	10·0	10·75	12·0
D. Diameter centre of gravity of rim $= \frac{D_1}{1{\cdot}1}$ feet	5·9	6·4	6·8	7·05	9·1	9·8	10·9
B. Width belt inches $= 19{\cdot}3 \frac{B.h.p.}{v}$ single ($\frac{1}{4}''$) inches	3·75	6·0					
$= 9{\cdot}0 \frac{B.h.p.}{v}$ double ($\frac{1}{2}''$) inches	—	—	4·0	5·5			
Coefficient of irregularity $= \frac{1}{120} = a$.							
W. Weight of rim (approx.) $= \frac{475{,}000\ B.h.p.}{D^2\ a\ n^3}$ . . . tons	—	—	—	—	7·5	10·2	17·5
W_1. Total weight of wheel $= 1{\cdot}4\ W$ . . . . . tons	—	—	—	—	10·5	14·25	24·5
S. Sectional area of rim $= 228{\cdot}5 \frac{W}{D}$. square inches	—	—	—	—	188·5	238·0	367·0
Type of section (Fig. XIII.—2)	—	—	—	—	III.	III.	III.
Coefficient of irregularity $= \frac{1}{80} = a$.							
W. . . . . . . tons	1·35	2·18	3·1	4·75	5·0	6·8	11·7
W_1. . . . . . . tons	1·89	3·05	4·34	6·65	7·0	9·5	16·5
S. . . . square inches	52·3	77·5	104·0	154·0	125·0	158·5	245·0
Type	I.	I.	II.	II.	II.	III.	III.
Coefficient of irregularity $= \frac{1}{40} = a$.							
W. . . . . tons	·675	1·09	1·55	2·375	2·5	3·4	5·85
W_1 . . . . tons	·945	1·525	2·17	3·325	3·5	4·75	8·25
S. . . . square inches	26·15	38·75	52·0	77·0	62·5	79·25	122·5
Type	I.	I.	I.	I.	I.	II.	II.

DIMENSIONS OF FLY-WHEEL RIM.

Section I. Fig. XIII.—2.

B.H.P.	Coefficient of Irregularity.	l	h	m	c	n	d	b	Area Square Inches.
15	$\frac{1}{40}$	5·5	6·05	1·3	·75	—	—	—	26·15
25	,,	6·75	7·0	1·5	·75	—	—	—	38·75
35	,,	8·0	8·5	1·25	1·0	—	—	—	52·0
50	,,	9·75	9·875	1·375	1·125	—	—	—	77·0
75	,,	8·33	9·5	1·5	1·0	—	—	—	62·5
15	$\frac{1}{80}$	8·0	8·5	1·25	1·0	—	—	—	52·0
25	,,	9·75	9·875	1·375	1·125	—	—	—	77·0

Section II. Fig. XIII.—2.

B.H.P.	Coefficient of Irregularity.	l	h	m	c	n	d	b	Area Square Inches.
100	$\frac{1}{40}$	13·75	7·5	1·5	2·375	1·5	·875	4·5	79·15
150	,,	14·25	11·5	1·5	2·875	1·5	·375	9·5	122·25
35	$\frac{1}{80}$	16·25	9·0	2·0	3·5	2·25	2·0	4·75	104·0
50	,,	19·5	12·0	2·0	4·75	2·75	2·0	7·25	154·0
75	,,	17·375	9·0	2·0	2·4375	2·75	1·9375	4·25	125·0

Section III. Fig. XIII.—2.

B.H.P.	Coefficient of Irregularity.	l	h	m	c	n	d	b	Area Square Inches.
100	$\frac{1}{80}$	16·25	11·25	2·0	1·375	2·0	1·0	7·25	158·8
150	,,	20·0	15·0	2·5	1·5	2·5	1·0	10·0	245·0
75	$\frac{1}{120}$	18·0	11·5	2·75	1·5	2·0	1·0	7·0	188·5
100	,,	20·0	13·0	2·5	1·5	2·0	1·25	9·0	238·0
150	,,	24·0	16·5	3·5	1·875	3·0	·5	10·0	367·0

Arms or Spokes.—As already mentioned, wheels of less than 10 feet diameter are made with six arms, while above this eight arms are provided.

The sectional area of the arms near the rim are calculated by the following empirical formula for wheels with six arms :—

$$A = \frac{W\,n^2\,D}{15{\cdot}8} \times \left(\frac{1}{1400} \text{ to } \frac{1}{1700}\right)$$

and those with eight arms—

$$A = \frac{W\,n^2\,D}{21{\cdot}0} \times \left(\frac{1}{1400} \text{ to } \frac{1}{1700}\right)$$

according to the practice adopted for taking the figures of 1,400 to 1,700 lbs. per square inch, the limits recommended for fly-wheel calculations. In the above formulæ:—

A = Area of the arms in square inches.
W = Weight of rim in tons.
n = Revolutions per minute.
D = Diameter of centre of gravity of rim in feet.

The arms are given an elliptical form, the smaller axis being about

Dimensions of Fly-wheel Spokes.

Based on resistance of 1,700 lbs. per square inch.

B.H.P.	15	25	35	50	75	100	150
Number of Spokes.	6	6	6	6	6	6	8
n = Revolutions per minute	230	220	210	200	190	180	160
D = Diameter, centre of gravity of rim . . feet.	5·9	6·4	6·8	7·05	9·1	9·8	10·9
Coefficient of Irregularity $\frac{1}{150}$							
W = Weight of rim . tons.	—	—	—	—	7·5	10·2	17·5
A = Sectional area of spokes *near rim* sq. ins.	—	—	—	—	92	120	137
a = Length *major* axis $a = \sqrt{\dfrac{A}{\cdot 7854 \times \cdot 6}}$ ins.	—	—	—	—	14	16	17
b = Length *minor* axis $b = \cdot 6a$ . . ins.	—	—	—	—	8·375	9·6	10·25
A_1 = Sectional area *near boss* $A_1 = 1{\cdot}45\ A$ sq. ins.	—	—	—	—	133·75	174·	196·
a_1 = Length *major* axis ins.	—	—	—	—	16·875	19·25	20·5
b_1 = ,, *minor* ,, . ins.	—	—	—	—	10·125	11·625	12·375
Coefficient of Irregularity $\frac{1}{80}$							
W = ... tons.	1·35	2·18	3·1	4·75	5·0	6·8	11·7
A = ... sq. ins.	15·7	25·2	34·6	50·	61·	80·5	91·5
a = ... ins.	5·75	7·3125	8·5	10·3125	11·375	13·25	14·0
b = ... ins.	3·5	4·5	5·125	6·125	6·75	7·875	8·375
A_1 = ... sq. ins.	22·8	36·5	50·	72·5	88·5	117·	132·5
a_1 = ... ins.	7·0	8·75	10·3125	12·5	13·75	15·75	16·812
b_1 = ... ins.	4·25	5·25	6·125	7·5	8·25	9·375	10·0
Coefficient of Irregularity $\frac{1}{50}$							
W = ... tons.	·675	1·09	1·55	2·375	2·5	3·4	5·85
A = ... sq. ins.	7·85	12·6	17·3	25·	30·5	40·25	45·75
a = ... ins.	4·125	5·25	6·125	7·3125	8·0	9·125	9·75
b = ... ins.	2·5	3·125	3·625	4·375	4·8125	5·5	5·875
A_1 = ... sq. ins.	11·375	18·25	25·1	36·3	44·2	58·5	66·5
a_1 = ... ins.	4·8125	6·25	7·3125	8·6875	9·75	10·75	11·875
b_1 = ... ins.	2·875	3·75	4·375	5·25	5·8125	6·5	7·125

0·6 of the larger axis. The larger axis is disposed at right angles to the centre line passing through the length of the boss.

The dimensions of the arms near the boss are calculated from:—

$$A_1 = 1{\cdot}4 \text{ to } 1{\cdot}5A$$

Subject to verification that the sum of the tensile and bending stresses does not exceed 1,400 to 1,700 lbs. per square inch for this section.

Boss.—The diameter at the extremities of the boss is 2·0 δ.

The diameter at the centre of the boss is 2·1 δ to 2·4 δ.

δ = diameter of the thickened portion of the crank shaft upon which the fly-wheel is placed.

The length of the boss can be taken as equivalent to 1·7 δ to 2·0 δ.

Fly-wheels in Halves.—For convenience in transit and erection, fly-wheels exceeding five tons in weight should be made in two sections, and coupled by means of four or two bolts *and* two bands, or by either bolts *or* bands.

The calculation of the coupling to resist the tension due to centrifugal force acting on half the fly-wheel is based on the formula :—

$$A = \left(\frac{1}{4250} \text{ to } \frac{1}{5000}\right) \frac{W n^2 D}{4 \cdot 8}$$

in which :—

A = Sectional area of bolts and bands in square inches.
W = Weight of fly-wheel rim in tons.
D = Diameter of centre of gravity of rim in feet.
n = Revolutions per minute.

When both bolts and bands are used the sectional area of the latter is taken as 60 per cent., and the former 40 per cent., of the total area found by the equation.

BOSSES, BOLTS AND CLAMPING RINGS FOR FLY-WHEELS IN HALVES.

EFF. H.P.	75	100	150	75	100	150	150
a. Coefficient of irregularity	$\frac{1}{120}$	$\frac{1}{120}$	$\frac{1}{120}$	$\frac{1}{80}$	$\frac{1}{80}$	$\frac{1}{80}$	$\frac{1}{40}$
w. Weight of rim in tons .	7·5	10·2	17·5	5·0	6·8	11·7	5·85
D. Diameter centre of gravity, feet . . .	9·1	9·8	10·9	9·1	9·8	10·9	10·9
n. Revolutions per minute .	190	180	160	190	180	160	160
A. Total sectional area of bolts and rings, square inches	102·5	135·0	203·0	68·5	90·0	136·0	68·0
·6A. Sectional area each of two rings, square inches	61·25	81·0	122·0	41·25	54·0	82·0	41 0
Dimension for each ring, breadth = depth, inches	4·0	4·5	5·5	3·25	3·7	4·5	3·25
·4A. Sectional area of 4 bolts square inches . .	41·25	54·0	81·0	20·625	36·0	54·0	27·0
Diameter of each bolt . .	3·625	4·1875	5·125	2·5625	3·375	4·1875	3·0

Fly-wheel Couplings.—Fly-wheels made in two or more separate pieces are coupled at the boss and rim in one or other of the methods shown in Figs. XIII.—4 to 10. Fig. XIII.—4 shows an arrangement with internal clamping rings and tongued side plates as adopted by

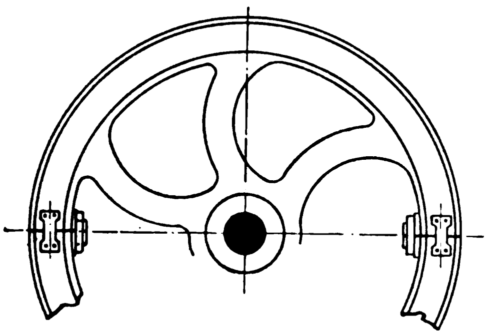


FIG. XIII.—4. Fly-wheel Rim Couplings.

several makers. Both the methods of flanges and bolts (Fig. XIII.—5) and of flush wedges in the rim, tightened by keys (Fig. XIII.—10) are good. The arrangement shown in Fig. XIII.—9 is defective because the maximum stress is produced at the periphery of the rim, whilst the coupling bolts are applied inside the rim. Fig. XIII.—6 shows how

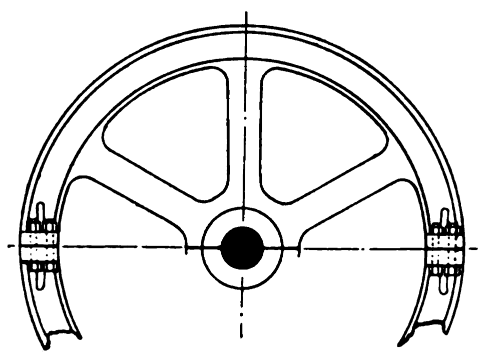


FIG. XIII.—5. Fly-wheel Rim Couplings.

this arrangement may be improved by the addition of a tongued piece of steel extending the whole width of the periphery. This method has been patented by Messrs. Crossley Brothers, Ltd.

Coupling by means of bolts passing through the arms as shown in Fig. XIII.—8 diminishes the resistance by about one-half. It is therefore indispensable to add four clamping rings at the rim.

Bosses coupled by bolts passing through the middle of the metal (Fig. XIII.—7) are preferable to those with bolts through side bearers (Fig. XIII.—10). It is advisable to use clamping rings for couplings, as these, being fitted when hot, either on boss or rim over projecting

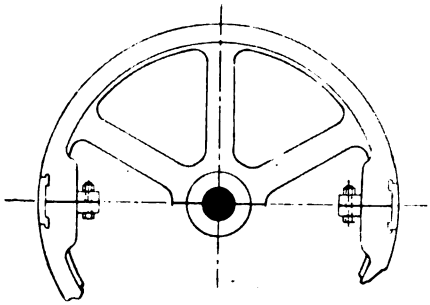


FIG. XIII.—6. Fly-wheel Rim Couplings.

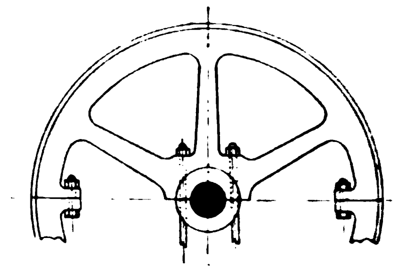


FIG. XIII.—7. Fly-wheel Rim Couplings.

portions, set up a tension in the metal, the amount of which is beyond control.

In fly-wheels made in several pieces the resistance of the coupling is only one-fourth or one-fifth as regards the boss, and about two-

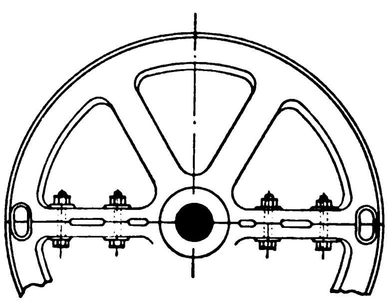


FIG. XIII.—8. Fly-wheel Rim Couplings.

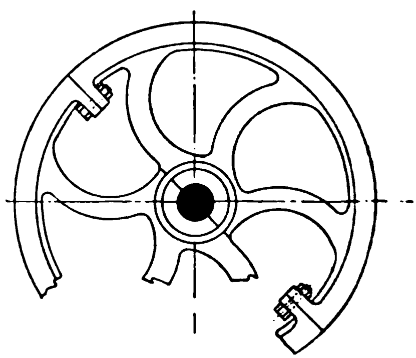


FIG. XIII.—9. Fly-wheel Rim Couplings.

thirds with respect to the rim, of that of the solid cast-iron. That is to say, the amount of resistance of the cast-iron in these portions is increased in inverse proportion to the ratios mentioned. It is therefore necessary to make allowances accordingly.

Fly-wheel Keys.—No lateral movement of the fly-wheel can be permitted during work, and therefore extreme care is required in keying them to the shaft.

The simple key for a flat on the shaft, without groove, should never be used (Fig. XIII.—11). Neither should the round key let into the shaft be employed (Fig. XIII.—14). For the lighter wheels and those

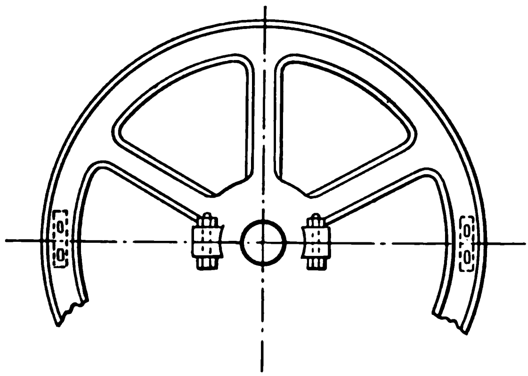


FIG. XIII.—10. Fly-wheel Rim Couplings.

of less than 8 feet diameter a single key placed in a keyway is generally deemed sufficient (Fig. XIII.—12). Above this size and for heavy wheels it is better to use two keys fitted at 90°.

When tangential keys are fitted, which are to be preferred, two keys are placed at 120° (Fig. XIII.—13). A tangential arrangement

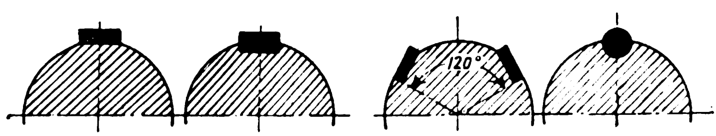


FIG. XIII.— 11. 12. 13. 14.
Fly-wheel Keys.

is shown in Fig. XIII.—15, with double keys, C. The fly-wheel bosses are bored eccentric, following a semi-circle traced by the radius

$$R = \frac{D}{2}.$$

The eccentricity (distance between the centres A and O) is from ·04 to ·08 of an inch. The spaces shown in Fig. XIII.—18 round the shaft are provided in the middle portion of the boss, the two extremities being of correct diameter to come into contact with the crank shaft.

The big end of key is usually taken as 0·24 times the diameter of the shaft, and the small end 0·66 of the end.

The Foos Gas Engine Co., of Springfield, U.S.A., who make their small wheels with cleft bosses, adopt the following method of fixing:

The coupling bolts being loosened, iron wedges are placed in the clefts to slightly separate the two halves of the boss. The wheel is then placed on the shaft and the wedges removed. After examination to ensure that the boss is properly seating, the bolts are fully tightened. Oil is then put in the keyway and on the key to prevent

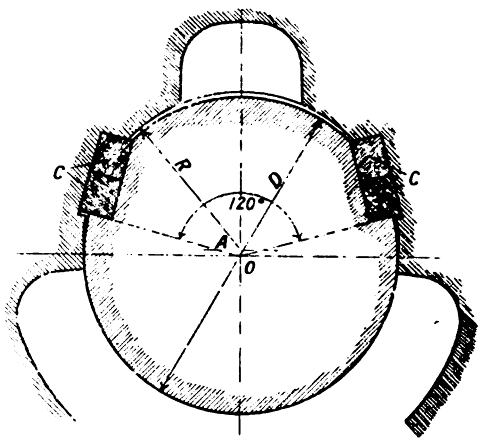


FIG. XIII.—15. Fly-wheel Keys.

its binding, and the key driven right home. According to the practice of some makers, white lead soaked in oil is preferable to oil only.

Sundry Examples.—Mr. H. A. Humphrey, in one of his papers, gives the weights of fly-wheels according to the type and arrangement of gas engine cylinders, taking, for comparison, a steam engine of the same power, same diameter of fly-wheel, and running at the same number of revolutions per minute. In order to realise equal cyclic irregularity, the fly-wheel weights are given as follows :—

	Tons.
Single-cylinder steam engines, double-acting . .	10
,, ,, gas ,, single-acting, four-cycle	62·8
Two-cylinder ,, ,, opposed cylinders, single-acting, four-cycle .	40

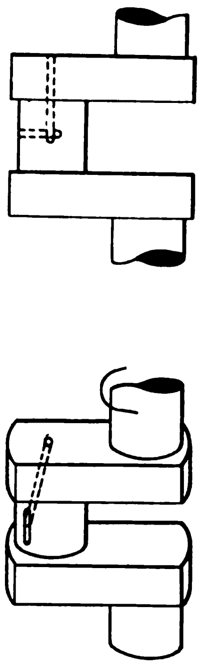
			Tons.
Single-cylinder gas engines,	double-acting,	four-cycle .	40
Two-cylinder „ „		twin, on the same crank angle	30
Four-cylinder „ „		twin tandem, working on same crank angles . .	30
„ „ „ „		twin, *vis-à-vis*, opposed cranks	10
Two-cylinder „ „		tandem, double-acting, four-cycle	10

Dingler gives the following figures for gas engine fly-wheels:—

EFF. H.P.	Revolutions per Minute.	External Diameter of Wheel, Feet.	Width of Wheel, Inches.	Ordinary Service.		Electric Lighting.	
				Weight, Tons.	Coefficient of Irregularity.	Weight, Tons.	Coefficient of Irregularity.
Single-acting engines.							
		ft. in.					
65	180	10 6	12	2·75		6·2	
80	180	10 6	12	3·44		7·66	
110	160	12 0	12	5·1	$\frac{1}{40}$	11·8	$\frac{1}{70}$
140	160	12 0	14	6·875		14·75	
180	150	12 6	14	9·05		21·1	
Double-acting engines.							
		ft. in.					
200	160	12 0	17·7	5·9		8·35	
260	160	12 0	19·7	7·5		10·8	
335	150	12 6	24·8	9·8	$\frac{1}{70}$	14·25	$\frac{1}{100}$
390	140	13 4	17·0	12·8		18·3	
510	125	15 0	20·8	19·15		27·0	
Tandem double-acting engines.							
		ft. in.					
365	160	12 0	17·7	4·9		8·35	
520	160	12 0	20·8	6·5		10·8	
670	150	12 6	24·8	8·55	$\frac{1}{100}$	14·25	$\frac{1}{150}$
780	140	13 4	29·5	10·8		18·3	
1020	125	15 0	43·3	16·7		27·0	

The Cockerill Co., for their high-speed electric light type of engines, use cast-iron rims with rigidly connected forged steel arms.

Crank Shafts.—The crank shafts should be made of best quality mild steel with throws slotted out. Siemens-Martin steel or crucible cast-steel is used, having a limit of resistance of 65,000 to 70,000 lbs. per square inch, with an elongation of 18 to 20 per cent. For very large engines nickel steel (1 to 3 per cent.) is successfully employed, having



FIGS. XIII.—15 and 16.
Crank-pin Lubricator (correct). Crank-pin Lubricator (incorrect).

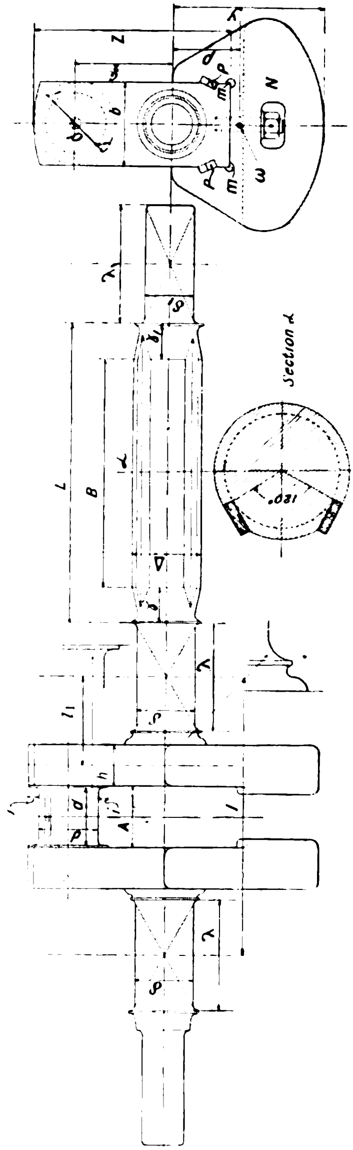


FIG. XIII.—17. Detail of Crank Shaft.

a resistance of 100,000 lbs. per square inch, with 20 to 22 per cent. elongation.

The shaft should be strong enough to resist bending when subjected to the stresses set up by premature explosions, which, in some instances, may reach 640 lbs. per square inch.

The portion of the shaft running in the side bearings should have oil throwers turned in the metal to recover the oil coming from the journal and to prevent splashing or creeping of the oil to the fly-wheel, pulley, or crank webs. For details see pp. 292 and 294.

For engines with a single fly-wheel the shaft is strengthened by being made of a larger diameter where the wheel is supported, as well as for the driving pulley, or, if necessary, a dynamo armature. This swelling either tapers off on each side conically or by a well-rounded fillet.

The crank pin is a difficult part to lubricate. If oil rings are fitted to one of the crank webs the crank pin should be drilled obliquely by a passage opening to an oil groove cut in the pin as shown in Fig. XIII.—15. A great many makers form this passage by two holes drilled at right angles as Fig. XIII.—16. This arrangement does not lend itself to cleaning as well as the first.

The crank pin lubrication must be provided for by a passage formed of various holes starting from the free end of the crank shaft to open into the pin itself. The oil flows into a small basin provided for the purpose.

For engines over 100 H.P. it is advisable to use a duplicate system by fitting a ring on each of the crank webs.

The outlet of the oil to the pin should be placed in the position shown in Figs. XIII.—15 and 17, and not as in Fig. XIII.—16, because, in the latter case, the shock which is produced at the moment of the explosion has a tendency to force the oil back into the passage.

For cranks of over 12 or 14 inch radius as small a ring as is possible is used, fed by a tube to the beginning of the hole drilled in the web as shown in Fig. X.—39. By this means the centrifugal force acts upon all the column of oil contained in the tube and causes a greater pressure.

The shaft represented in Fig. XIII.—17 shows the form of the details for the oil throwers and for the fillets at the dangerous sections. It must be remembered that the distance between the crank webs and the portion forming the journals in the side bearings must be reduced to a minimum. This distance depends upon the thickness of the crank pin oiling ring. It is necessary to provide a slight clearance to permit access to this ring.

Dimensions.—Length L—seating for fly-wheel and pulley—is determined by the length of the swelling B, and also of the two conical portions γ and γ_1, each of which may be taken as being equal to about 25 per cent. of the width of the fly-wheel.

B is given by the width of the fly-wheel rim and that of the driving pulley, if the fly-wheel is not used for transmitting the power.

In connection with the type of engines that are under consideration it will be assumed that the calculation for the crank shaft is made on the basis of the greatest cyclic regularity, that is, $a = \frac{1}{120}$ for engines of 75, 100 and 150 H.P.; and, $a = \frac{1}{80}$ for 15 to 50 H.P.

For the former it is not advisable to use the fly-wheel for belt driving if the peripheral speed reaches 100 feet per second and it is necessary to provide a pulley the width of which should be

$$W = 9\,\frac{B.h.p.}{V} + 1 = \text{inches},$$

$9\,\dfrac{B.h.p.}{V}$ being the width of the belt of $\frac{3}{8}$ to $\frac{1}{2}$ inch thick, as mentioned on p. 345, and V being taken as 80 feet per second.

The space between the fly-wheel and pulley should be at least $\frac{3}{4}$ inch, the fly-wheel being fitted on the side nearest to the engine frame.

For the 15 to 50 H.P. engines, it is assumed that the fly-wheel is used to transmit the power by a belt, and that no pulley exists.

The total length of the portion marked B is about 40 per cent. greater than the width of the fly-wheel.

Upon these assumptions the following values of B in inches are given :—

B.H.P.	Width.		Space.	40 per cent. of Fly-wheel Width.	Total Length of B.
	Fly-wheel.	Pulley.			
Degree of irregularity $a = \frac{1}{80}$.					
15	8·0	—	—	3·25	11·25
25	9·75	—	—	3·875	13·625
35	16·25	—	—	6·5	22·75
50	19·5	—	—	7·75	27·25
Degree of irregularity $a = \frac{1}{120}$.					
75	18	9·5	1	7·25	35·75
100	20	12·0	1	8·0	41·5
150	24	18·0	1	9·5	52·5

From the above the values of L are deduced by adding about one-half the fly-wheel width to the measurements determined for B to take into account the two tapered portions γ and γ_1.

B.H.P.	B.	s and s_1.	Total Length of L, Inches.
15	11·25	3·75	15
25	13·625	4·375	18
35	22·75	7·25	30
50	27·25	9·75	37
75	35·75	8·25	44
100	41·5	9·5	51
150	52·5	11·5	64

Pitch of Bearings.—The distance between the bearings l (Fig. XIII.—17) is given either by the outline of the frame which is determined by the method of lubrication for crank pin, the space for rings, the shape of bearings, &c.; or empirically by taking it as the equivalent of 1·8 to 2·0 times D (D being the piston diameter).

The distance between the centre of crank web and centre of adjacent bearing l_1 (Fig. XIII.—17) is:—

$$l_1 = \frac{l - d}{2{\cdot}4 \text{ to } 2{\cdot}44}$$

in which l = distance between centres of main bearings
d = length of crank pin bearing,
or, in function of piston diameter D,

$$l_1 = 0{\cdot}56 \text{ to } 0{\cdot}64\ D.$$

Crank Webs.—The *thickness of the webs* h (Fig. XIII.—17) is determined by the equation:—

$$h = 0{\cdot}45 \text{ to } 0{\cdot}50\ l_1,$$

account having been taken of the compression and bending moment at the dead centre, and, by taking respectively for the sum of the corresponding stresses, a practical resistance not exceeding 10,000 to 12,000 lbs. per square inch,
or, in function of piston diameter, D:—

$$h = 0{\cdot}25 \text{ to } 0{\cdot}32\ D$$

or, in function of crank pin diameter, d:—

$$h = 0{\cdot}52 \text{ to } 0{\cdot}75\ d.$$

The *depth of the webs* b is generally taken as:—

$$b = 0{\cdot}45 \text{ to } 0{\cdot}64\ D.$$

For the values of h and b the lowest corresponds to $l = 1{\cdot}8\ D$ and the highest to $l = 2\ D$.

The value of b should be such that by introducing it into Bach's formula (see p. 363), for the torsion and bending moments, the metal is not exposed to more than 8,500 to 9,250 lbs. per square inch.

The *height of the webs* Z is determined by the radius $G/2$ and the diameters of the crank pin d and of the journals δ (Fig. XIII.—17).

The dimension i is taken as the distance between the arc described from the centre line of the shaft to meet the sides of the webs.

The underside of the web is made square, so as to facilitate the fitting of the balance-weights. The distance between this squared end of the webs and the under side of the shaft itself is taken as $= 2\ i$ (about).

CRANK WEBS. DIMENSIONS IN INCHES.

B.H.P.	15	25	35	50	75	100	150
Piston diameter D .	8	$10\frac{1}{4}$	12	$13\frac{3}{4}$	$16\frac{1}{2}$	$18\frac{3}{4}$	$22\frac{1}{2}$
Thickness $h = 0{\cdot}28\ D$	$2\frac{1}{4}$	$2\frac{7}{8}$	$3\frac{3}{8}$	$3\frac{7}{8}$	$4\frac{5}{8}$	$5\frac{1}{4}$	$6\frac{1}{4}$
Depth $b = 1{\cdot}9\ h$.	$4\frac{1}{4}$	$5\frac{1}{2}$	$6\frac{1}{2}$	$7\frac{3}{8}$	$8\frac{3}{4}$	10	12

Diameter of crank pin d. This is obtained from the equation:—

$$d = 0{\cdot}38\ \sqrt[3]{D^2\ l}$$

in which:—

D = Piston diameter in inches.
l = distance between centres of bearings in inches.

The formula takes into account an initial pressure on the piston of 430 lbs. per square inch, and a practical resistance of the metal of about 15,650 lbs. per square inch.

If the value of d is expressed in function of D, the diameter of crank pin would be:—

$$d = 0{\cdot}43 \text{ to } 0{\cdot}48\ D.$$

Length of crank pin is taken as equal to the diameter d.

The pressure per square inch upon the projected area $d \times d$ should not exceed 1,700 lbs.

The width between the webs A is equal to $d + {\cdot}1$ to ${\cdot}15$ inches.

Journals of Main Bearings.—Direct calculation of the diameters and lengths of these bearings is a somewhat tedious process, owing to the complicated formulæ applied for determining the resistances of the metal to torsion and bending. Usually, therefore, a definite measurement is taken and verified to see that the dimension corresponds to the limits imposed by practice for the metal employed.

In this way the diameter of the *main bearing journals* is given by

$$\delta = 0{\cdot}90 \text{ to } 0{\cdot}95\, d,$$

d = crank pin diameter as previously determined,

the length, λ, is taken as $= 2\,\delta$.

The product $2\,\delta$ or $2\,\delta^2$ in square inches should be such that the specific pressure per square inch $\frac{\rho}{2\,\delta^2}$ shall not exceed 500 to 570 lbs.

The total pressure ρ is given by the equation :—

$$\rho = \sqrt{\left\{W_1\left(1 - \frac{x}{L + \frac{\lambda_1}{2} + \frac{\lambda_2}{2}}\right)\right\}^2 + \left(167{\cdot}5\, D^2\right)^2}$$

in which :—

W_1 = total weight of fly-wheel in lbs.
x = the distance between the centre of bearing and the centre of fly-wheel when in its permanent position, in inches.
D = piston diameter in inches.
L = space between the main bearings and outer bearings.
λ_1 and λ_2 = length of journals in these bearings.

Journal of Outer Bearing.—For the sake of uniformity, some makers give the outer bearing the same leading dimensions as those in the main frame. Smaller dimensions can, however, be provided.

OUTER BEARING. JOURNALS. MINIMUM DIMENSIONS.

B.H.P.	15	25	35	50	75	100	150
1. $x_1 = \frac{L}{2} + \frac{\lambda_1}{2}$.	10·8125	13·25	20·	24·25	39·25	46·75	60·3125
2. $L + \frac{\lambda}{2} + \frac{\lambda_1}{2}$.	21·625	26·5	40·	48·5	57·5	66·5	82·625
3. Weight of fly-wheel . tons	1·89	3·05	4·34	6·65	10·5	14·25	24·5
4. Value of ρ_1 . lbs.	2,120	3,420	4,850	7,450	7,460	9,500	14,800

Crank shaft Dimensions, Crank pin, and Journals.

		B.H.P.	15	25	35	50	75	100	150
1.	a	Coefficient of irregularity	$\frac{1}{80}$	$\frac{1}{80}$	$\frac{1}{80}$	$\frac{1}{80}$	$\frac{1}{120}$	$\frac{1}{120}$	$\frac{1}{120}$
2.	w_1	Total weight of fly-wheel . . . tons (p. 347)	1·89	3·05	4·34	6·65	10·5	14·25	24·5
3.	D	Piston diameter . . . inches	8·	10·25	12·	13·75	16·5	18·75	22·5
4.	A	Piston area . . . square inches	50·265	82·516	113·098	148·490	213·825	276·117	397·609
5.	Pi	Total explosion pressure at 430 lbs. per square inch	21,600·	35,500·	48,500·	64,000·	92,000·	118,500·	170,000·
6.	d	Crank pin, diameter and length = 0·43 to 0·48 D	3·625	4·625	5·375	6·25	7·375	8·5	10·125
7.	$d \times d$	Projected area . . . square inches	13·15	21·4	29·	39·	54·5	72·5	102·5
8.		Specific pressure per square inch $\frac{\text{line 5}}{\text{line 7}}$ . . lbs. (not to exceed 1,700 lbs. per square inch)	1.640·	1,660·	1,670·	1,640·	1,680·	1,635·	1,655·
9.		Diameter journals of main bearings δ = 0·90 to 0·95 d (line 6)	3·3125	4·25	5·	5·75	6·75	7·75	9·3125
10.		Length of main bearings $\lambda = 2\,\delta$	6·625	8·5	10·	11·5	13·5	15·5	18·625
11.		Projected area, line 9 × line 10 . . square inches	22·	36·	50·	66·	91·	120·	174·
12.		Distance between bearings L, Table, p. 359	15·	18·	30·	37·	44·	51·	64·
13.		Distance between centre of bearing to centre of fly-wheel $x = \frac{L}{2} + \frac{\lambda}{2}$	10·8125	13·25	20·	24·25	28·75	33·25	41·3125
14.		$L + \frac{\lambda}{2} + \frac{\lambda_1}{2}\ (\lambda = \lambda_1)$ . . . inches	21·625	26·5	40·	48·5	57·5	66·5	82·625
15.		$\rho = \sqrt{\left\{P_1\left(1 - \frac{x}{L + \frac{\lambda}{2} + \frac{\lambda_1}{2}}\right)\right\}^2 + (167{\cdot}5\,D^2)^2}$	10,930	17,900	24,500	32,500	47,000	61,000	89,000
16.		Specific pressure per square inch $\frac{\text{line 15}}{\text{line 11}}$ lbs. per square inch	497	498	490	492	515	509	512

The length given for the bearing can be verified by the equation for pressure :—

$$\rho_1 = P_1 \left(1 - \frac{x_1}{L + \frac{\lambda}{2} + \frac{\lambda_1}{2}}\right)$$

in which :—

x_1 = the distance between the centre line of the outer bearing and centre of fly-wheel.

The effect of belt tension is so small as to be negligible for determining ρ_1.

The specific pressure per square inch $\frac{\rho_1}{\delta_1 \lambda_1}$ should be lower than 570 lbs., δ_1 being the diameter of the journal, and λ_1 the length.

Fly-wheel seat. Diameter Δ.

The sectional area of the shaft is calculated to resist bending due to the weight of the fly-wheel, the weight of the shaft itself, and belt tension, and also to resist the torsion caused by the motive impulses.

For all engines of less than 500 H.P. the effect of belt tension and weight of the crank shaft itself can be neglected, and the weight of the fly-wheel only need be taken into account.

Under these circumstances the expressions for the twisting and bending moments are the following simplified equations :—

$$\text{for bending} \quad R_b = \frac{W\,xy}{0{\cdot}1\left(L + \frac{\delta_1}{2} + \frac{\delta_2}{2}\right)\Delta^3}$$

$$\text{for twisting} \quad R = \frac{1{\cdot}6ES}{\Delta^3}$$

and, according to Bach's formula :—

$$R = 0{\cdot}35\ R_b + 0{\cdot}65\ \sqrt{R^2 + 4\ R_t^2}$$

x = distance from centre of main bearing to centre line of fly-wheel in inches.

y = distance from centre of outer bearing to centre line of fly-wheel in inches.

W = total weight of fly-wheel in lbs.

$L + \frac{\lambda_1}{2} + \frac{\lambda_2}{2}$ = the distance between the centre of the main bearing and the centre of outer bearing in inches.

CRANK SHAFT. DIAMETER OF FLY-WHEEL SEAT.

B.H.P.		15	25	35	50	75	100	150
1. Coefficient of irregularity		$\frac{1}{80}$	$\frac{1}{80}$	$\frac{1}{80}$	$\frac{1}{80}$	$\frac{1}{120}$	$\frac{1}{120}$	$\frac{1}{120}$
2. Stroke of piston	inches	18·	19.	21·	22·5	25·5	28·	32·5
3. Explosive force at 430 lbs. per square inch, line 5, Table, p. 362		21,600	35,500	48,500	64,000	92,000	118,500	170,000
4. Total weight of fly-wheel	tons	1·89	3·05	4·34	6·65	10·5	14·25	24·5
5. Distance between main and outer bearing centres $L + \frac{\lambda}{2} + \frac{\lambda_1}{2}$	inches	21·265	26·5	40·	48·5	57·5	66·5	82·625
6. Distance between centres of main bearing and fly-wheel x	inches	10·8125	13·25	20·	24·25	23·5	26·5	31·8125
7. $L - x = \gamma$	,,	10·8125	13·25	20·	24·25	33·5	40·0	50·8125
8. Diameter Δ	,,	5·625	6·875	8·	9·25	11·3125	12·3125	14·25

E = Explosive force upon the basis of 430 lbs. per square inch of piston area in lbs.
S = Stroke of piston in inches.
$\triangle$ = Diameter of fly-wheel seat in inches.
R = 500 lbs. per square inch.

Sundry Examples.—For large engines the crank shafts necessarily reach considerable dimensions. Shafts have been made weighing nearly 60 tons in the rough and about 30 tons when finished.

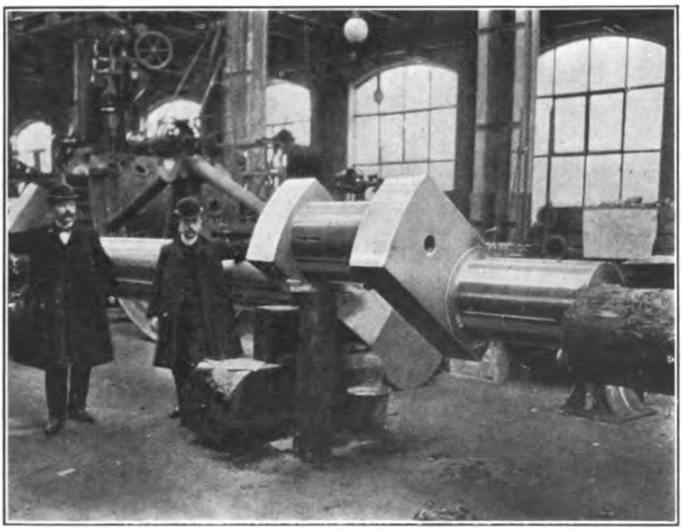


FIG. XIII.—18. Crank Shaft of 3,000 H.P. Nürnberg Engine.

Fig. XIII.—18 shows a crank shaft made by Maschinenfabrik Augsburg Nürnberg. It is 600 mm. (23·62 inches) in diameter and 31·4 feet in length. It weighs nearly 20 tons and is designed for a 3,000 H.P. twin-tandem engine. The diameter of the engine piston is 37·4 inches with a stroke of 47·24 inches.

Fig. XIII.—19 illustrates a crank shaft for a 1,200 H.P. engine with a piston 34·25 inches diameter and stroke of 43·31 inches, and running at 107 revolutions per minute. The diameter of the shaft where the fly-wheel is supported is 22·44 inches diameter, and its total weight is nearly 15 tons. These shafts are fitted to engines forming part of an installation of 3,700 H.P. in the "Burbacher Hütte" (Alsace).

The Cockerill engine shafts are built up. The two arms are of cast-steel in one piece, with the balance-weights. The crank pin| is forged steel.

The cranks hafts of the 5,400 H.P. Snow Steam Pump Co.'s engines

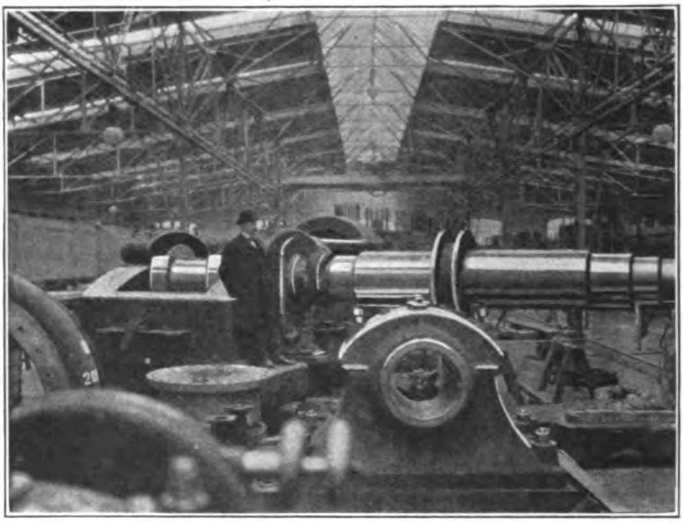


FIG. XIII.—19. Crank Shaft for 1,200 H.P. Nürnberg Engine.

weigh nearly 45 tons. The main bearings are 30 inches diameter by 56 inches long. The crank pin is 19 inches diameter by 19 inches long.

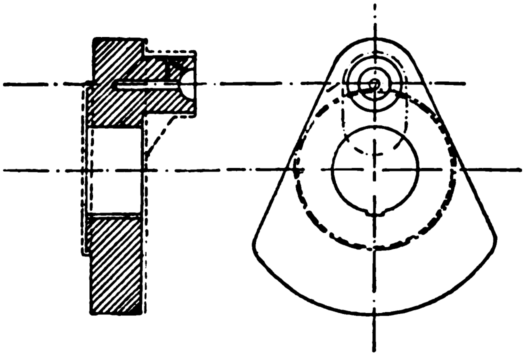


FIG. XIII.—20. Side Crank. Snow Steam Pump Works.

Side Cranks.—The side cranks generally adopted for steam engine work are not used for European gas engines, although one of the first

200 H.P. Cockerill engines was made with a side crank. But all the American firms who are building large engines have applied this arrangement. They have an advantage in not requiring such extreme care during erection, because it is much easier to place two bearings in line than three or four, as in the case of twin engines.

Fig. XIII.—20 shows the type adopted by the Snow Steam Pump Co. of Buffalo. It is a solid forging, and comprises both crank pin and balance-weight. The dotted line shows the forging before machining. The crank is shrunk on to the shaft and fitted with a key. Lubrication is by centrifugal action as generally adopted in steam engine practice.

Balance-weights.—The weights designed to partially counteract the moving masses should be very securely fixed to the crank webs by well

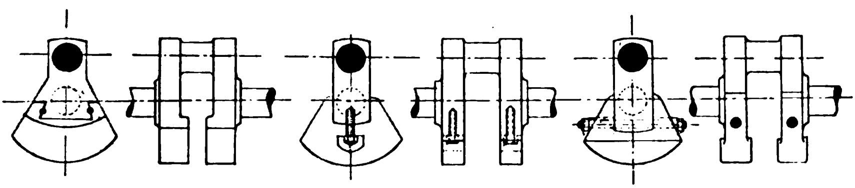


FIG. XIII.—21. FIG. XIII.—22. FIG. XIII.—23.
Method of attaching Balance-weights to Crank-webs.

fitted surfaces, and secured by bolts and keys. Their shape is generally trapezoidal.

For engines of less than 10 to 15 H.P. the balance-weights are usually made part of the fly-wheel rim.

Fig. XIII.—17 shows the method generally adopted for securing the balance-weights. Two circular clearing holes are arranged at *m* to prevent wedging during fixing.

A stud *N* is screwed into the web, and a nut and cotter-pin, placed in a slot provided for the purpose in the balance-weight, secures the latter in position. This with two double keys complete the coupling arrangement. Usually the stud serves no other purpose than to facilitate fixing, the keys securing the weights and resisting the stresses set up during work. The dimensions are calculated accordingly.

Figs. XIII.—21, 22, and 23, illustrate other methods which may be similarly applied for small engines, but these are not so good as the first (Fig. XIII.—17). The arrangement shown in Fig. XIII. —23 is frequently used for large engines, but it should be strengthened by the addition of keys.

Dimensions of Balance-weights.—The equation used is:—

$$G = 0{\cdot}7 \left(W + W^1\right) \frac{S}{2\rho}$$

in which:—

G = Weight in lbs. of the prolonged portion of balance-weight and that let into the webs.

W = The weight of crank pin plus the weight of the two upper portions of the webs, plus one-half the weight of the connecting rod in lbs.

W^1 = The total weight of the piston and its accessories (piston pin, nuts, lubricator, &c.), in lbs.

S = Stroke of piston in feet.

ρ = The distance in feet between the centre of gravity of W of the balance-weights to the centre of crank shaft.

The total length V of the balance-weight is taken as being practically equal to the length of the upper portion of the webs above the centre of the crank shaft. The obliquity of the side faces of the balance-weight is such that extension of the line would pass through the centre of the crank pin, and the arc of the lower portion is struck with a radius from the same centre.

Keys.—The dimensions of the keys are determined to resist the shearing stress E due to centrifugal force. The following equation is used:—

$$E = \frac{G}{g} \times \frac{v^2}{\rho}$$

in which:—

E = Shearing stress in lbs.

G = Weight of balance-weight in lbs.

g = Gravity = 32·2.

v = Circumferential speed of centre of gravity W.

ρ = Radius of centre of gravity W.

The results found are divided equally between the two balance-weights.

Connecting Rods.—The connecting rods are usually made of forged steel, the length between bearing centres being:—

$$\frac{Lb}{S} = 2{\cdot}5$$

or for small engines :—

$$\frac{Lb}{S} = 3$$

in which :—

Lb = Length of connecting rod between bearing centres.
S = Stroke of piston.

The section of the rod is usually either rectangular or circular; for small engines cast-steel of I section is sometimes used.

Large End or Head of Connecting Rod.—The connecting rod heads of gas engines engaging with the crank pin are necessarily of open type. For small engines the marine type of head may be used. The

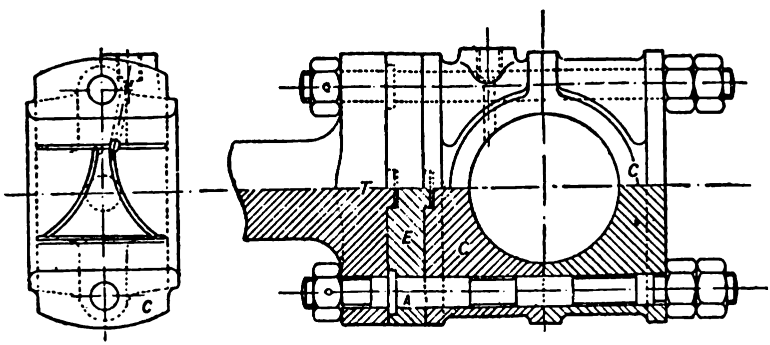


FIG. XIII.—24 and 25. Connecting Rod Head.

head terminates in a flat plate of T shape (Fig. XIII.—24 and 25), on which the solid gun-metal bearings C are fixed with one or more pieces of metal intervening to allow for adjustment of compression if necessary.

The bolts should be held and tightened in the head itself by means of a bearing surface at A and a nut as shown in Fig. XIII.—25.

For large engines, it is preferable to make the head in one forging, stamped out, being afterwards split and secured by bolts with the interposition of liners to permit precise adjustment of the bearings. The latter are made in two parts from cast-steel, lined internally with anti-friction metal. (Figs. XIII.—27, right hand drawing, XIII.—28 and 29.)

The bolts are usually two in number. The use of four bolts involves an extremely large connecting rod head, and sometimes occasions difficulties to provide sufficient room for it. Fig. XIII.—29 represents the details of the arrangement in general use. The bolts are fitted with lock-nuts, and the bolt-heads have each a stop peg, which enters into a cavity provided in the connecting rod head in the direction remote from the rod itself, and not towards the interior. This is done to avoid weakening the metal exposed to the motive impulses.

The bolts are turned with strengthened portions designed to guide

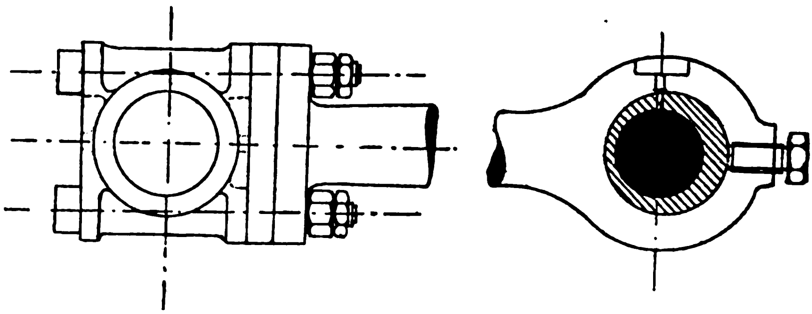


FIG. XIII.—26. Connecting Rod Ends.

them within the bolt-holes, while the remainder of the length are of smaller diameter to the threaded portion, so that breakage is more likely to occur in the body of the bolt, at the least diameter, than at a point hidden by one of the nuts, and therefore more difficult of inspection.

As shown in Fig. XIII.—29, the bolt-heads fit into notches or recesses

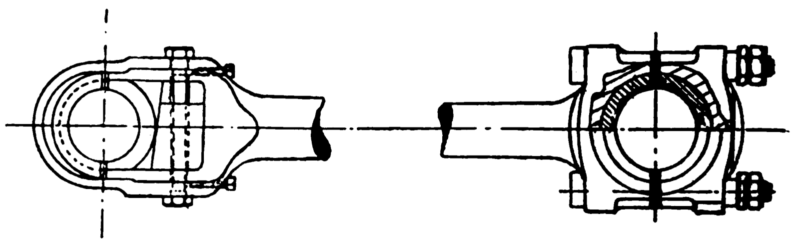


FIG. XIII.—27. Connecting Rod Ends.

in the outer portion of the brasses to prevent the latter from turning, but notwithstanding this provision, the back brass is fitted with a stop peg to prevent any side strain upon the bolts.

The cover plate of the connecting rod head does not rest directly upon the other part, but several thicknesses of liners are interposed (*C*, Fig. XIII.—29), to enable the bolts to be tightened right "home" without causing abnormal heating during work. The use of liners is preferable to bringing the metal surfaces in contact, as, with the latter,

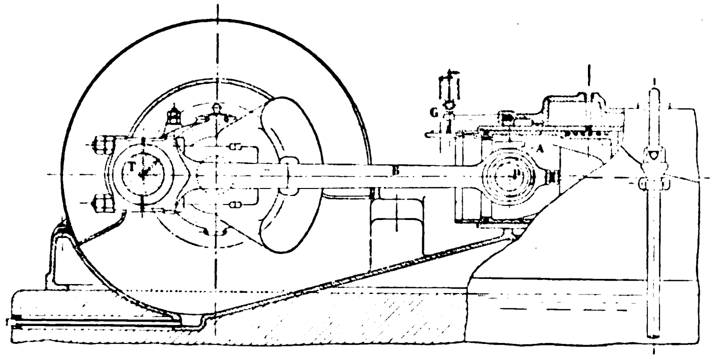


FIG. XIII.—28. Connecting Rod.

a very precise adjustment is necessary to give the bearings the correct amount of play.

Small End or "Foot" of Connecting Rod.—The small end of the connecting rod can be made from a solid forging, bored and fitted with a

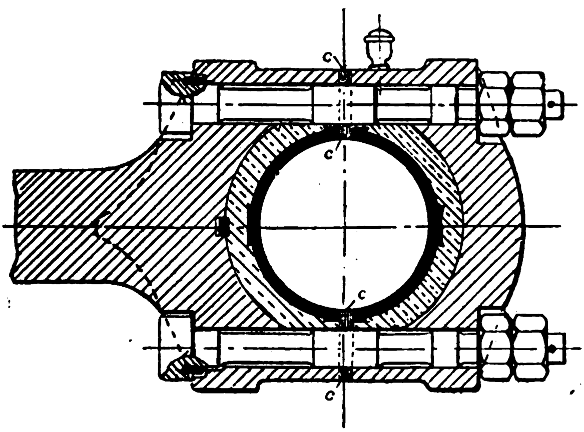


FIG. XIII.—29. Connecting Rod Head.

split bush forming a bearing adjustable by means of a screw (Figs. XIII.—26, right hand drawing, and XIII.—28). Some makers have adopted cast-iron bearings instead of gun-metal for small engines, but in the author's opinion the use of cast-iron for such a purpose is not to be commended.

At G (Fig. XIII.—28), a drip lubricator is shown, to feed the piston pin exclusively by a wiping action.

Fig. XIII.—30 shows another type of connecting rod with a solid foot, the bearings being tightened by a wedge. This wedge in single-acting engines should be placed at the back and not in the front of the bearing as in Fig. XIII.—27, left hand drawing, so as to shield it from the stresses set up at the instant of explosion.

The open type of foot, of the form adopted for the connecting rod head (Figs. XIII.—26, left, and XIII.—21, right), is not advised, numerous accidents having been caused by fracture of the bolts.

To cool the piston it is a good plan to prolong the connecting rod by a fin, so that its movement may assist the circulation of the air.

Dimensions of Connecting Rods.—The circular section of connecting

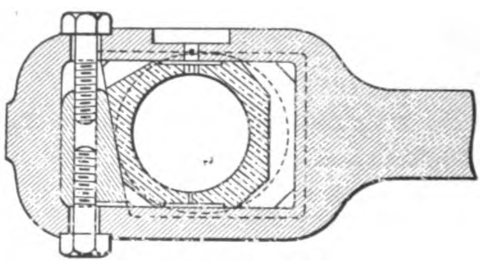


FIG. XIII.—30. Small end Connecting Rod.

rod, being less costly to make than rectangular, will be assumed for the purpose of calculation.

The diameter of the middle portion of the rod is given by the equation:—

$$d_b^4 = \frac{E\ L_b^2}{1{,}422{,}320}$$

The diameter of the part near the head $= 1{\cdot}1\ d_b$.

" " " foot $= 0{\cdot}9\ d_b$.

d_b = Diameter of middle portion in inches.

E = Total initial explosion pressure at 430 lbs. per square inch on piston area.

L_b = Length of connecting rod = 2·5 stroke of piston.

The factor 1,422,320 takes into account the elasticity of the steel at 30,000,000 lbs. per square inch and a factor of safety of 10 per cent. for a piston speed of 13·0 feet per second.

Connecting Rod Head Bolts.—Theoretically, these bolts, when well tightened up, have only to overcome the tension set up during the suction stroke, which does not exceed 70 lbs. per square inch upon the piston area, or one-sixth of the initial explosion pressure taking into account the vacuum and the inertia.

Presumably for this reason many constructors have made these bolts too weak. The author has been consulted on many occasions when more or less serious accidents have been caused by the fracture of these bolts.

To take into account some stresses that cannot be exactly determined by calculation, it is advisable to increase the amount of tensile stress by 25 per cent. and also, in the case of two bolts, to determine their diameter by the empirical formula :—

$$\delta = \sqrt{\frac{E}{32000}}$$

in which :—

δ = Minimum diameter of bolt in inches.
E = Total initial explosion pressure at 430 lbs. per square inch of piston area.

The thickness of the connecting rod head forging should be about two-thirds of the width of the crank pin. The width of the forging is about 1·8 to 2·0 times the diameter of the crank pin. The length practically equals the width.

CONNECTING ROD HEAD BOLTS.

B.H.P.	15	25	35	50	75	100	150
Total initial explosion pressure on piston area, lbs. . . .	21,600	35,500	48,500	64,000	92,000	118,500	170,000
Length of connecting rod L_b = 2·5 stroke, inches	46·25	47·5	52·5	56·25	63	70·0	81·25
Diameter at middle of connecting rod. $d_b = \sqrt[4]{\frac{E\,L_b^2}{1422320}}$ inches	2·375	2 75	3·125	3·5	4·0	4·5	5·25
Diameter of 2 head bolts $\delta = \sqrt{\frac{E}{32000}}$	·875	1·0	1·25	1·375	1·6875	1·9375	2·3125

For the solid type of small end, such as that illustrated in Fig. XIII.—30, the length should be about 1·8 to 2·0 times the diameter of the piston pin. The maximum thickness of metal left at the extremity is from 0·45 to 0·5 times the diameter of the piston pin. The width of the foot is equal to the distance between the bosses provided within the piston less the amount of projection of the bearings.

Examples.—Fig. XIII.—31 shows the connecting rod of the Foos engine provided with marine ends at the two extremities. The brasses

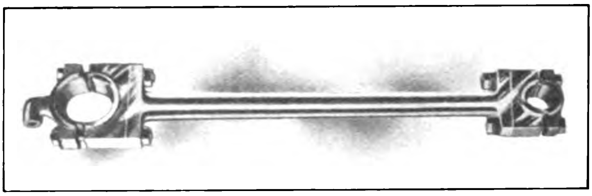


FIG. XIII.—31. Connecting Rod, Foos.

are in halves, and are brought solid one against the other when tightened up; wear is taken up by filing the contact surfaces.

The Benz connecting rod is of great length with a view to minimising, as much as possible, the strains upon the piston due to the inclination of the rod. It is made of forged steel, the head being

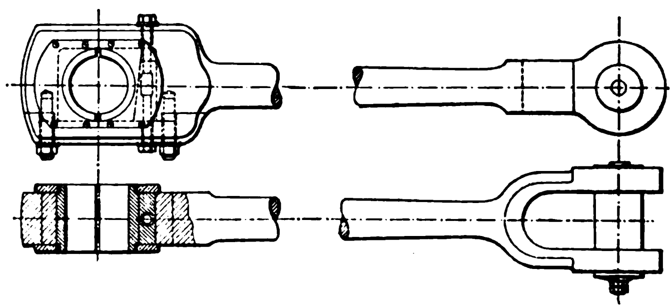


FIG. XIII.—32. Connecting Rod, Siegener.

fitted with two gun-metal bushes of large surface. The small end is solid and contains two half brasses in gun-metal adjustable for wear by a screwed wedge. For engines over 25 H.P. the bearings are lined with anti-friction metal.

The Cockerill Co., in their double-acting engines, use forked connecting rod ends for the crosshead, the head being forged or cast-steel.

The Elsassische rods are of forged steel with fork for the crosshead.

The bearings are of steel, lined with anti-friction metal, and fitted with adjustments for taking up wear.

The Siegener M. A. G. use a solid crosshead and a connecting rod, the small end of which is in the shape of a fork holding the pin. The head is formed with flush side plates fixed by means of studs (Fig. XIII.—32). Adjustment is made by a screw and wedge.

Pistons.

In single-acting engines the pistons are generally of the form shown in Fig. XIII.—33, made from very hard cast-iron and as light as possible. The back of the piston is flat or very slightly rounded. The edges are also rounded.

The piston should be fitted with at least five rings, cast from special

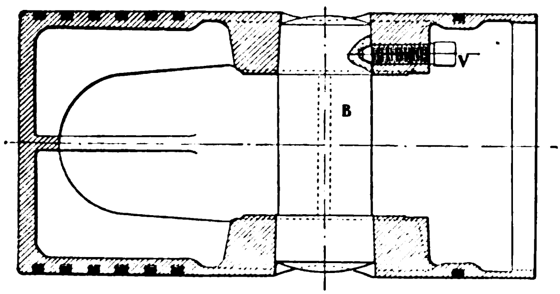


Fig. XIII.—33. Piston for single-acting Engine.

iron for engines of less than 20 H.P., and with at least six rings for large engines.

The pistons of engines less than 100 H.P. are not usually water-cooled, but it is advisable to quicken circulation of air at the back of the piston by means of a plate or fin forming an extension of the connecting rod as has been already mentioned.

Non-cooled pistons naturally expand to a greater extent than the liners which are water-cooled. To this end the back part of the piston is turned of slightly smaller diameter than the cylinder. Güldner suggests a difference of 0·2 to 0·5 per cent.

Haeder proposes 0·18 per cent. for engines with 20-inch pistons and 0·17 per cent. for a piston of 40-inch diameter.

The piston pin is made of the best quality mild steel, hardened to a depth of about ·08 inch and trued up by grinding.

The small end brasses are fitted between the two bosses inside the piston to receive the pin. The latter is usually placed towards the open end of the piston at a point from 0·5 to 0·7 of the full length. The pin is turned slightly conical, and should be fixed in such a way that it can be readily removed without risk of its becoming displaced whilst the engine is at work.

Figs. XIII.—33 and 34 show two different methods of securing the piston pins, *B*. As advocated by the author the portions of the pins within the bosses are turned taper whilst the bearing surface is parallel.

The first arrangement shown is the usual method consisting of a single screw, but it is prudent to use a lock nut in addition. The screw is placed on the side which is of smaller diameter, so as not to impede elongation by expansion. For this same reason the use of two screws, one in each boss, is not good practice.

The second arrangement (Fig. XIII.—34) shows the pin secured by a

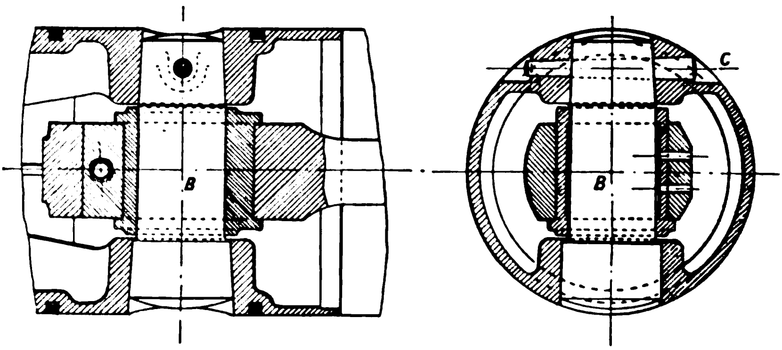


FIG. XIII.—34. Method of securing Piston Pin by Cotter.

cotter, *C*, passing through the boss and the smaller end of the pin. This method can be advantageously adapted for engines above 50 to 60 H.P.

The usual form of piston is not applicable to engines exceeding 150 to 175 H.P., that is, for pistons over 24 inches in diameter, and this is one of the reasons why single-acting engines of this size are rarely made with one cylinder only. The length of pistons cannot be less than 1·4 or 1·5 times their diameter, and when this is over 24 inches the weight becomes excessive, especially if water-cooled. To overcome the difficulty resulting from the increased weight of these long pistons, some makers have had recourse to crossheads (Fig. X.—37), but at the present time these are only applied to double-acting engines.

Lubrication.—For lubricating the piston, the orifice *A* (Fig. XIII.—35) is made in the cylinder in such a position that when the piston is at the outer dead centre this hole opens on to the front rings. The oil is conducted by a tube *B* screwed into the liner *A* and passing through the water-jacket C. To prevent leakage of water and to provide an arrangement that will not impede expansion a stuffing-box is screwed on the tube, the packing being of cotton or asbestos.

The piston pin is oiled by a sight feed adjustable lubricator G (Fig. XIII.—36), with a brush, b, or a roller, from which the oil is fed to one or two wiping blades L projecting above a small cup fitted to the piston. A tube from this dish, T, carries the oil along to a second dish in the top of the piston pin bearing.

Piston Dimensions.—Assuming an initial explosion pressure of 430 lbs. per square inch, the maximum pressure in lbs. operating vertically on the piston is about $4{\cdot}2\ D^2$, D being the piston diameter in inches.

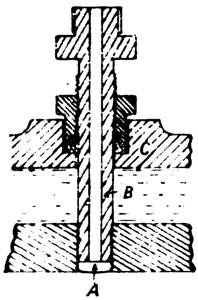


FIG. XIII.—35. Cylinder Lubrication Tube.

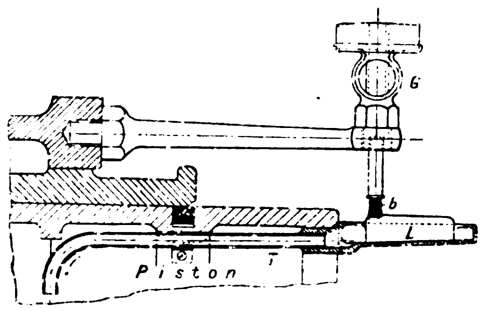


FIG. XIII.—36. Piston Pin Lubrication.

Upon the basis that the useful bearing surface of the piston corresponds to one-third of its circumference throughout its length, less 10 to 15 per cent. for large engines, and 25 to 35 per cent. for small, to allow for the space taken up by the piston rings, the specific pressure is from 18·5 to 21·5 lbs. per square inch (20 lbs. mean).

Length.—The piston's total length is given therefore by:—

$$\text{for small engines, } lp = 2{\cdot}3\ D$$
$$\text{,, large ,, } lp = 1{\cdot}65\ D.$$

The piston pin is placed in the front portion of the piston at a distance of about 40 per cent. of the full length from the front edge. It is necessary to take into account the length of the connecting rod and to make sure that it can be placed in all positions without fouling the piston.

Thickness of Metal.—The flat ends with internal ribs should have a thickness equal to 0·06 or 0·07 D. It is advisable to increase the number of ribs to facilitate cooling. The back portion of the piston where the rings are placed should be of the same thickness, while the front portion may be reduced by about 40 per cent.

Piston Pin.—The diameter can be taken empirically as 0·3 to 0·32 D, the dimension thus obtained being checked to determine its resistance to bending:—

$$d^3 = 0{\cdot}000887\ E\ l$$

in which:—

d = Piston pin diameter in inches.
l = Length of bearing surface of pin in inches.
E = Initial explosion pressure at 430 lbs. per square inch on piston area in lbs.

This formula assumes a practical coefficient of resistance for steel of 11,400 lbs. per square inch.

The length of bearing surface of pin l, in inches, is determined by the formula relating to the specific pressure:—

$$l = \frac{E}{1800\ d}.$$

The taper given to the two extremities of the pin corresponds to an angle of about 2·5 to 3°.

Bosses.—The projection of the two bosses are fixed by the length of the piston pin bearing. To provide against lateral play the distance between should not be more than ·002 inch wider than the brasses. The external diameter of the bosses is twice that of the pin.

PISTON: LENGTH, THICKNESS. PISTON PIN: DIAMETER AND LENGTH.

	EFF. H.P.	15	25	35	50	75	100	150
1	Diameter, piston, inches	8·0	10·25	12·0	13·75	16·5	18·75	22·5
2	Length of piston . .	18·0	21·5	25·0	28·0	31·0	33·0	38·0
3	Ratio $\frac{\text{line 2}}{\text{line 1}}$ . . .	2·25	2·1	2·08	2·04	1·88	1·76	1·69
4	Thickness of metal at back of piston . .	·5625	·6875	·8125	·9375	1·0625	1·1875	1·375
5	Diameter, piston pin .	2·625	3·25	3·75	4·3125	5·0	5·75	6·75
6	Total explosion pressure at 430 lbs. per square inch on piston . .	21,600	35,500	48,500	64,000	92,000	118,500	170,000
7	Length of piston pin bearing . . .	4·625	6·125	7·25	8·25	10·25	11·5	14·0

Examples.—In the Benz engines the piston length is twice the diameter. From six to eight soft cast-iron rings are fitted and a circular rib extends round the middle where the pin is placed.

For their single-acting 175 H.P. engine, the Maschinenfabrik Augsburg Nürnberg obtained cooling by water circulation by means of a tube following the reciprocating movement (Fig. X.—37). The piston is connected to a crosshead. In the non-cooled pistons the pin is held at its two extremities in the forked end of the connecting rod, and oscillates within a central bearing carried on a cast-iron strut arranged

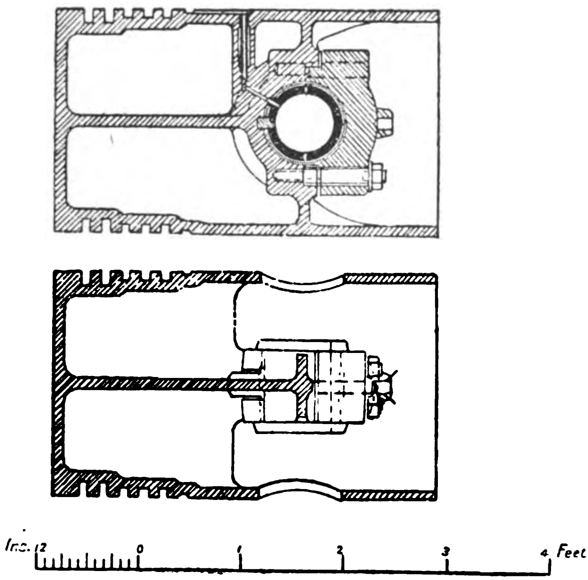


FIG. XIII.—37. Nürnberg Piston.

in the centre of the piston (Fig. XIII.—37). This arrangement was designed to protect the piston pin from the influence of high piston wall temperature, but appears to have been abandoned on account of its cost.

Fig. XIII.—38 is a horizontal section of a 100 H.P. engine made by Kynoch, Ltd., of Birmingham. The piston is fitted with a double end for circulation of cooling water. The inlet and outlet of the water is arranged by means of two sliding tubes and stuffing-boxes. The pistons of the single-acting, tandem Winterthur engines are similarly cooled by circulation of water (Fig. V.—10).

Piston Rings.—The method adopted by Tangyes Ltd., of Birmingham, and the Winterthur Co., of placing a piston ring in the front part of the pistons of single-acting engines is undoubtedly good, as this additional ring retains the larger portion of the oil, which has a

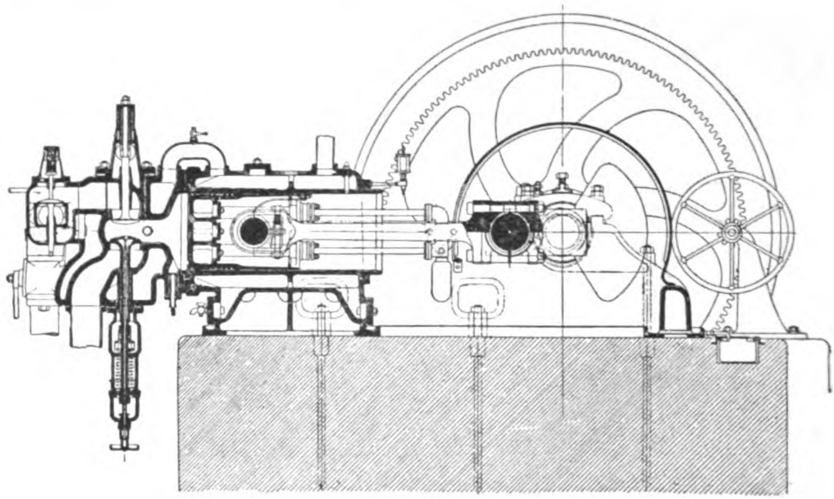


FIG. XIII.—38. Water-cooled Piston, Kynoch.

tendency to become wasted, and ensures more uniform wear throughout the length of the liner. See Figs. XIII.—33 and 36.

In the author's opinion, the rings turned out of centre are not suitable, because they do not give an equal pressure, as do those of a uniform thickness. Moreover the grooves in the piston being of the

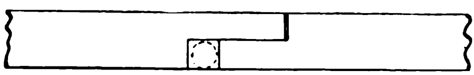


FIG. XIII.—39. Piston Ring Joint.

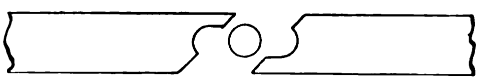


FIG. XIII.—40. Piston Ring Joint.

same depth, the eccentric rings, being thinner at the point, leave a space which favours the accumulation of deposit and possible lack of tightness.

According to general practice the thickness of the rings should be ·03 of the cylinder diameter, while the width should be from 1·5 to 2·0 times the thickness in inverse ratio to the number used.

The original external diameter of a turned ring before being cut, fitted to the piston and entered into the cylinder is

$$De = D + \frac{e}{3{\cdot}14} + 0{\cdot}2,$$

all measurements being in inches.

The amount to be cut out for the gap is 0·08 D.

To determine whether a worn ring still has sufficient elasticity or

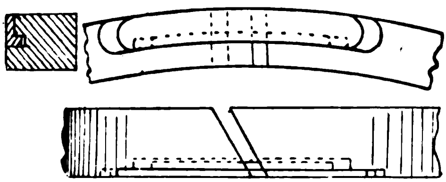


FIG. XIII.—41. Piston Ring Joint.

good work, the length and gap is measured and compared with the last formula. If it does not agree, the inside face of the ring is lightly and uniformly hammered.

The rings should be made of good quality, soft, homogeneous cast-iron, turned to the original external diameter De before being cut for fitting. When the gap has been cut to make the ring of the exact diameter of the cylinder bore, the ring is closed and put back in the lathe so as to obtain uniformity of bearing pressure throughout the periphery of the cylinder.

The gap is cut in one or other of the methods indicated in

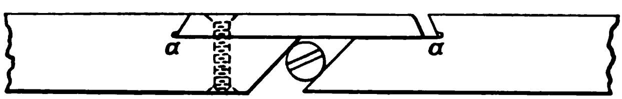


FIG. XIII.—42. Piston Ring Joint.

Figs. XIII.—39 and 40, being suitable for engines of less than 40 to 50 H.P. The first method is the better of the two. Above this power the rings are sufficiently large to permit the use of steel cover joints either free—as in Fig. XIII.—41—or fixed by a screw—as in Fig. XIII.—42—at the point of juncture. The flush headed screws should pass through all the width of the rings, and the ends should be lightly riveted over. It is advisable to slightly round the apex of the angles at a (Fig. XIII.—42), to prevent fissures in the metal. The length of the cover joint should be about one-third of the circumference of the ring.

To prevent the rings rotating in the grooves stop pegs are fitted. Usually these are made by a screw, the end of which projects into the interior of the piston where it is bent over (Fig. XIII.—43), so that the screw may not become slack and score the cylinder. This stop peg should be placed in the lowest part of the piston for lubrication and be arranged in such a manner that the joints do not come in the same direct line, so as to avoid all gas leakage.

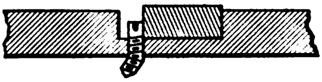


FIG. XIII.—43. Piston Ring Stop-peg.

When the rings are made as explained, it is needless to proceed to fit them to the cylinder itself. The latter method is still followed in France by some makers of gas engines and of steam engines, but has been abandoned elsewhere.

Pistons for Double-Acting Engines.—These consist of the following essential portions :—

1. Piston.
2. Rod.
3. Crosshead.
4. Water-cooling system.

The *piston* is made in cast-iron or steel with a water-jacket, fixed against a collar formed on the rod, either by bolts or a large nut, and is fitted with at least four rings.

Many constructors make their pistons as short as possible, and support them by means of the piston rod. Others, on the contrary, are of the opinion that it is to better advantage to permit the piston to rest upon the cylinder liner, and, accordingly, give it a great length to diminish the pressure of contact per unit of area and to allow the rings full freedom, which, it is considered, is one of the essential conditions for thorough gastightness. The latter makers think that the suspension of the piston by the rod causes vibrations and sets up play between the piston and cylinder in a vertical direction, which may reach ·16 inch, whilst this play never exceeds ·08 inch if the pistons themselves bear upon the cylinder.

Numerous difficulties have been encountered in the construction of pistons, and many designs originally applied when large gas engines were first set to work have had to be abandoned. Internal ribs in these pistons have nearly always given bad results; they are never long in use without cracking.

The block of metal forming the piston should never be used as a coupling to connect the front and back rods. The rod should be a single piece or formed of two portions rigidly bolted together. The method of coupling represented in Fig. XIII.—44 is defective, and fissures are soon produced between the stud holes and the central opening. When the rod is fixed in the manner shown in Fig. XIII.—45 by means of a collar and nut, the connection can be made tight at the outset, but after a certain time it becomes set fast to the thread, from which it is impossible to remove it. Very often it causes cracks to

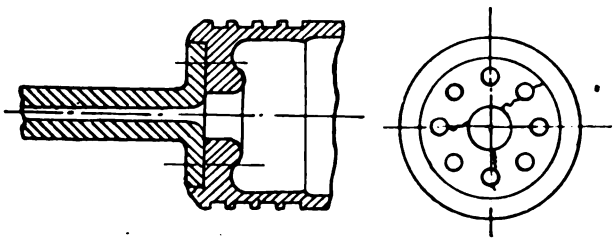


FIG. XIII.—44. Piston and Rod Coupling (defective).

develop. However, as far as the tightening and slackening of the nut is concerned, the difficulty can be got over by making the nut and threaded portion of metals of different hardness.

The *rod* is made of nickel steel or cast tungsten steel, having a resistance limit of 85,000 per square inch with 18 per cent. elongation.

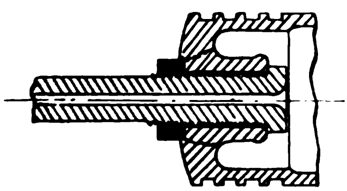


FIG. XIII.—45. Piston and Rod Coupling.

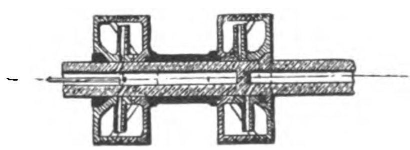


FIG. XIII.—46. Piston and Rod Coupling, Duisburger.

It is hollow throughout its length and fitted with a tube serving as flow and return for the water circulation.

The *crosshead* sometimes takes the form of the marine type steam engine crosshead, so that the piston may be dismantled without interfering with the crosshead. It is made of steel and fitted with one or two hardened or case hardened pins.

The *water circulation system* should be designed to ensure a constant renewal of water at a pressure determined by the speed of the moving masses. The firms of Koerting and Cockerill, acting entirely independently from one another, made the first water-cooled pistons.

Examples.—The arrangements adopted for the construction of pistons are very diverse; the principal forms only will be mentioned.

In the Dingler engines water-cooled trunk pistons are used, the

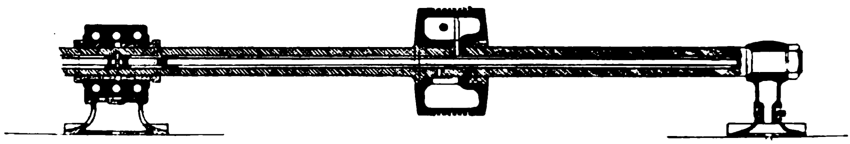
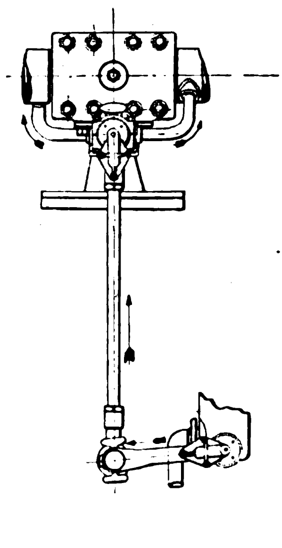


FIG. XIII.—47. Piston and Rod Coupling. Ehrhardt.

portion which serves as a slide being shod with alloy. At the front of the piston an additional ring is fitted to equalise the wear of the cylinder liner caused by the other rings. The piston rods and the



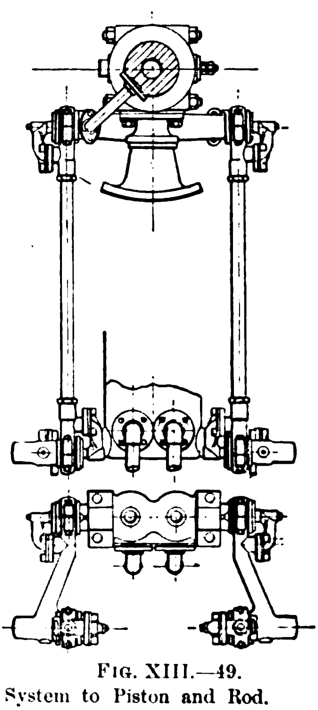


FIG. XIII.—48. FIG. XIII.—49.
Koerting Water Cooling System to Piston and Rod.

crosshead bolt are of nickel steel. The pistons are fixed in grooves in the rod by means of straps in two pieces. The tightness is ensured by steel spring rings. The pistons are very accessible and their tightness can be easily verified during work.

The Duisburger Co., formerly Bechem & Keetman, use a piston made in two parts, as shown in Fig. XIII.—46, and designed to uncover the exhaust ports at each end of the power stroke to permit complete expansion and cooling of the burnt gas.

Ehrhardt & Sehmer use pistons of hollow cast-iron and made as light as possible (Fig. XIII.—47). The piston rod is of large size and guided at its two extremities. It is hollow and, like the piston, is water-cooled. The crosshead has only one lower frictional surface, and is kept in place by two projecting side pieces, an arrangement which makes the front stuffing-boxes particularly accessible.

In the Haniel & Lueg engines, the piston is fixed to the rod by means of a cone and taper thread with nuts. The cooling water flows to the front piston and returns by the rod. The piston is fitted with

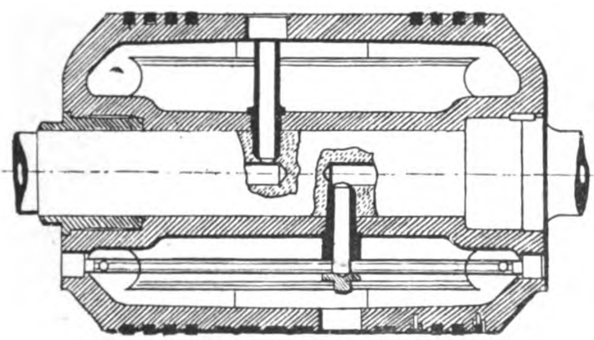


FIG. XIII.—50. Koerting water-cooled Piston.

five or six rings and is supported by the crossheads. The front crosshead is made in Siemens-Martin steel, and the cast-iron shoe is shod with white metal. Two side sliding surfaces keep the crosshead in its path.

In tandem engines the supply of water to the pistons and rods is brought to the middle of the two cylinders, the outlet being at the rear portion of the piston rod.

As shown in Figs. XIII.—48 and 49, the Koerting engine has the same arrangement with oscillating articulated tubes. In recently constructed Koerting engines long pistons cast in one piece are used. The collar on the rod is made either at one end or the other (Fig. XIII.—50) or at both extremities. In all engines of large power the piston rod is fitted with a tail guide.

The piston body of the Siegener Co.'s engine is cast with strengthening ribs at either end, allowing thin metal to be used and ensuring perfect water circulation. The piston is fixed against a solid collar on the rod by means of a nut; it has a long bearing surface and supports its own weight in the cylinder. The frictional surfaces are shod with white metal.

In Nürnberg engines the water used for cooling of the piston is introduced through a tube within the rod, from the rear crosshead by means of jointed tubes which follow the reciprocating movement

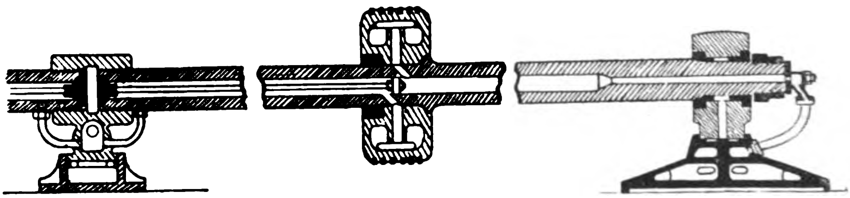


FIG. XIII.—51. Nürnberg Piston, Rod, and Crossheads.

(Fig. XII.—42). After passing through the piston, the water escapes from the rod at the front crosshead. Fig. XIII.—51 shows a new general arrangement of Nürnberg pistons and crossheads.

The Reichenbach double-acting engine pistons (Fig. XIII.—52) are cast in one piece with a water-jacket. They are fitted with four rings, and are fixed on the rod by two right and left hand nuts. This

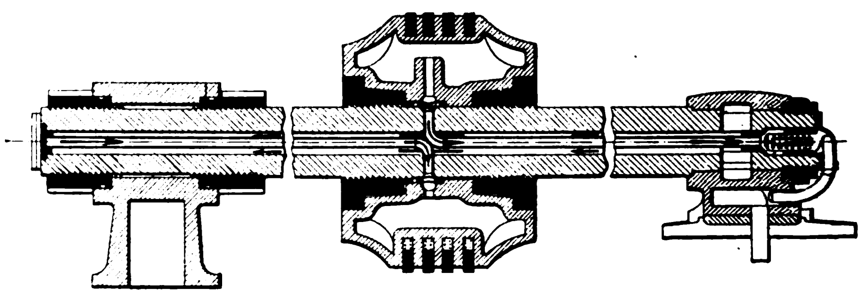


FIG. XIII.—52. Reichenbach Piston, Rod, and Crossheads.

arrangement allows the piston to be dismantled from one side or the other. The rod is hollow and contains two internal tubes joined at the centre of the piston with a two-way connection which ensure circulation of water in the direction of the arrows.

The Otto-Deutz pistons are suspended by the rods or the crossheads without resting their weight in the cylinder. The inlet and outlet of cooling water is by the piston rods and jointed pipes, connected to a pump. The inner passage in the rods contains a tube throughout its whole length forming an internal compartment. The water is sent to the piston at the portion where its diameter is greatest and, passing by the channel concentric to the tube, passes away by the same tube.

The Cockerill Co. have recently patented an arrangement of a water-cooled piston represented in Fig. XIII.—53. The piston proper is composed of two cast-iron water-jacketed portions which may be

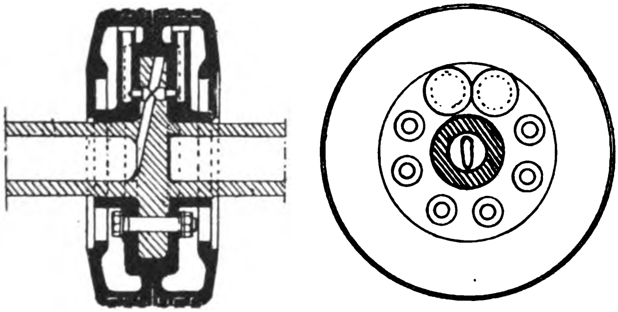


FIG. XIII.—53. Cockerill Piston and Rod Coupling.

coupled and tightened to a collar forming part of the hollow piston rod. By this arrangement the coupling joint is not in contact with the hot gases. The tightening bolts are let in, the water-jacket and the openings corresponding to these bolts are closed by separate cover plates. The rods support the entire weight of the piston, and are

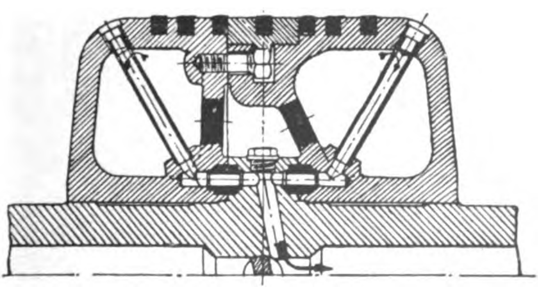


FIG. XIII.—54. Schüchtermann Piston and Rod Coupling.

curved slightly upwards so that they become correctly in line after erection under the influence of the weight of the pistons and water. This arrangement was first brought out by the Maschinenfabrik Augsburg Nürnberg.

Schüchtermann & Kremer similarly make use of pistons in two pieces connected by nuts placed within the water chamber (Fig. XIII.—54).

Figs. XIII.—55 and 56 represent two arrangements by the Elsassische

Co. In the first the rods after being drilled are reforged, annealed, and finished in the shop, so as to reduce the transverse dimensions

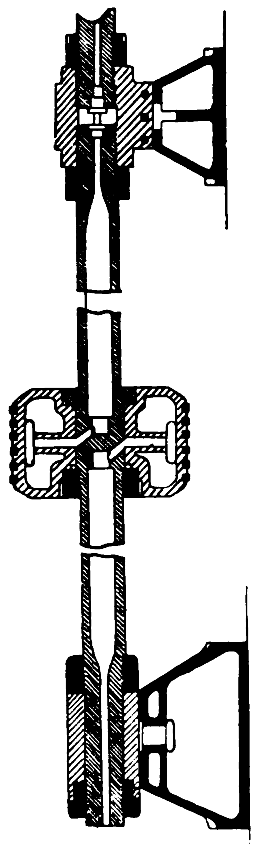


Fig. XIII.—55. Elsassische Piston, Rod, and Crossheads.

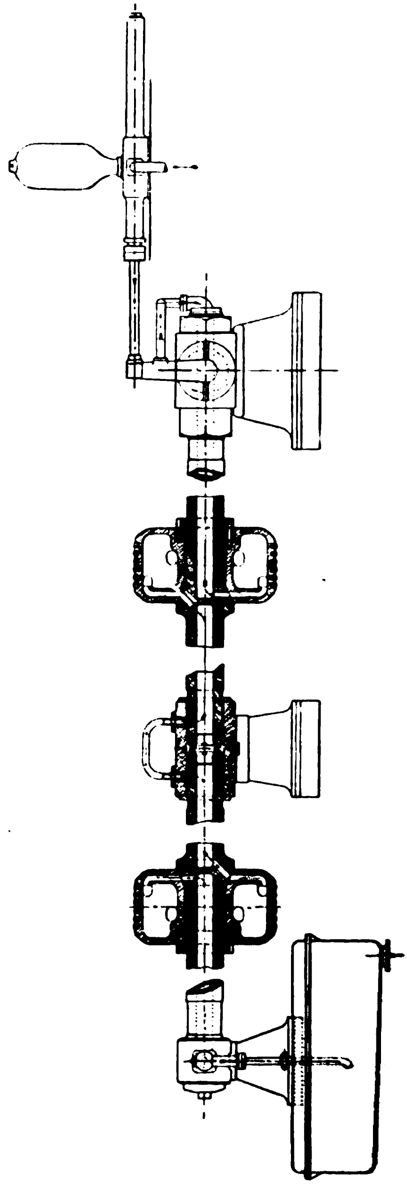


Fig. XIII.—56. Elsassische Piston, Rod, and Crossheads.

of the couplings. In Fig. XIII.—56 the internal pipe is of uniform area, and the drawing shows who the water circulates in the piston and rod. The water inlet is by means of a sliding tube and stuffing-box.

The pistons of the 5,400 H.P. engines made by the Snow Steam Pump Co. (Fig. XIII.—57) are in halves connected by bolts, the joint surfaces being ground. The nickel steel rod is in two parts—an external tube and a central rod. The latter is swelled out at the middle of the piston and acts as a support to the latter, forming a bulkhead to cause normal circulation of the water. At each outer end

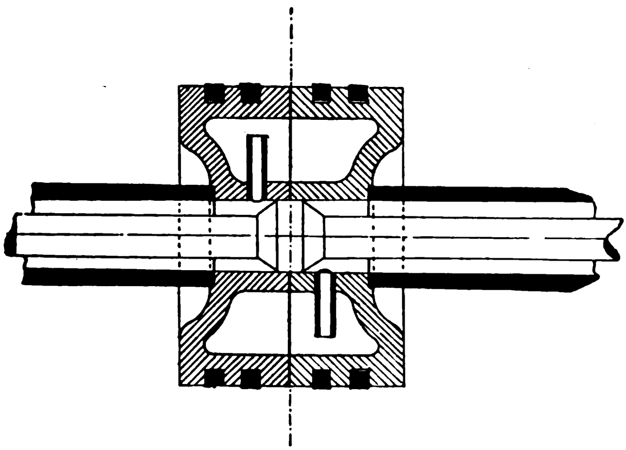


FIG. XIII.—57. Snow Steam Pump Works Piston and Rod.

a large nut is used to tighten the external tube firmly against the piston, forming a very rigid and perfectly tight connection.

VALVES.

All valves should be very accessible and arranged so as to be easily and quickly dismantled for grinding and cleaning purposes. Excepting only the gas valves disposed similarly to the practice adopted by Koerting's in their four-cycle engine, all valves are mechanically operated.

The valve spindles should be provided with lubricating tubes, and for engines over 50 H.P. these tubes should be fitted with sight-feed oil cups to prevent sticking of the valves and abnormal wear. The exhaust valve spindle particularly should be well cooled by the water circulation around its guide.

The spindle and valve head are usually of one piece of mild steel except when more than 8 or 10 inches diameter, the heads then being cast-iron with the spindle screwed or riveted to the disc.

The entire surface of the valve should be perfectly smooth so as to present no rough portion favourable to the accumulation of dirt or soot. In the upper part of the valve head, a projection or hole is provided for the reception of a grinding tool. Such projections are liable to become overheated and red-hot, while the holes may become filled with dirt which is likely to get incandescent. The better plan is to make a simple groove in the centre of the disc to receive the grinding tool.

The conical bearing surfaces at the valves should be at an angle of from 30 to 45° with the flat surface of the head.

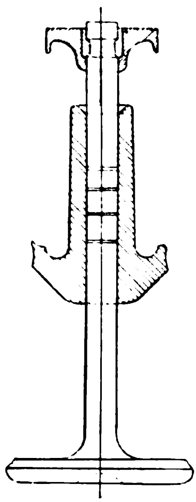


FIG. XIII.—58. Inlet Valve.

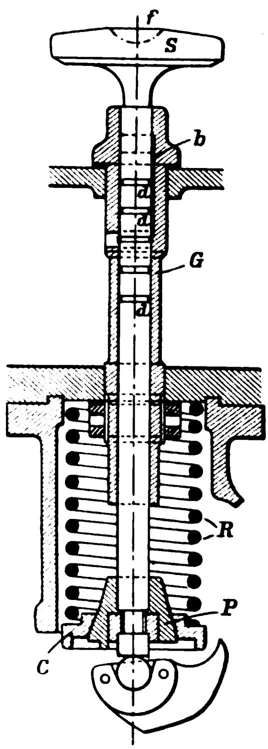


FIG. XIII.—59. Exhaust Valve.

The width of the seat should be from 0·2 to 0·4 inches. Flat-seated valves for the mixture inlet, and for exhaust, are practically abandoned.

In large single-acting engines, besides water-cooling the exhaust valves by means of the spindle, the covers over the openings giving access to the combustion chamber were separately cooled in the earliest designs. The majority of engine builders now place the inlet valve vertically in the same axis as the exhaust in order to reduce the chamber to the necessary size to give high compression. With such an arrangement the exhaust valve is partially cooled by the entry of the mixture during suction.

In modern engines the valves are placed vertically, horizontal valves being used only for the mixture inlet in some small engines. Horizontal valves are difficult to keep tight, and the least play due to wear causes wedging. They can only be used when the valves are very light with long guides and short strokes.

The inlet valves can be thin and light, as they are not subjected to heating but cooled by the entering mixture at each suction stroke (Fig. XIII.—58).

Fig. XIII.—59 represents an exhaust valve. In order to ensure uniform lubrication of the spindle, circular grooves, *a a*, are provided with a communicating longitudinal groove, *b*. The spindle is guided in a sleeve *G* supported and fixed between the walls of the water-jacket. At the end of the spindle is a disc *C* to receive the concentric coiled spring *R*. Contact between the lever and spindle is made by means of a roller in large engines or by an adjusting screw for small sizes.

Water-cooled exhaust valves are used in single-acting engines above 100 to 125 H.P., and as a general rule for double-acting engines. The discs of these valves are hollow and made of either cast-iron or cast-steel. Several examples are referred to later on.

For non-water-cooled exhaust valves, experience has shown that a very massive valve head (*S*, Fig. XIII.—59) is less likely to reach excessive temperatures, and for this reason the portion above the seat is thickened up cylindrically, terminating in a flat surface with angles slightly rounded. Non-water-cooled valves with such large discs are not recommended because the central portion cannot be cooled sufficiently well by mere contact with the seat. Some makers of large engines, however, have been able to secure good results with large uncooled exhaust valves.

The form given to the gas valve varies according to the system of governing as discussed in Chapter XI. When the gas valve is annular, fitted concentrically to the inlet valve spindle and moving with it, it can be furnished advantageously with a flat seat giving a tight contact even though the axis of one may be warped.

Valve Dimensions.—The dimensions of the valves are calculated as a function of the volume of the fluid drawn in or expelled to give the best efficiency, that is to say, to cause the minimum of negative pressure during suction and of back pressure during exhaust.

The instants at which the valves begin to open and completely close correspond practically to the inner and outer dead centre positions of the piston, except in those engines in which the closure of the inlet valve is timed by trip gear. Thus the areas of the valve openings should be in accord with the linear speed of the piston at the same moments or, in other words, the valve areas must be proportionate to the velocity of the fluid passing through.

Main Inlet Valves.—With regard to the determination of the area of opening of the inlet valve it is sufficient to take into account the mean linear piston speed and to give the entering fluid a certain mean velocity.

Some makers adopt a mean velocity of more than 160 feet per second, whilst others do not exceed 80 feet. In the author's opinion these two figures constitute the extreme limits, and he has deduced from a number of experiments that the speeds of from 100 to 130 feet give very good results. On the one hand they better conform to the necessity of giving such dimensions that their accommodation within a restricted combustion chamber is not a difficult matter, while, on the other hand, they ensure intimate mixture of the fluids without causing any excessive loss in the charges.

The area of opening is equal to the product of the circumference of the conduit under the valve and the maximum valve lift. Theoretically, for equal speeds on both sides of the fully lifted valve, the lift should be one-fourth of the effective diameter. Practically, on account of the space taken up by the spindles and guides, it is sufficient to provide a lift of about one-fifth the diameter. Some makers provide even less than this in cases where, for peculiar features of the governing, it is necessary to prevent a quick and noisy return of the valve on its seat.

Assuming a mean speed of 115 feet per second and a valve lift equal to $\frac{d_v}{5}$ the diameter of the inlet valve is given by the equation

$$d_v = \sqrt{\frac{S\ N\ D^2}{33300}}$$

in which:—

d_v = diameter of valve in inches.
S = stroke of piston in inches.
N = revolutions per minute.
D = diameter of piston in inches.

Gas Valves.—In determining the area of the gas valves it is necessary to take into account the nature of the gas and its ratio to the bulk of the mixture for the desired amount of compression.

The practical proportions of air and gas are as follows :—

Air to	illuminating gas	= 8 to 9 to 1.
,,	coke oven gas	= 6 to 7·5 to 1.
,,	producer gas	= 1·3 to 1·5 to 1.
,,	blast furnace gas	= 0·9 to 1·0 to 1.

The average speed through the valve should be the same as for the inlet valve, viz., 100 to 130 feet per second. Taking, as an example, producer gas in the proportion of 1·4 air to 1·0 gas, the formula giving the diameter of the gas valve for a mean speed of 115 feet per second will be :—

$$d_g = \sqrt{\frac{S\ N\ D^2}{57000}}.$$

Exhaust Valves.—The diameter given to the exhaust valve is usually the same as that of the inlet valve.

Air, Gas, and Exhaust Passages.—The gas and air passages are calculated on the basis of 40 to 60 feet per second. The area of the exhaust passage should be equal to 1·5 to 1·8 times that of the valve so as to bring about a rapid fall of pressure favourable to cooling the products of combustion and their silent exhaust.

VALVE DIAMETERS (inches)

B.H.P.	15	25	35	50	75	100	150
Diameter of piston . .	8	10·25	12	13·75	16·5	18·75	22·5
Stroke ,,	18·5	19	21	22·5	25·5	28	32·5
Revolutions per minute .	230	220	210	200	190	180	160
Diameter inlet and exhaust valves	2·875	3·625	4·375	5	6·3125	7·3125	8·875
Diameter gas valve for producer gas . . .	2·1875	2·75	3·3125	3·875	4·8125	5·5625	6·8125

Examples.—Mr. A. Vennell Coster, of Messrs. Crossley Brothers, Ltd., was one of the first to think of balancing the exhaust valves to relieve the operating mechanism from the enormous loads due to the gas pressure towards the end of the power stroke. His first arrangement consisted of an ordinary valve made hollow for water circulation, and with its seat towards the interior of the combustion chamber, whilst a swollen part in the shape of a piston formed a chamber in which a

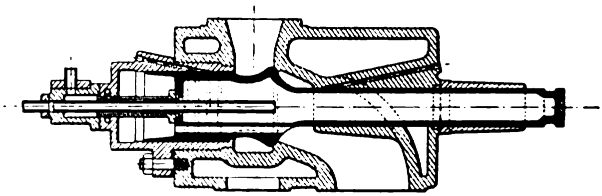


FIG. XIII.—60. Crossley double-seated Exhaust Valve.

passage opened to give access to the exhaust gas. The admission of this gas under the piston was assured by means of a small valve

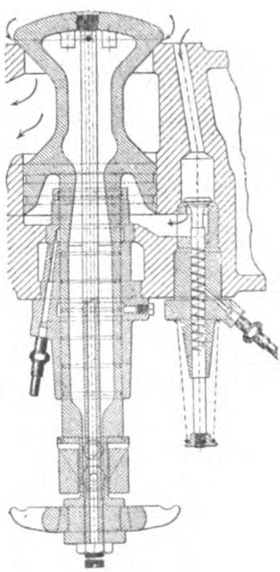


FIG. XIII.—61. Crossley double-seated Exhaust Valve.

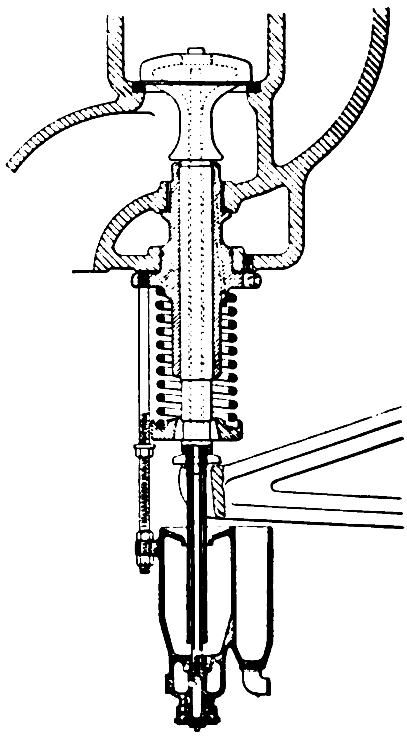


FIG. XIII.—62. Koerting water-cooled Exhaust Valve.

operated by the principal lever actuating the main valve, but opening in advance of the latter.

At a later period Messrs. Crossley Brothers, Ltd., adopted an arrangement consisting of a valve with one annular portion forming a piston, and balanced without needing an auxiliary valve (Fig. XIII.—60). The piston of this exhaust valve has an area of about 75 per cent. that of the valve itself, so that the power required to lift it is reduced by three-fourths. Fig. XIII.—61 represents the latest form of valve adopted by them, which comprises also an auxiliary valve.

Fig. XIII.—62 is a Koerting water-cooled valve. The water enters by a central fixed tube and flows into a kind of bucket fitted with an overflow arranged below the valve. Fig. XIII.—63 gives an end view of the operating lever.

Fig. XIII.—64 shows an exhaust valve of the "Union" engine.

Fig. XIII.—65 is a reproduction of a photograph of three exhaust valves and seats belonging to a 2,000 H.P. Nürnberg engine, with

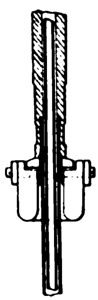


FIG. XIII.—63. End view of Exhaust Valve Lever, Koerting.

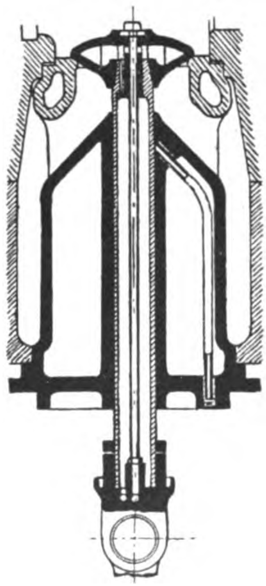


FIG. XIII.—64. Union water-cooled Exhaust Valve.

piston 40 inches diameter and 51 inches stroke, running at 90 revolutions per minute. It has been previously mentioned that the Maschinenfabrik Augsburg Nürnberg in their single-acting engines have slightly lowered the exhaust valve seat below the base of the cylinder so as to naturally evacuate the excess of oil and dust brought into the engine.

In the large Otto-Deutz engines the exhaust valve is provided with water circulation by means of its spindle in the manner adopted for the piston. Water circulation is also arranged in the box containing the exhaust valve seat. In still larger engines, such as the 2,000 H.P. size, with two cylinders tandem, having pistons 43·3 inches diameter by 51·2 inches stroke, the exhaust valve seat measures 15 inches. During the exhaust a pressure of about 2 tons would be required to lift it. It is, therefore, constructed with a bye-pass which establishes the same pressure above and below the valve head, and, being thus balanced, only has a resistance equal to the tension of the springs.

Exhaust valves always constitute one of the delicate portions of the mechanism of internal combustion engines, and they cannot be too well cared for or inspected. It is for such a reason that the firm of Krupp, of Essen, have made an innovation consisting of placing the exhaust valve in the top of the cylinder (Fig. XIII.—66).

Practice and experiment has shown that this combination has not

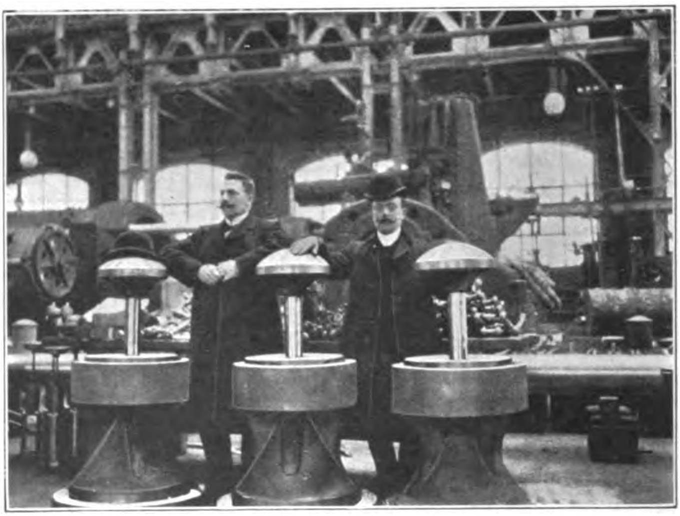


FIG. XIII.—65. Three Exhaust Valves for 2,000 H.P. Nürnberg Engine.

given the satisfaction expected by the makers, the advantage gained in the facility of access not being enough to compensate for the inconveniences that, in the author's opinion, may be summed up in the three following points:—

1. Increase of temperature of the explosion chambers and inlet valves, which heat the gas during admission.

2. The complex form of the explosion chamber, which is unfavourable to the propagation of flame in the explosive mixture.

3. The absence of the natural means of getting rid of oil and dirt, as when the exhaust valves are placed in the base of the cylinders.

The unfavourable influence of the temperature of the enclosure upon the efficiency of explosion engines has already been mentioned. The

Krupp engine as well as the Sargent engine may be criticised in this connection. The transverse section through the valve gear of the Sargent engine (Fig. XIII.—67) shows that a common valve serves both for admission and expulsion. A detailed drawing of this valve is given on p. 137.

Usually the valve chamber is arranged upon the cylinder itself and not at the end, a form of construction originally due to the Otto-Deutz Co. It has been adopted by the Cockerill Co. and their licensees, who

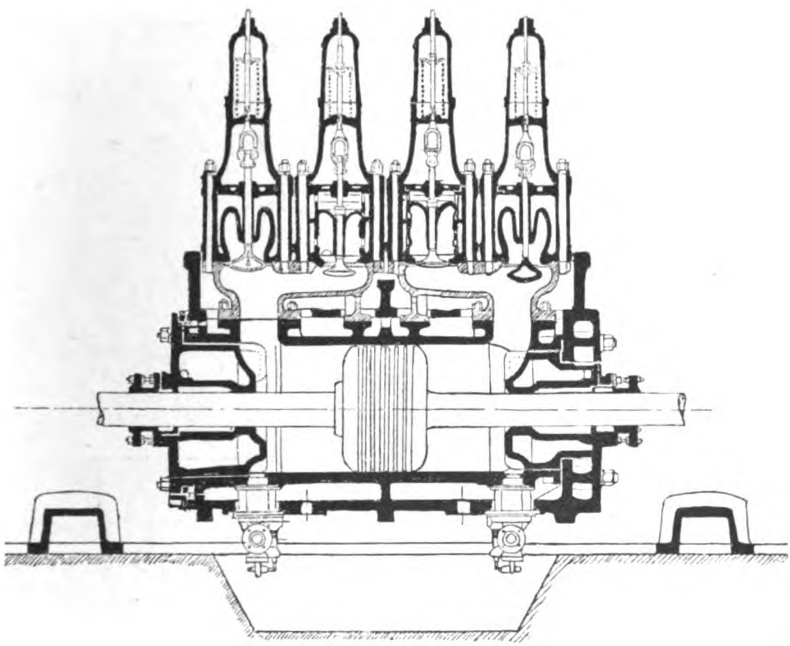


FIG. XIII.—66. Krupp double-acting Engine with Exhaust Valve at top of Cylinder.

have abandoned the original type of double-acting engines in which the inlet and outlet valves were underneath in a separate cylinder head.

The first 200 H.P. single-acting blast furnace gas engine built by Cockerill from the designs of Delamarre-Deboutteville had its inlet valve at the back of the breech end and the exhaust valve below.

The majority of makers put the gas and air valves or slide concentrically above the inlet valve. The Otto-Deutz large engines have the gas valve at the side of the admission valve, which allows the details to be very accessible for the necessary cleaning.

The Nürnberg gas valves, as well as the Ehrhardt & Sehmer gas and air valves, are arranged at the side of the inlet valve in the longitudinal axis of the engine.

Fig. XIII.—68 shows how the different portions of the Winterthur inlet valve are dismantled.

Half-Speed Shafts.—The arrangement most widely adopted for actuating the valves consists of a single valve gear shaft placed at the

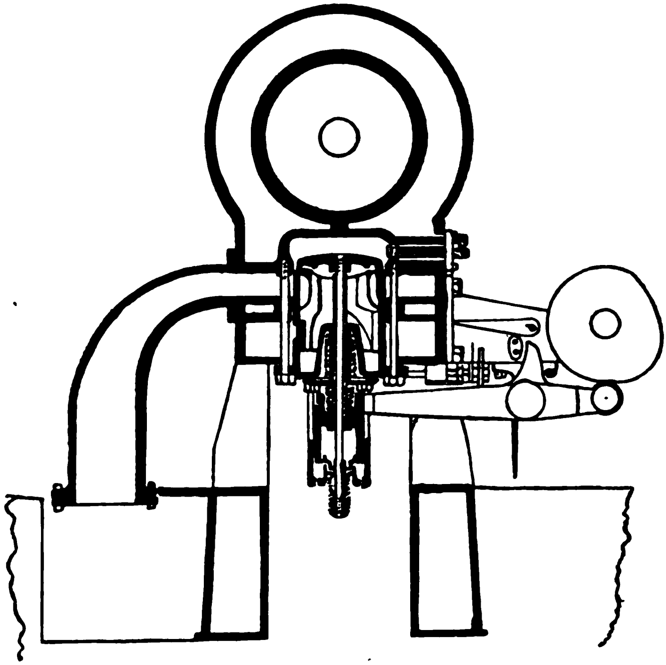


FIG. XIII.—67. Sargent combined Inlet and Exhaust Valve.

side of the engine frame and extending throughout its length, receiving its motion from the engine shaft.

Upon this steel shaft eccentrics or cams are keyed to operate the valve levers, as well as the gear wheels for driving the governor.

Some builders, however, use a shaft parallel to the axis of the engine shaft, the valves being actuated by levers and rods. Fig. XIII.—69 shows the arrangement of this type of valve gear as made by the Dudbridge Ironworks, Ltd., for some of their small engines. The Foos Co. and L. Gardner & Sons (Fig. XIII.—74), have also made engines presenting similar features.

Diameter.—The diameter of the cam shaft is given by the equation :—

$$d_1 = 0{\cdot}15 \text{ to } 0{\cdot}2\ D.$$
$$D = \text{diameter of piston.}$$

The higher coefficient is applied to the smaller diameters, and for small engines $1\frac{1}{4}$ to $1\frac{3}{8}$ inches is the minimum diameter. Calculations based upon the strains set up to effect the opening of the valves give dimensions much too weak, and involve vibrations due to the shock produced by the action of the cams. For single-acting engines above 75 to 100 H.P. it is a good plan to fit a third intermediate bearing for preventing vibrations.

Gear Wheels.—To drive the half-speed shaft bevel wheels or skew gears are employed, the revolutions of which are in the ratio of 2 to 1.

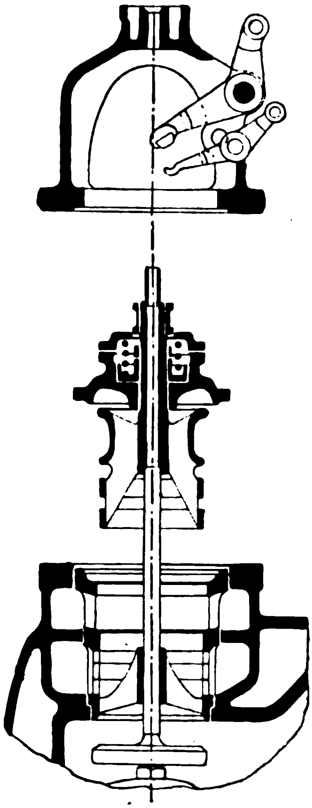


FIG. XIII.—68. Method of dismantling Inlet Valve, Winterthur.

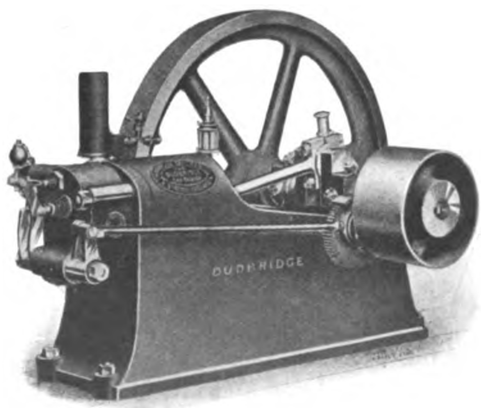


FIG. XIII.—69. Engine with Valve Gear operated by Eccentric, Dudbridge.

For bevel gears the diameters and number of teeth are in inverse proportion to the number of revolutions. For skew gears the number of revolutions are independent of the diameters, permitting either the use of equal diameters, for instance, or of one twice the diameter of the other. In the first case the space required is much reduced, but the wear of the teeth is more rapid.

Spherical wheels, upon which some makers have specialised, are to be recommended, presenting as they do the advantage of gearing several teeth simultaneously, and thus offering a large contact surface which diminishes the wear.

The gear wheels are usually made of hard cast-iron or cast-steel. To obtain silent working the driving wheel is sometimes made of bronze. In all cases the teeth should be cut with extreme precision.

The diameter of the gear wheel bosses should be twice the diameter of the bore. From this rule it is easy to determine the relative

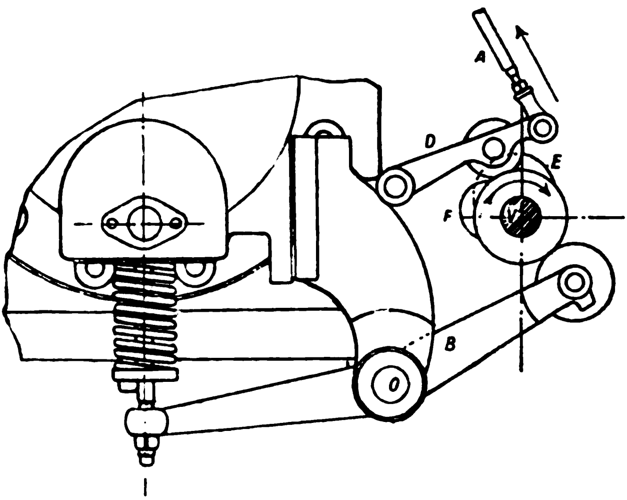


FIG. XIII.—70. Valve Gear with separate Cams.

position of the crank and half-speed shafts according to the ratio given for the diameter of the two wheels.

Details of bearings and brackets have already been given on p. 297.

Levers and Cams.—When the valve lift is obtained by the simple movement of levers and cams, a powerful spring keeps the valve on its seat. As far as the inlet valve levers are concerned their duty is confined to overcoming the spring resistance. For the exhaust valves it is necessary to overcome the force of the pressure of the burnt gases before their expulsion from the cylinder. This is equal to about 40 to 45 lbs. per square inch.

Fig. XIII.—70 shows the arrangement generally given to the valve levers when each is operated by a separate cam.

Fig. XIII.—71 represents an arrangement of a single cam *C* fixed to the half-speed shaft V. The inlet valve is actuated by a rod A held in position by the connecting rod D. The exhaust valve receives its movement by the interposition of a beam F oscillating round the point O, and against which the rod B works, kept in place by the connecting rod E.

In large engines the cam and levers are called upon to transmit a force of from about 2 to 2½ tons.

When the engines are governed by the variable admission of a quantity of mixture, causing a partial vacuum in the cylinder when

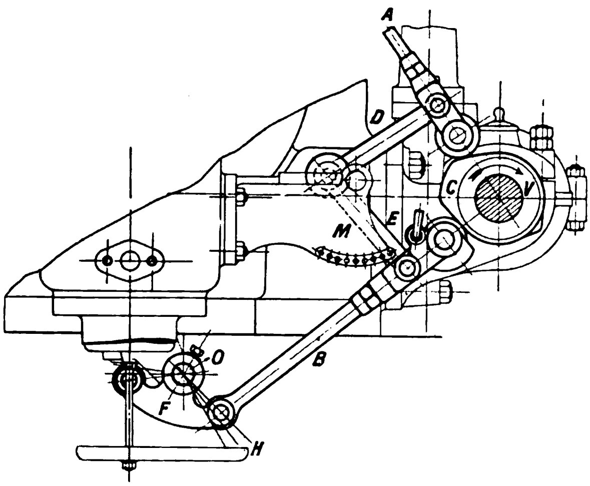


FIG. XIII.—71. Valve Gear with single Cam.

not fully loaded, the inlet and exhaust valves are likely to be opened, unless furnished with powerful springs.

The levers are made of either cast-iron, or cast or forged steel, as light as possible to reduce the effects of inertia, but very strong and rigid.

The rollers should be of either chilled cast-iron or best quality hardened steel. The case hardening should extend to a depth of at least ·08 inch, the surfaces being afterwards trued by grinding.

If d_1 = diameter of the half-speed shaft, the rollers are given a diameter of 1·8 or 2·0 d_1; their width being 0·45 or 0·5 d_1 for the inlet valve lever and 0·55 or 0·6 d_1 for the exhaust.

It should be remembered that rollers of large diameter revolve more easily and produce smoother working, and are less likely to become set fast. When possible it is advisable to let the rollers work in a bath of oil contained in a dish forming part of the lever.

Spindles and Pins.—To withstand the effect of the sudden movements of the cams and levers the spindles and pins are of hardened steel trued by grinding. Their dimensions should be such that the specific pressure per square inch does not exceed 600 to 650 lbs. per square inch.

The figure taken for the resistance of the inlet valve spring is 42·5 lbs. per square inch of the valve. For the exhaust valve the total resistance is taken as 85 lbs. per square inch, half to overcome the tension of the spring, and the other half to resist the pressure of the exhaust gases.

The force acting upon the fulcrum pin is the resultant of those acting at each of the extremities. In the case of levers with equal arms, the pressure upon the pin is double that acting on the valve.

The effective bearing length of the fulcrum pin is usually twice its diameter.

SIDE SHAFT, ROLLERS, AND PINS.

B.H.P.	15	25	35	50	75	100	150
Piston diameter . . inches	8·0	10·25	12·0	13·75	16·5	18·75	22·5
Diameter, side shaft . „	1·5	1·875	2·0	2·25	2·5	3·0	3·375
Diameter, lever rollers . „	3·0	3·75	4·0	4·5	5·0	6·0	6·75
Width, inlet valve roller . .	·75	1·0	1·0	1·125	1·25	1·5	1·75
Width, exhaust valve roller . .	·875	1·125	1·1875	1·25	1·5	1·75	2·0
Area of inlet and exhaust valves, square inches	6·5	10·32	15·0	19·6	31·0	42·0	62·0
Pressure on inlet valve . . lbs.	276	440	637	833	1,320	1,785	2,640
Pressure on exhaust valve . „	552	880	1,274	1,666	2,640	3,570	5,280
Diameter of fulcrum pins							
Inlet valve lever . inches	·6875	·875	1·0	1·25	1·5	1·75	2·0
Exhaust valve lever . „	1·0	1·25	1·5	1·75	2·0	2·5	3·0

Cams.—To facilitate the projection of cams it is advantageous to give them a large diameter, but, when the peripheral speed of the body or boss of the cam reaches about 3 feet per second, the movements of the valve levers become noisy in ascending and descending the profile.

From the dimensions indicated for engines from 15 to 150 H.P., if the body of the cam is given a diameter equal to twice that of the side shaft, the peripheral speed will be within 2·5 feet per second.

The cams are made in extra hard cast-iron or hardened steel in one piece with the boss; the projecting portion should be set off tangentially. The hardened steel cams should be trued up after hardening.

To avoid abnormal resistances it is necessary that for each opening position of the valves, the speed of the entering fluid should be practically proportioned to the linear speed of the piston. It should be remembered that as the speed of rotation of the crank shaft is double that of the cam shaft, the angles of lead and lap measured on the cams are doubled with regard to the crank angles.

The curve of the accelerations of piston speed is plotted graphically, and from that the corresponding valve lifts for each valve are deduced, observing the rule just given. These valve lifts are set out to scale on the working drawing, attention being paid to the ratios adopted for the lever arms, and the proper heights being marked off along the

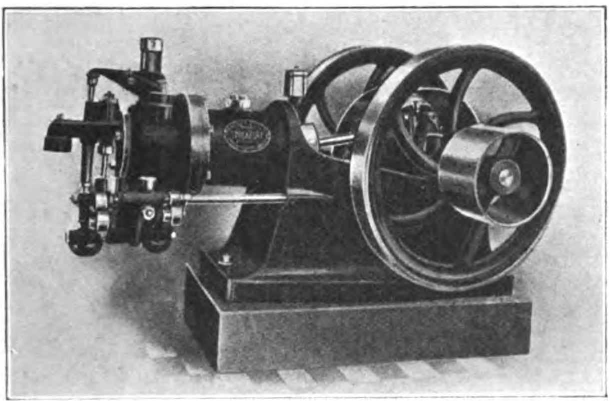


FIG. XIII.—72. Valve Gear of small Hornsby-Stockport Engine.

radii of the cam projection between the extreme limits determined by the angles of lead and lap.

Examples.—Figs. XIII.—72 and 73 show the system of valve gear applied in the small and large types of Hornsby-Stockport gas engines.

In the small sizes the side shaft passes through the frame and is driven from the engine shaft by skew cut gear wheels placed inside the frame next to the bearings (Fig. XIII.—72). In the larger sizes the side shaft is put outside the frame in the usual manner.

In the majority of cases the inlet valve gear is placed at the back of the engine, the operating lever being arranged in the centre line of the cylinder. This is a peculiarity of the Hornsby-Stockport engine, and is carried out in this way to leave the side of the combustion chamber free for the ignition device. However, in several of their larger sizes the admission valve gear is placed at the side.

Fig. XIII.—74 shows the valve gear as applied to a $1\frac{1}{2}$ H.P. Gardner oil engine.

The transverse section (Fig. XIII.—75) refers to a new type of engine built by the Olds Gas Power Co., of Lansing, and shows the inlet and exhaust valve gear actuated by a single cam. On the admission-valve spindle, and within the same box, is a gas valve and a cylindrical air slide valve fitted with multiple ports.

From Fig. IV.—2, p. 33, the very compact arrangement of the

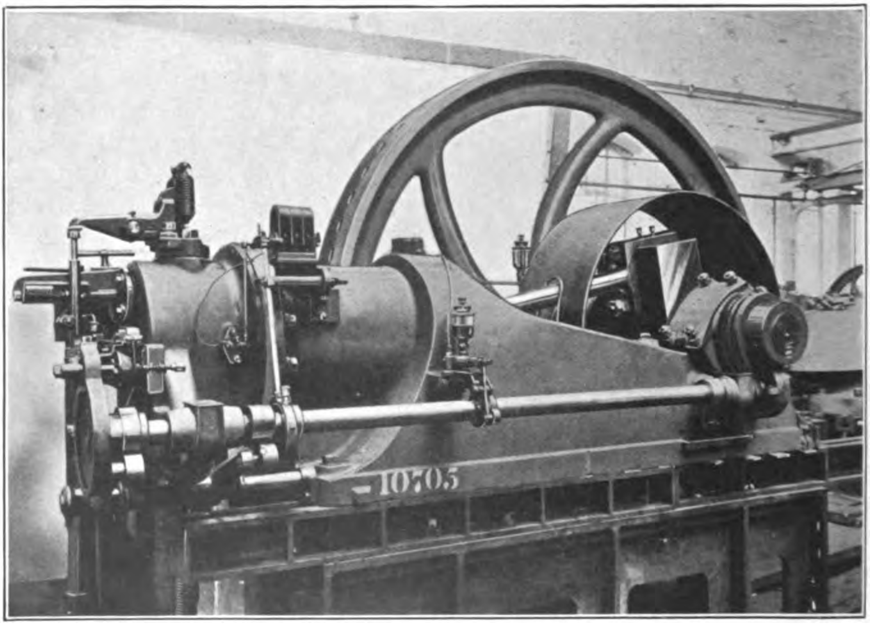


FIG. XIII.—73. Valve Gear of medium power Hornsby-Stockport Engines.

valve gear will be remarked. At the extremity of the shaft is a cam for the mechanically operated compressed air starting valve; another cam operates the magneto and sparking plug.

Fig. XIII.—76 shows a section of the mechanism employed by the Premier Gas Engine Co., Ltd. The gas valve G is annular, with double seat and double opening. It is connected to a circular slide valve, which partially closes the air inlet when the gas valve opens. The passage for the gas increases with the lift of the gas valve, whilst the air passage diminishes. When the gas valve is closed the air inlets are fully open, thus permitting free passage for the scavenging air.

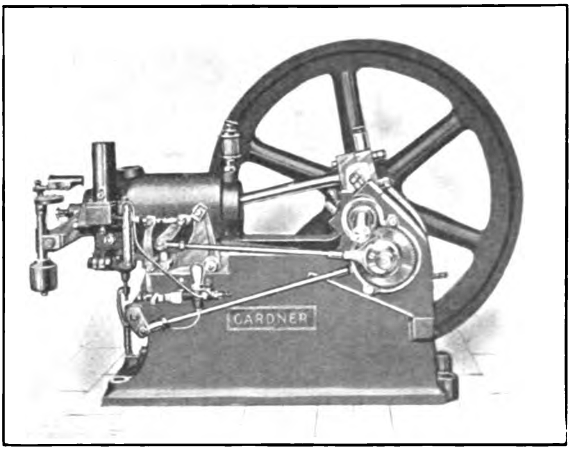


FIG. XIII.—74. Valve Gear of 1½ H.P. Gardner Oil Engine.

The inlet valve is operated by the lever *L*, and the gas valve by the lever *K*, through a knife blade *Y*, connected to the governor by the

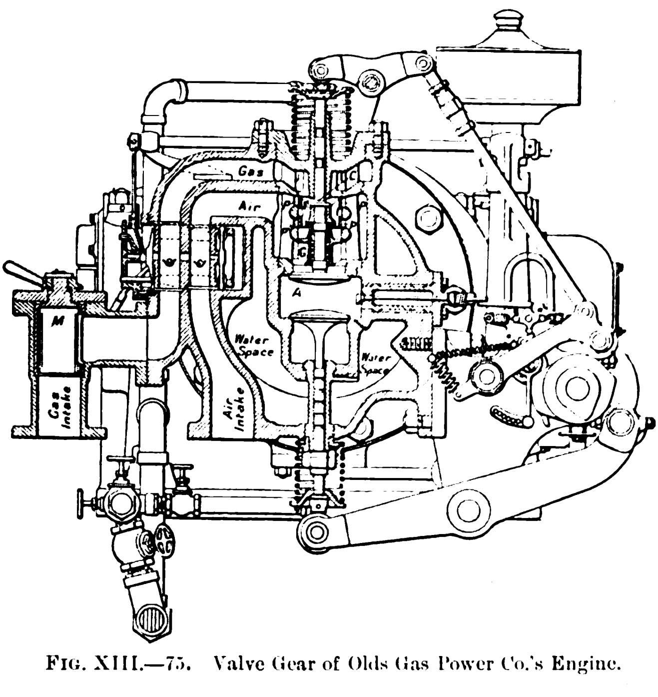


FIG. XIII.—75. Valve Gear of Olds Gas Power Co.'s Engine.

rod *R* and lever *T*. When the governor rises the blade *Y* is pushed towards the left, and becomes engaged at a greater distance from the support of the arm X, to which it is attached, and as a consequence the opening of the gas valve is reduced, and *vice versâ*.

Dingler similarly actuates the admission and exhaust valve by means of a single cam.

In large double-acting tandem engines, the side shaft, being very long, is sometimes made in several pieces. The coupling flanges

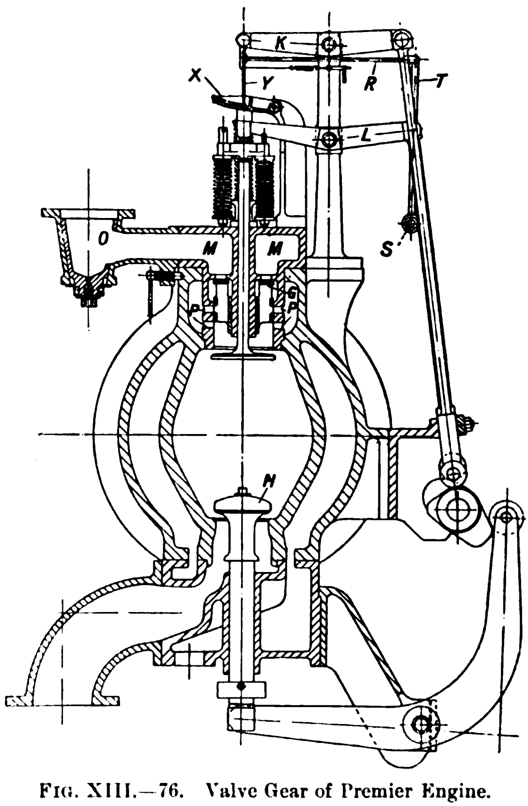


FIG. XIII.—76. Valve Gear of Premier Engine.

should be designed to respond to the differences in expansion that may be set up. The bearings supporting this shaft should be placed in such a way that no vibration is caused by the movements of the eccentrics or valve levers.

Eccentrics, Rolling-paths.—The valve gear of large gas engines include eccentrics and levers with rolling-paths similar to those used for steam engines of the Sulzer type, in order to ensure smooth and quieter working with the increased size and weight of the working parts.

The eccentrics should be of large size, as only about 15 per cent. of their stroke can be utilised. This certainly may be increased by means of the rolling contacts, which permit also adjustments to be made during erection, and useful correction of the acceleration of the speed of the actual opening and closing of the valves.

To avoid the valves opening irregularly they are positively operated both for opening and closing, as, for example, in the ingenious Cockerill device, Fig. XI.—51, where the exhaust valve rod is provided with a double set of rollers. A short spring with only a slight movement is fitted merely to compensate for the clearance and ensure smooth and silent operation.

The Krupp valve gear includes an arrangement upon the same principle, where a cam moves in a kind of collar.

Both these designs preserve the advantages of the cam, the profile of which better accords with the movement that should be given to the valves than that obtainable with an eccentric.

In the new 2,000 H.P. Otto-Deutz engine (Figs. XI.—35 and 36) the manner in which the movement of the cam keeps the valve closed whilst a spring opens the valve when the cam is out of contact has already been described.

The Maschinenfabrik Augsburg Nürnberg give preference to eccentrics which with the levers with rolling contacts ensure very smooth working. The Elsassische Co., of Mulhouse, and Soest, also employ eccentrics.

In the Elsassische engine the exhaust valve opens by means of rolling levers, and is closed by a special lever connected to the eccentric rod being kept to its seat during the suction stroke, a spring forming an elastic connection.

The Siegener M. A. G. use eccentrics and rolling levers. The valves are fitted with air pistons to avoid the need of powerful springs, which increase the weight of the valve gear.

Several arrangements of trip gears have been described in Chapter XI.

Governors.—For engines below 15 H.P. controlled by hit-and-miss the governor may be of the inertia type, and above this power a centrifugal governor should be used. The variation in the number of revolutions between "full" and "no" load should not exceed 3 per cent. for "hit-and-miss" governing, and 4 per cent. for engines governing by variable admission.

Belt-driven governors are not advised, and skew-cut gear wheels are preferable to bevel gears, a certain amount of play being left for axial expansion.

Horizontal spindle governors are but little used. They are exposed to greater wear than vertical spindle governors, unless made extremely light.

The Hallesche Maschinenfabrik place the governor upon the cam shaft itself.

In the case of variable admission,

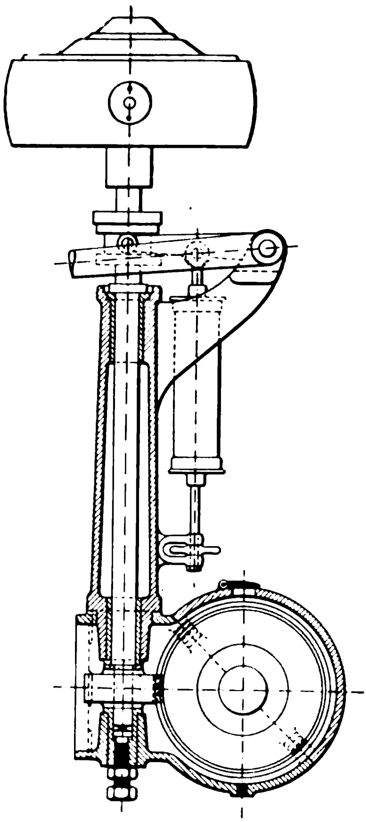


FIG. XIII.—77. Vertical Spindle Governor and Dash-pot.

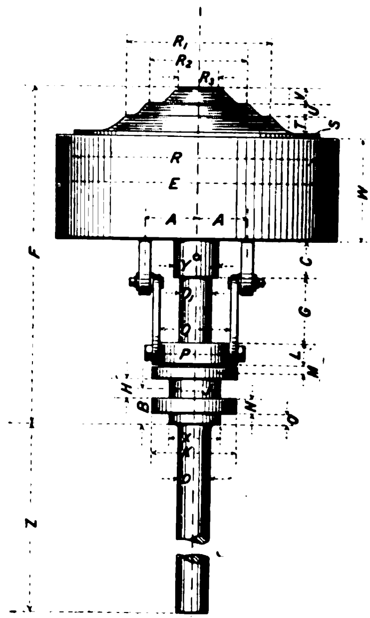


FIG. XIII.—78. Hartung Governor.

steadiness of governing is obtained by means of an oil piston or dash-pot, with facilities for regulating the resistance easily without having to stop the engine. A spring, the tension of which can be adjusted, also enables adjustments to be made during work to vary the engine speed with the limits of 5 to 10 per cent. of the normal.

Fig. XIII.—77 shows the usual arrangement of the governor bracket. The vertical spindle rests upon a steel pin or disc, the position of which is adjusted by a screw and lock nut.

Several firms, particularly in Germany, have specialised in the construction of governors, and amongst others:—Hartung & Co., Düsseldorf; Wiede Theodore, Chemnitz; and Zabel & Co., Quedlinbury am Harz.

Hartung makes ball governors and spring governors of the Hartung, Temple, and Stumpf types.

In the Hartung governor (Fig. XIII.—78) the joints are arranged so as to be shielded from friction caused by the action of the horizontally displaced weights and of the springs which counteract them. In such conditions the sensitiveness of the apparatus is very great, the coefficient of unsensitiveness being only 0·3 per cent.

The energy of the governor increases in proportion to the number of revolutions. Tables I., II., and III. (pp. 410 and 411) reproduce the details communicated by the makers relative to the Hartung type of governors.

The Temple governors have similarly balanced joints and are of two different types. One with constant energy, whatever the number of revolutions of the apparatus, and the other with a great speed variation that is in the proportion of 100 per cent. The relative details are given in the Tables IV. and V., pp. 411 and 412.

Stumpf governors, Table VI., p. 412, have a large speed variation, but the static springs are so designed as to maintain the maximum number of revolutions without closing up.

Springs.

Springs are employed in gas engines to bring the valves back to their seats after having been lifted by the positive action of the valve gear. Two distinct kinds of springs are used, either those acting under compression and fitted concentrically with the valve spindles, or those under tension, placed either in an extension, or by the side of the valve spindle, one end engaging with the latter while the other is anchored to a fixed point.

The first arrangement (Fig. XIII.—79) enables the springs to be accommodated in the valve boxes, where they are partially let into the engine casing, improving the general appearance of the engine but rendering their inspection rather more difficult. The springs under tension (Fig. XIII.—80), involves a more cumbrous arrangement, but are more accessible, and, consequently, may be inspected with greater ease. External springs can be very easily renewed when necessary.

TABLE I.—HARTUNG GOVERNORS.

Pattern Number.	89	90	91	92	93	94	95	96	97	98	99	100	101	102
1. Revolutions per minute	400	380	340	310	240	240	210	200	190	180	165	160	140	130
2. Average energy in lbs.	55	82·5	123·75	178·75	244·75	330	412·5	522·5	577·5	660	880	1,320	1,540	2,200
3. Stroke in inches	·4724	·5906	·7874	·9843	1·1811	1·1811	1·5748	1·9685	2·3622	2·7559	3·1496	3·5433	3·9370	4·3308
4. Power in inch-lbs. Energy × Stroke	26	48·7	97·4	175·7	290	390	650	1,027	1,365	1,820	2,775	4,675	6,500	9,500
5. Average variation in energy per 2 per cent. speed variation (lbs.)	2·2	3·3	4·95	7·15	9·9	13·2	16·5	20·9	23·15	26·4	35·2	52·8	61·6	88·0
6. Power variation per 2 per cent. speed variation.—Energy variation × Stroke (inch-lbs.)	1·04	1·95	3·9	7·04	11·7	20·75	26	41·2	54·6	73	111	187·5	260	382

TABLE II.—HARTUNG GOVERNORS.

Pattern Number.	89*a*	90*a*	91*a*	92*a*	93*a*	94*a*	95*a*	96*a*	97*a*	98*a*	99*a*	100*a*	101*a*	102*a*
1. Revolutions per minute	600	580	550	500	460	420	380	350	310	280	250	220	190	170
2. Average energy in lbs.	165	220	275	357·5	440	550	660	796	935	1,100	1,375	1,650	1,925	2,475
3. Stroke in inches	·47	·59	·78	·98	1·18	1·18	1·57	1·96	2·36	2·75	3·15	3·54	3·93	4·33
4. Power in inch-lbs. Energy × Stroke	77·5	129	215	350	529	650	1,035	1,560	2,200	3,025	4,300	5,800	7,560	10,700
5. Average variation in energy per 2 per cent. speed variation (lbs.)	6·6	8·8	11	14·3	17·6	22	26·4	31·90	37·4	44	55	66	75·6	99
6. Power variation per 2 per cent. speed variation.—Energy variation × Stroke (inch-lbs.)	3·1	5·2	8·6	14·1	20·8	26	41·6	62·8	88·3	121	173	234	304	430

TABLE III.—HARTUNG GOVERNORS.

Pattern Number.	91*b*	92*b*	93*b*	94*b*	95*b*	96	97*b*	98*b*	99*b*	100*b*	101*b*	102*b*
1. Revolution per minute	550	500	460	420	380	350	310	280	250	220	190	170
2. Average energy in lbs.	192·5	264	330	385	440	577	660	741	1,020	1,375	1,870	2,470
3. Stroke in inches	1·18	1·49	1·77	1·77	2·36	2·95	3·54	4·13	4·72	5·31	5·90	6·49
4. Power in inch-lbs. Energy × Stroke	227	393	585	682·5	1,040	1,700	2,334	3,060	4,820	7,300	11,000	16,000
5. Average variation in energy per 2 per cent. speed variation	7·7	9·9	13·2	15·4	17·6	23·2	26·4	29·7	40·7	55	72·7	99
6. Power variation per 2 per cent. speed variation.—Energy variation × Stroke (inch-lbs.)	9·1	14·75	23·3	27·2	41·5	68	93·5	121	192	292	440	642

TABLE IV.—TEMPLE CONSTANT ENERGY GOVERNORS.

Pattern Number.	201	202	203	204	205	206	207	208	209	210	211	212
1. Revolutions per minute	440	380	330	290	270	250	220	200	185	170	160	150
2. Average energy in lbs.	123·5	165	234	288	414	660	850	1,240	1,595	5,120	2,750	3,850
3. Stroke in inches	1·18	1·38	1·57	1·77	1·96	2·36	2·75	3·15	3·54	3·93	4·33	4·72
4. Power in inch-lbs. Energy × Stroke	146	227	368	560	810	1,560	2,350	3,900	5,650	8,320	11,950	18,200
5. Mean variation of energy for 2 per cent. speed variation (lbs.)	4·95	6·6	9·15	11·55	16·5	26·4	34	49·5	64	85	110	154
6. Power for variation of 2 per cent. in speed. Variation of Energy × Stroke (inch-lbs.)	5·85	9·1	14·8	20·5	32·6	62·5	94	156·5	226	335	632	730

TABLE V.—TEMPLE GOVERNORS—LARGE SPEED VARIATION.

Pattern Number.	301	302	303	304	305	306	307	308	309	310
1. Revolutions per minute, the auxiliary springs being without compression	440	380	330	290	270	250	220	200	185	170
2. Average energy in lbs.	125	174	253	300	436	687	890	1,300	1,690	2,370
3. Stroke in inches	·787	·945	1·02	1·18	1·378	1·57	1·97	2·16	2·36	2·75
4. Power in inch-lbs. Energy × Stroke	98·5	164	260	354	600	1,175	1,760	2,810	4,000	6,500
5. Mean variation of energy for 2 per cent. speed variation (lbs.)	5·04	7·05	10·15	11·9	17·4	27·5	35·6	51·9	68·1	95
6. Power for variation of 2 per cent. in speed. Variation of Energy × Stroke (inch-lbs.)	4	6·66	10·4	14	23·9	43·4	70	112·5	215	260

TABLE VI.—STUMPF GOVERNORS.

	401	402	403	404	405	406
Total stroke G. inches { with auxiliary spring	2·12	2·79	3·11	4	4·92	5·47
Total stroke G. inches { without ,, ,,	1·96	2·56	2·79	3·62	4·45	4·92
Effective stroke ,, { with ,, ,,	1·34	1·81	1·97	2·56	3·15	3·5
Effective stroke ,, { without ,, ,,	1·18	1·57	1·65	2·16	2·67	2·95
Safety stroke ,,	·787	·98	1·14	1·45	1·77	1·96
Revolutions per minute lowest	160	130	105	95	85	80
,, ,, extremity of effective stroke	360	315	290	245	215	195
,, ,, highest	390	340	312	260	230	210
Admissible resistance of transmission mechanism to valve gear, at sleeve, up to (lbs.)	2·2	4·4	11	17·6	35·2	61·6

The tension springs are always of the type known as coil-spring, formed of a spiral of good round steel wire. The square section springs are usually more costly. The steel should have a minimum tensile resistance limit-breaking point of 60,000 to 70,000 lbs. per square inch, and be worked at about 30,000 to 35,000 lbs. per square inch.

Compression springs are usually cylindrical. Conical springs are not

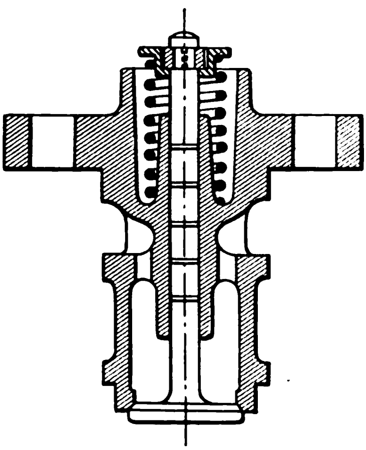


FIG. XIII.—79. Compression Spring on Inlet Valve.

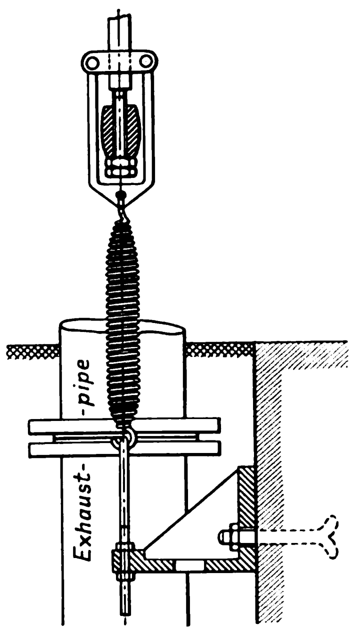


FIG. XIII.—80. Tension Spring on Exhaust Valve.

so much used on account of their increased cost with no appreciable advantage over the ordinary form.

The determination of the power of the springs should be judiciously made. If too weak they will permit the valves opening into the cylinder to be raised from their seats during the suction stroke. If too strong, they throw a useless strain upon the working parts affecting the mechanical efficiency and causing abnormal wear.

In engines governed by hit-and-miss or by constant volume of mixture, the vacuum produced by the suction is, of course, less in amount than in engines governed by the volume admitted. In the first, a maximum depression of ·7 to ·9 lb. is reckoned, and 1·3 to 1·5 lbs. in the second, per square inch of valve area.

The inlet and exhaust springs should therefore be sufficient to resist this depression plus the internal mechanical resistance, and sometimes, according to the condition of the inlet valve, the resistance due to the presence of tar and dirt which tends to make the valves stick to their seats or guides.

In practice, the springs are calculated for three times the power—namely, 6·5 lbs. for inlet and 13·2 lbs. for exhaust, per square inch of valve area. The formula given below is based upon the data just

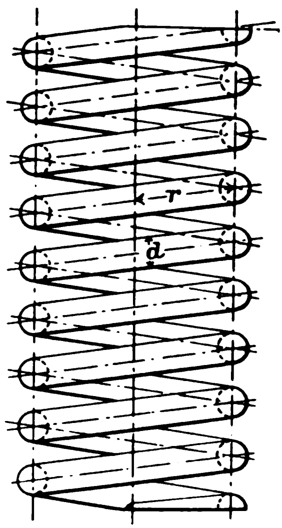


FIG. XIII.—81. Cylindrical Coiled Spring.

mentioned and is used to give the diameter of the wire of which the spring is made (Fig. XIII.—81) :—

$$d = \sqrt[3]{\frac{p\,r}{0{\cdot}196t}}$$

in which :

d = diameter of wire in inches.
p = the load to be applied in lbs.
r = the radius from the centre of the spiral to centre of wire, determined by the drawings, in inches.
t = stress on the metal, taken as 35,500 lbs. per square inch maximum.

To determine the length of the spring, being given the amount of deflexion, f, which is determined by the lift of the valve, the equation used is :—

$$f = \frac{4 \pi n r^2}{d} \times \frac{t}{G}$$

G = the practical coefficient of elasticity for the spring steel.
$G = 0{\cdot}4\,E$.
E = 30,000,000 lbs. per square inch.
n = number of coils.

The extremities of the springs should be flattened so as to bear practically upon their entire circumference. The bearing surface against which the spring acts should be fitted with either a hollow or a projection to keep the spring concentric to the spindle during work. The springs should be shielded from the influence of high temperatures which cause them to become weakened.

CHAPTER XIV

TESTING AND TESTING APPARATUS

Test Sheets.—The preparation of test sheets and schedules for results obtained from gas engine trials has been the subject of consideration by the American Society of Mechanical Engineers which, in 1902, published a series of suggestions to ensure that all desirable data should be properly and systematically noted. The Verein Deutscher Ingenieur and the German Engine Builders' Union have also prepared similar schedules.

Without wishing to criticise these documents it must be stated that a great number of the particulars enumerated cannot be ascertained in the majority of practical tests. In fact the opportunity of carrying out such complete tests occurs only under exceptional and very rare circumstances.

The engines to be tested are always, almost without exception, intended for industrial work and not for scientific investigations and, consequently, they are not provided with the necessary arrangements to enable some of the particulars mentioned to be verified and noted. The experimenter generally finds the engines ready for work and not dismantled, and therefore it is impossible to take all the dimensions required for a complete scientific survey.

Moreover, the tests are frequently conducted under conditions which do not permit absolute accuracy being obtained with the available measuring instruments either in an industrial installation or even upon the test bench of a factory. Finally, as the extent of the test is limited by the expense entailed, this consideration very often forces the operator to confine himself to those points only which are strictly necessary to be determined.

The Object of Tests.—The first rule of any test is to record the purpose of the trial. In the majority of cases the object is to verify the accomplishment of contract specifications, and particularly to check the effective power of the engine and to measure the consumption of fuel, oil, and water. Similar tests are also conducted to establish the most economical conditions of working or to study the operation of the engine.

These form the whole of the indispensable objects of all tests, while other items of a report are principally of a scientific nature, and should be omitted when circumstances are unsuitable for accurate results. At the same time, these scientific particulars must not be deemed as of secondary importance. On the contrary, they are of the greatest value in deducing correct theories relating to internal combustion engines, and all engineers conducting tests should note such points whenever possible. They should always work with the utmost care, taking every possible precaution to guard against errors, repeating similar tests when opportunities permit, and using every available means of checking and verifying results.

The Personal Coefficient.—It is entirely unnecessary to take certain measurements to the hundredth part of an inch, or temperatures to a tenth part of a degree. The desire for exactitude should not extend to three or four places of decimals, which only slightly influence results and are of less importance than the personal skill of the experimenter.

The ability of the experimenter affects definite figures by what may be called a "personal coefficient" to the extent of perhaps 5 per cent., more or less, quite apart from cases where larger errors are sometimes made.

State of the Installation.—When the relations of power and consumption are to be determined in any installation, it is indispensable that all parts should be in good working order. If some essential details permit a considerable amount of play and show excessive wear, if there are any leaky places, or if the governing mechanism be defective, it will be evident from the first that the results will not be favourable. It would, therefore, be useless to proceed under such conditions and would even be unfair if the object of the test were to verify the accomplishment of the contract specifications.

In such a case not only should the contractor be called upon to be present during the test, but he should be permitted to do what is necessary to put the engine in order before the test is commenced.

The different operations comprised in a complete test will be reviewed and an examination made, as briefly as possible, of the methods to apply and apparatus to use.

A complete test includes :—

1. Inspection of working parts.
2. Determination of effective H.P.
3. ,, ,, indicated H.P.
4. ,, ,, speed.

5. Determination of temperatures.
6. " " consumption.
7. Examination and analysis of residues.
8. Conditions of working.
9. Interpretation of results.

1. Inspection of Working Parts.

Description of Engine.—The important details to be noted are:—

(*a*) Position of the cylinder axis or axes (*i.e.*, whether horizontal or vertical).

(*b*) Number of cylinders and motive impulses per revolution of crank shaft.

(*c*) Number of fly-wheels.

(*d*) Number of bearings.

(*e*) Nature of cycle.

(*f*) Type of valve gear.

(*g*) Type of ignition apparatus.

The description of the engine should be briefly expressed thus:—

Example, No. 1.—Horizontal engine; made by *X*; two cylinders; twin; single-acting; fitted with four bearings and one fly-wheel placed between the two cylinders; four-stroke cycle; valve gear; governed by constant ratio of mixture in variable volume; governor varies the lift of mixture inlet valve; low-tension magneto.

Example, No. 2.—Horizontal engine; made by *Y*; one cylinder; double-acting; fitted with three bearings and one fly-wheel on left hand side of engine looking from cylinder to crank; two-stroke cycle; admission by valves and exhaust by central ports; governed by constant ratio of mixture in variable volume; governor varies the admission period of mixture; two low-tension magnetos at each end of cylinder.

Example, No. 3.—Vertical engine; made by *Z*; four cylinders; side by side; single-acting; fitted with five bearings; one fly-wheel overhanging against one of the outer cylinders; four-cycle; valve gear; automatic admission valves; governed by throttling the mixture by a butterfly valve in suction pipe; ignition by jump spark and battery.

Dimensions.—The principal measurements to be taken are the diameter and stroke of the piston or pistons, for these enter into the calculation of the indicated H.P. The piston rod—if existing—should also be measured.

It is wise to take all dimensions before the test is commenced, as it occasionally happens that it is impossible to do so at the finish ; besides which, it is very convenient to calculate various results during the trial, based upon the cylinder dimensions, so as to roughly compute the results and thus avoid large errors due to incorrect measurements.

When the object of the test is the verification of contract guarantees, the experimenter should avoid, as much as possible, communicating any of such rough calculations, which may afterwards be found to be inexact from one cause or another.

The effective diameter of the piston is really that of the cylinder and not of the piston. As a matter of fact if the measurement of the piston be taken from the back, the diameter would be less than the effective diameter owing to the expansion of the piston rings which exactly fit against the internal circumference of the cylinder.

The diameter of the cylinder can be measured, either by callipers or gauge, to the nearest thirty-secondth part of an inch. Any closer measurement does not sensibly affect results in large engines, while in small engines with 6-inch pistons the difference will be less than 1·4 per cent. Although it is preferable to measure the cylinder when hot in order to obtain more exact figures, the difference is, after all, so small that in practice the measurement when cold is quite sufficient.

The length of stroke is taken by placing the crank first at one dead centre and then the other, measuring in one case the distance between the piston and the front edge of the liner—or some convenient point legibly marked—when the piston is right in, and the distance between the same points when the piston is right out. For correct measurement it is important that there shall be no "play" either at the crank pin bearing or piston pin bearing, as, if such exists, the real length of stroke is increased by the total amount of play.

The speed of the engine in revolutions per minute and the corresponding rated output of the engine are usually given by the maker.

Valve Gear.—The complete verification of the valve gear of an engine is a complicated operation, involving, in fact, the dimensions of cams, operating levers, valve lifts, valve diameter, &c., and the position of the piston when the individual valves open and close. As a rule, however, only such details are noted as can be observed without dismantling each piece, the valve-settings being carefully recorded in order to explain the diagrams taken by the indicator.

To do this the fly-wheel is turned by hand, or by the barring gear, to place the crank successively in the different positions co-incident with the opening and closing of the valves, and measurements are taken in each position either of the piston in relation to the front edge of the liner, or a similarly convenient fixed point, or of the crank by noting its inclination from the horizontal by means of a spirit level fitted in a quadrant graduated in degrees.

Supply.—It is necessary to briefly describe the method of serving the engine, to indicate the nature of the fuel and the means adopted for producing it. Under the heading of "consumption" reference will be made to the subject of fuel analysis. For the present the method of feeding of the engine will be dealt with.

Town Gas.—The experimenter should note:—the normal output for which the meter is marked; the type of meter; whether the gas is supplied at sufficient pressure (at least 7·5 tenths (inches) of water); whether any abnormal resistances exist in the supply pipes; and the arrangements, if any, to ensure a regular flow of gas.

In the majority of cases the readings of the gas company's meter is accepted as correct. It is then necessary to note whether the apparatus has been properly levelled on erection. Especially so if a wet meter be employed, as it is important, for correct measurement, that the water level is normal. To observe this, the gas should be shut off from the meter and the plug removed from the orifice placed on the water line, so that the actual water level may be determined and, if necessary, adjusted. If any doubt exists as to the correctness of the counter, it should be checked with a certified standard apparatus.

It is important to measure the leakage of gas from the connections, as it is very rarely the case that these are thoroughly gastight. The gas should be permitted to fill all the gas pipes, including the gas bag, under the normal pressure prevailing, and readings of the meter taken before and after a lapse of 15 to 30 minutes, during which period, of course, all the gas cocks must be properly closed.

Producer Gas.—When the engine is served by producer gas, a brief description should be given of the apparatus employed for generation, washing and purifying, as well as any special arrangement of connections that may exist. When a gas-holder is included of known capacity, it is easy to measure the quantity of gas produced or consumed in a given time.

The observations take different forms according as to whether the working of the whole group "engine and plant," or "engine" and "plant" as separate units, is to be determined. In the first case the most frequent method is to record the consumption of anthracite, coal, or coke in the generator, and in the steam boiler, and to measure also, whenever possible, the consumption of water for gas production and purification. In the second place it will be indispensable to measure the quantity of gas produced in order to appreciate whether the generator furnishes, per lb. of coal consumed, a rational quantity of gas of sufficient calorific value and normal composition, and also whether the engine uses the gas fed to it in a satisfactory manner.

Petroleum, Benzine, &c.—The arrangements provided for carburation should be taken note of, as they influence the consumption of fuel as well as the working conditions of the engine.

2. Determination of Effective Power.

The effective power of an engine corresponds to the work actually available at the crank shaft.

Dynamo.—When an engine drives a dynamo of at least equal power to itself, the most practical method of determining the effective power is to measure the electrical output as shown by aperiodic instruments of proved accuracy. The operations should be carried out in such a manner that the electrical output remains practically constant, being absorbed either by an electric lighting circuit, or by the load of a battery of accumulators, or by "liquid resistance" suitably disposed.

Readings of the instruments should be taken at regular intervals, and the mean electrical load deduced throughout the trial. The efficiency of the dynamo should be determined at the same load and speed as that reached during the test to calculate the effective work transmitted by the engine. If the dynamo is belt-driven, the efficiency of the drive should be taken into account. Usually it amounts to 95·0 to 97·0 per cent.

With a well-built dynamo and a pliable belt of sufficient width, one effective H.P. produced by the engine is converted into electrical work at the rate of 560 to 670 watts per H.P. according as the power varies between 10 and 100 H.P. and over.

Brakes.—If the engine drives gearing, pumps, or other apparatus of which the work and efficiency cannot be exactly determined, the effective power must be measured by means of a brake. Many kinds of brakes are available, and some very ingenious arrangements have been designed to provide automatic working. The details of these various equipments will not be gone into, as other publications are available specially devoted to the subject. Attention will be confined to the two most simple arrangements in common use.

Rope Brake.—This brake is applicable to engines of small power. It consists of a rope or band encircling the fly-wheel or a specially fitted pulley. One end of the band is fastened to a spring balance, and the other is connected to a weight sufficiently heavy to give the necessary friction against the face of the wheel. The arrangement presents several great advantages, such as the simplicity of the devices used,

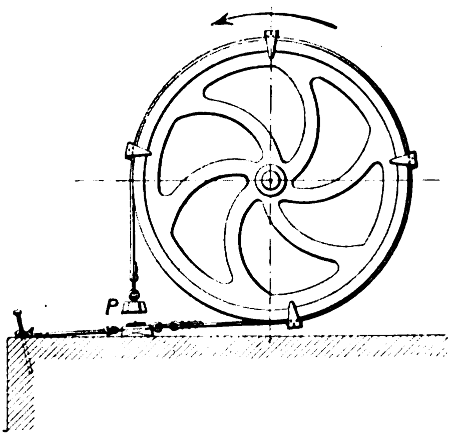


Fig. XIV.—1. Rope Brake.

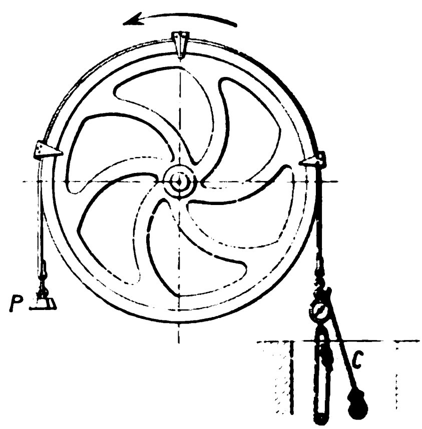


Fig. XIV.—2. Rope Brake with Safety Rope.

and its ready application to the different circumstances which present themselves in practice.

To keep the band or rope on the face of the wheel, three or four pieces of wood are arranged at intervals, having side pieces or cheeks attached to prevent lateral displacement. The negative weight can be replaced by a second spring balance upon which tension is produced. The rope should preferably be made of Manilla hemp, and work by dry friction on the face of the wheel, and, if lubrication be found necessary, a little graphite should be applied. Different arrangements are adopted as circumstances may decide.

To avoid corrections in respect of the weight of the spring balance it should be arranged horizontally as much as possible (Fig. XIV.—1), or very slightly obliquely, or vertically on a support in such a manner that it works freely. When the arrangement shown in Fig. XIV.—2 must be used, care should be taken to suspend the spring balance so that its weight has no influence upon the indications of the pointer, but the weight of the spring balance must be added to the weight that gives the tension to the cord or a coefficient of correction applied if placed obliquely.

To prevent accidents, a flexible safety rope should be fitted, as shown at *C* (Fig. XIV.—2) to one end of the band and fastened to an entirely distinct anchorage.

It is important to arrange the brake so that it embraces the whole of the circumference of the fly-wheel, or, at all events, as large an arc as can possibly be obtained, in order that sufficient adherence may be gained without excessive negative tension. The back pull should not exceed 10 per cent. of the positive tension.

In spite of its advantage, this style of brake should not be used for motors of more than, say, 30 H.P., on account of the dangers involved in its application, unless a specially designed water-cooled pulley can be fitted to take the brake. All the work absorbed by the brake is transformed into heat, and the temperature of the fly-wheel rim becomes appreciably increased while the arms remain cold. As a consequence, stresses due to expansion are set up which may exceed the limit of elasticity in the metal, and if there are any faults in the casting, or even if the metal be perfect, the breakage of the wheel may result. Several serious accidents have thus occurred, causing loss of life and destruction of important material.

When the band can only be applied to the fly-wheel, it should be used for but a very short period, the test being stopped when a noticeable increase in the temperature of the rim is observed. Some builders use the rope brake for large engines on their test benches, but they provide a special pulley with water circulation.

Prony Brake.—The arrangement most frequently adopted is shown in Fig. XIV.—3. The brake is made of two similar wooden jaws bolted to channel irons, the whole forming a balanced mass. It is placed upon a split pulley the deep side plates of which form a trough to receive the water used to absorb the heat of the brake. Keys or suitable expanding bushes are placed between the boss of the brake pulley and the shaft to ensure proper fastening. The same pulley can be used for a series of different size engines under such conditions. Usually the amount of work which can be absorbed per square foot of contact surface is estimated at 12,250 foot-lbs. The temperature of these surfaces must not exceed 165° F. to prevent steam being generated to hinder the operator. Certain precautions must be observed as to the arrangements to apply these brakes satisfactorily.

To prevent the levers moving round with the shaft in the event of any sudden variation of the load, wooden stop blocks are placed in suitable positions to limit the movement of the lever arms.

When the weights are attached in the manner shown in Fig. XIV.—3 care must be taken to maintain the arms in a horizontal direction to avoid the necessity of making a correction on account of obliquity. In some apparatus the weight hangs from a cord bearing against a

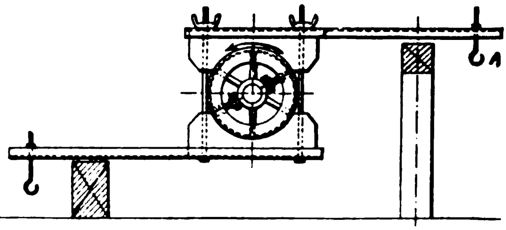


FIG. XIV.—3. Prony Brake.

sector placed at the end of the lever, so that the oscillations of the latter are without influence on the load.

The load can be put on the brake in two ways, either in lifting some weights suspended from (*A*) (Fig. XIV.—3) or by exerting pressure upon a weighing machine, (Fig. XIV.—4). The latter method gives greater stability if arranged as indicated. Reference to Fig. XIV.—3 will show that the centre of gravity is below the centre of suspension and therefore the equilibrium of the parts is unstable. In Fig. XIV.—4 the

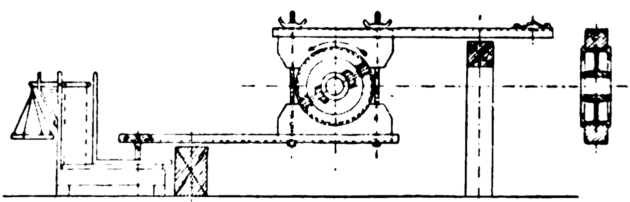


FIG. XIV.—4. Prony Brake with Platform Scales.

illustration makes it clear that the centre of gravity is above the point of application and is therefore preferable.

For ready tightening the bolts should be screwed with a fine thread and the tightening nuts fitted with large wings.

The brake jaws should have internal oil grooves and a hole to pour oil in, should excessive tightening cause heating.

To prevent the water splashing about, by reason of a too copious supply in the pulley or leaky joints, the brake jaws are surrounded by a shield. The lever arms should be counterbalanced either by a symmetrical opposite member or by a counterweight. The equilibrium of the arrangement can be verified by placing the upper jaw on a knife edge placed in the vertical plane of the axis of the pulley.

Effective Work.—This is given by the formula :—

$$B.h.p. = r \times N \times w \times 0{\cdot}001907$$

in which :—

r = radius of brake wheel + half diameter of brake rope, in feet, or,

r = length of lever arm from centre of shaft to centre of weight suspension, in feet.

N = number of revolutions per minute.

w = the net load expressed in lbs.

Prony brakes as described can be relatively easily fitted to engines up to 100 to 150 H.P. For engines of 300 to 500 H.P. the same type can be used if two apparatus are fitted upon the engine shaft. Above 500 H.P. brake tests are usually considered to present too many difficulties and to involve too great an expense. It is therefore upon rare occasions that such tests are made.

The testing of an engine in industrial work by means of a Prony brake is always a difficult operation. Usually the engine pulley has to be taken off and a special brake pulley fitted, and if the engine has three bearings, and if one of these pulleys are cast solid, it is necessary to lift the shaft. Very frequently the space available round the engine is very limited and no facilities exist for cooling the brake by water under pressure. According to circumstances, arrangements must be made so as to ensure the easy working of the appliances.

Several makers, such as, for instance, the Swiss Society for the Construction of Locomotives and Machines, Winterthur, provide their engine crank shafts with a short overhanging portion on the cam shaft side, which is an arrangement much to be commended. It gives a manufacturer a chance of placing a driving pulley in this position, while it becomes a very simple matter to conduct a brake test of the engine.

It will be evident from the preceding remarks that the testing of an engine by a brake when erected upon its permanent foundations is not at all a simple matter, and it is much preferable to conduct all such tests for power and consumption at the makers' testing shop. Then one or other of the usual brakes can be used or some form of dynamometer such as that made by Heenan & Froude of Worcester (England) (Fig. XIV.—5).

This apparatus is usually direct coupled to the shaft of the engine or machine of which the power is to be determined, but it can also be operated by belt.

It consists of a turbine full of water, turning in a box mounted on rollers. The water put into motion by the turbine produces a reaction on the box and tends to make it rotate. The power developed is measured by the weight applied to equalise the effort of reaction as in

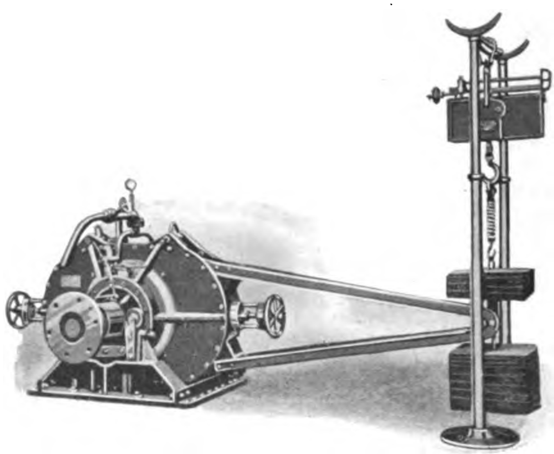


FIG. XIV.—5. Heenan & Froude Dynamometer.

the case of the ordinary brake. A movable balance-weight is provided with which the equilibrium can first be adjusted.

The turbine vanes and those in the box are combined in such a manner that when the apparatus is in action the water continually circulates and passes from one vane of the turbine to an opposing vane of the box and so on. By the effect of centrifugal force the speed of water circulation is excessively great.

To reduce the power absorbed by the dynamometer, a thin metallic sheet is interposed between the turbine and the box to annul the gyratory action. The power can thus be reduced to one-thirtieth of the total work.

The water should be constantly renewed. It is introduced at about 5 lbs. pressure and the heated water flows from the upper part of the apparatus together with the steam formed and any air which may have penetrated into the casing. The maximum power absorbed by the machine varies proportionally to the cube of the speed.

The difference in temperature between the inlet and outlet of the water and the quantity used, enables the exact power to be calculated. Professor Reynolds and Mr. Marley have been able to effect the verification of Joule's law for the mechanical equivalent of heat, by the aid of this apparatus.

3. Determination of Indicated Power.

The indicated H.P. of an engine, single-cylinder, four-cycle, single-acting, is given by the formula :—

$$\text{I.H.P.} = \frac{A\ L}{792000} \times N \times Pm$$

in which :—

A = area of the piston expressed in square inches.
L = stroke of the piston expressed in inches.
N = number of revolutions per minute.
Pm = mean pressure per square inch of piston area during one cycle expressed in lbs.

To measure the mean, or average, pressure, diagrams must be taken by an indicator. Indicators which have been strikingly improved during recent years, are derived from the instrument invented by Jas. Watt and modified by McNaught. It is impossible to mention in this book all the different types at present available, but mention must be made of the indispensable qualities that an indicator should possess :—

1. The piston should be absolutely gastight, without, however, resulting in friction between the piston and the cylinder.
2. The piston, its rod and levers should be of very light weight to reduce inertia effects to a minimum.
3. The adjustment of all pivots and knuckle-joints should be made with great care, the least play in the joints falsifying the diagrams traced by the pencil.
4. For use with producer gas engines, the piston and cylinder should be of steel, as the ammonia and sulphuretted hydrogen in the gas attack gun-metal.

The first three points play a very important part in the accuracy of the diagrams. The results obtained would be absolutely false if the indicator did not possess these qualities.

Fitting the Indicator.—The engine should be fitted with a passage communicating with the combustion chamber for the reception of the indicator. The orifice is usually closed by a screwed plug, but in high compression engines the plug is provided with a prolongation to completely fill the aperture up to the internal wall of the chamber so as to prevent any early firing that might be caused by the accumulation of burning gases in the passage.

Usually the indicator plug is tapped for a $\frac{3}{4}$-inch Whitwood thread, but some makers adopt other sizes and threads. It would be most convenient if all engine builders and indicator makers would adopt the same thread, so as to obviate the use of special adaptors to suit each particular case. Many engine makers provide all their engines with the necessary indicating passage, but this practice has not yet become general.

It is absolutely necessary to make use of an indicator not only for the determination of indicated power, but also for the verification of the valve setting, especially upon erection. Many contractors undertake the erection and setting the engine to work without providing their erectors with indicating gear, and under such circumstances it frequently happens that difficulties in working cannot then be detected in the absence of indicator diagrams.

Position of the Indicator.—Great variation exists in the position of the indicator passage. In the former type of town gas engines in which the valves were arranged under the combustion chamber, or at the side, the indicator was placed vertically above the breech end. In later types this place is taken up by the inlet valve and its mechanism, and the indicator is fixed in a less convenient position, either obliquely near the flange of the combustion chamber, horizontally at the side of the casing, or horizontally at the back.

It is difficult to formulate a general rule for the choice of a position for the passage. It is a matter already determined by the maker, and the operator must take the necessary steps to obtain suitable and proper working conditions.

Indicating Gear.—For single-acting engines, the indicator motion can be obtained by means of one of the following methods:—

(*a*) Parallel motion.
(*b*) Crank pin screwed in the end of the crank shaft.
(*c*) Mathot's patent claw.

For double-acting engines:—

(*d*) Rod fixed to the crosshead.

(*a*) *Parallel motion.*—This is rarely used because the engine frame

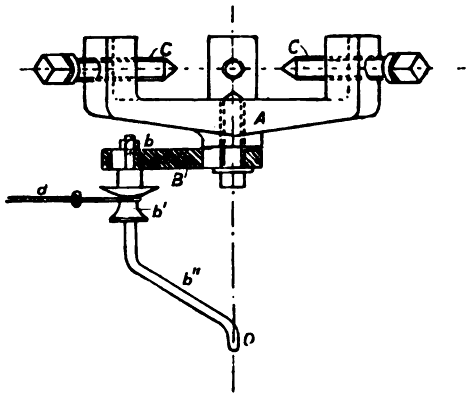


FIG. XIV.—6. Mathot's patent "Claw" Reducing Gear.

is not fitted with the necessary arrangements for fixing the different parts. It is principally used in makers' workshops.

(*b*) *Crank pin.*—For this arrangement, a hole must be drilled and tapped in the end of the crank shaft to correspond with the crank pin.

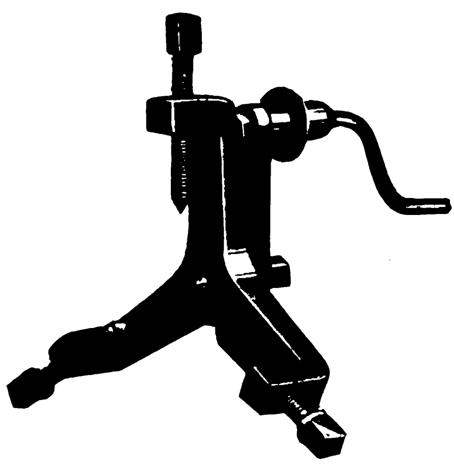


FIG. XIV.—7. Mathot's patent "Claw" Reducing Gear.

Usually the hole has not been drilled by the makers during construction, and it is consequently somewhat difficult to make it coincide exactly with the main crank centre, with the result that the pin is more or less out of truth.

(c) *Mathot's patent claw.*—This is composed of three branches at 120°, and each fitted with a steel pointed screw (Figs. XIV.—6 and 7), to enable the apparatus to be firmly attached to the engine shaft. The central screw *A* permits the claw to be accurately centred on the engine shaft, and fixes the crank *B* in the proper

FIG. XIV.—8. Indicator Cord Adjuster.

direction and at the desired length. The crank pin is furnished with a curved rod *b*, *b'*, *b''*, having a groove at *b'* to retain the driving cord. The ring attached to the end of the latter is readily threaded over the curved rod owing to the fact that the extremity *O* forms a prolongation of the axis of the crank shaft, and has no eccentric movement. The cord can just as readily be detached when it is desired to disengage the indicator.

(d) *Rod fixed to crosshead.*—This is fitted in the same manner as for steam engines, and the connecting cord is taken from reducing gear.

Indicator Cord.—The indicator cord should be made of plaited hemp, not twisted, preferably with a core of very fine brass wire. It should be protected from moisture. When using the cord upon a crank pin, a light copper ring should form the connection, and it is necessary to give this a drop of oil occasionally to avoid resistance due to friction. The cord should be as short as possible and free from kinks and knots. The length can be readily adjusted by means of a plate, as shown in Fig. XIV.—8, or by means of a clamping hook, which usually is one of the accessories provided with the instrument (Fig. XIV.—9).

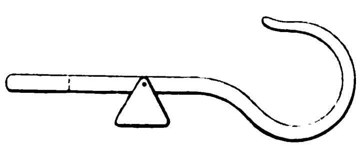


FIG. XIV.—9. Indicator Cord Adjuster.

The length of the cord necessarily depends upon the position of the indicator. Usually the points of attachment upon the indicator drum and the reciprocating gear are not in the same vertical plane, and it is therefore necessary to use a small pulley, suitably arranged, to lead the cord in the required direction without abnormal resistance due to friction. The spindle of the pulley should be carefully oiled.

To adjust the travel of the cord, the engine crank shaft is placed successively at each dead centre, and the stroke of the indicator drum should be a little less than the extreme movement possible Care should be taken to see that the indicator drum does not reach its limit of stroke in advance of the engine crank, otherwise the diagrams will be affected as shown in Figs. XIV.—10 and 11, which show, respectively, the drum motion stopped momentarily at the inner and outer dead centres of the crank.

Manipulation of Indicator.—Before using the instrument it should be seen that it is securely fixed to the engine. The piston should be removed and carefully lubricated with cylinder oil. At the outset a

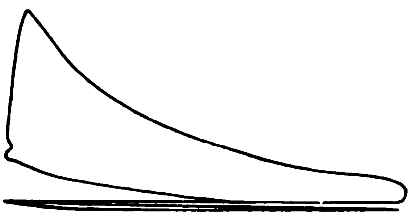


FIG. XIV.—10. Indicator Diagram showing faulty Cord Adjustment.

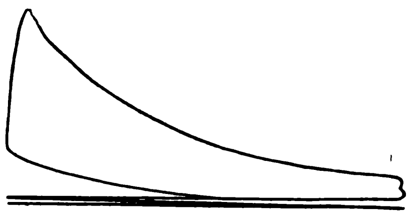


FIG. XIV.—11. Indicator Diagram showing faulty Cord Adjustment.

strong spring should be used to record the explosions, so as to protect the levers from damage in the event of a sudden shock, due to unexpectedly high explosion pressures.

The springs and cylinder cover should be very carefully secured, and be screwed right home. The pencil should be adjusted to mark the paper legibly but very lightly.

Diagrams.—The different kinds of diagrams to be taken are : —

(*a*) Explosion or combustion.
(*b*) Compression.
(*c*) Resistance.

Explosion Diagram.—The explosion diagram takes the form shown in Fig. XIV.—12, and it is from this that the indicated power is measured. There is a distinction between engines governed by "hit-and-miss" and those by variable admission of mixture.

If the engine is in good order all the diagrams taken from "hit-and-miss" engines are of similar shape and have practically the same area, but it is necessary to count the number of "misses" when the engine is not working at its full output. These can be counted by observing the movements of the gas valve, or by listening to the noise of the exhaust, which is easily recognised after each explosion. It is the better plan, however, to record the exact number of cycles

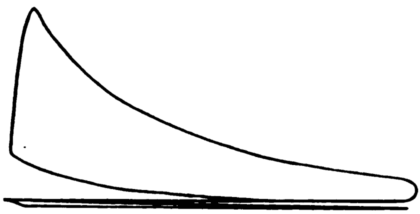


FIG. XIV.—12. Typical Indicator Diagram—Four-cycle Engine.

by means of the author's explosion recorder (described in Chapter XV.), which can be adapted to all ordinary instruments.

In engines governed by variable admission of mixture, the diagrams are frequently very different from one another, according to the action of the governor, the mixture being varied either in quantity or composition and, consequently, the explosive power, and the conditions of expansion change for each cycle. It is indispensable, therefore, in

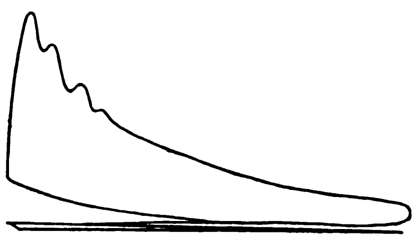


FIG. XIV.—13. Indicator Diagram taken with unsuitable Spring.

such cases to take a sufficient number of diagrams to obtain a mean area corresponding exactly to the indicated work of the engine.

The calculation of the indicated power depends upon the measurement of the mean pressures of the diagrams. It is therefore of the utmost importance to adopt certain precautions to avoid all risk of error.

Springs.—The calibration of the springs used should be carefully checked before and after the test. As a rough guide it is a good plan to use two springs of varying strengths, so as to obtain a practical verification of their accuracy. Springs that are too light should not be used, as they give rise to inertia effects similar to those depicted in Fig. XIV.—13.

Planimeter.—The diagrams should preferably be measured by means of a polar planimeter, an instrument sufficiently well known to require no description in these pages. The measurements should be made with the utmost care.

Calculation of Indicated Power.—The indicated H.P. of a single-cylinder, four-cycle, single-acting engine, can be obtained from the formula already given, but repeated here :—

$$I.h.p. = \frac{L\ A}{792000} \times N \times Pm$$

in which L = stroke of piston in inches.
A = area of piston in square inches.
N = number of revolutions per minute.
Pm = mean pressure measured from average diagram, in lbs. per square inch.

When the engine is running without load the indicated H.P. then found, distinguished by the symbol P_o, will correspond to the work absorbed by the friction of working parts and will usually give the equation

$$I.h.p. = B.h.p. + P_o.$$

However, it should be noted that the work developed under "no load" to overcome the mechanical friction is a little more than the same work when under load, owing to the suction resistances of the mixture being greater in the former than in the latter.

The no load resistances should be determined after the working load has been noted, while the engine is under the same conditions as regards temperature. The diagrams should not be taken while the speed of the fly-wheel is accelerating or diminishing. For a given load the indicated power measured by the indicator is equal to the sum of the corresponding effective power, and of the indicated no load power. The verification of this formula at various degrees of load enables the correctness of the indications to be demonstrated.

The mechanical efficiency of a given load will be :—

$$R = \frac{B.h.p.}{I.h.p.} = \frac{B.h.p.}{B.h.p. + P_o} = \frac{1}{1 + \frac{P_o}{B.h.p.}}$$

The above formula is generally accepted with respect to ordinary four-cycle engines, but in the case of four-cycle engines with separate compressing pump, such as the Diesel, for example, or of two-cycle engines, engineers are not in accord with regard to the manner in which the mechanical efficiency should be expressed.

Some say that the following formula applies :—

$$R = \frac{B.h.p.}{I.h.p.\,(gross)},$$

but others consider that

$$R = \frac{B.h.p.}{I.h.p.\ (net)}$$

is the correct form—*I.h.p.* (*net*) representing the indicated work

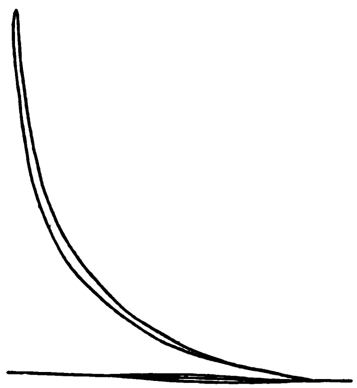


FIG. XIV.—14. Indicator Diagram of Compression of Charge (without ignition).

measured in the engine cylinder, as accounted for by the diagrams, less the indicated work measured in the pumps.

The author agrees with the opinion given by Professor Diederichs, that the first formula gives the nearer approach to truth, except, perhaps, for the Diesel engine, in which, up to a certain point, the work done in compression of the separate air charge is recovered in the engine cylinder.

Compression.—The terminal compression pressure can be measured from the explosion diagrams, but it is useful to record them separately to a larger scale by using a lighter spring, but of sufficient resistance to ensure the full height being traced before the limit of movement of the spring is reached.

When using such a spring care should be taken to prevent the force of the explosion causing damage to the pencil movement by putting the ignition gear momentarily out of action before opening the indicator

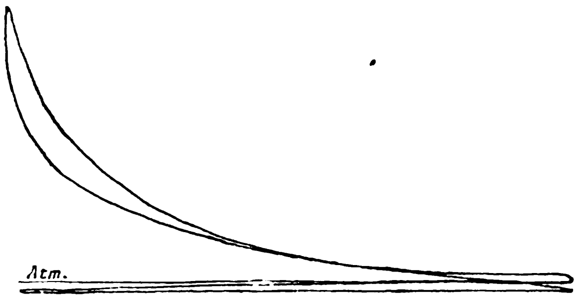


FIG. XIV.—15. Compression Diagram showing loss on Expansion of Charge.

cock and taking the diagram. Fig. XIV.—14 shows the form of the curves then obtained.

The records obtained by closing the gas cock would show a diminution in the compression pressure when power gas is used, since the volume of gas is, in such cases, practically equal to the volume of air.

If the cylinder is not gastight the diagram will be of the form

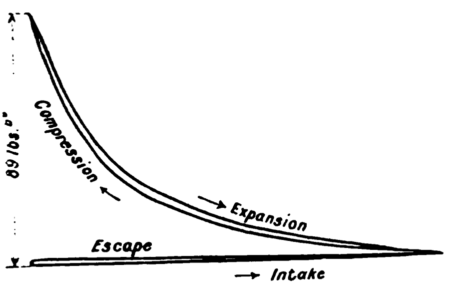


FIG. XIV.—16. Compression Diagram from Engine with cold walls.

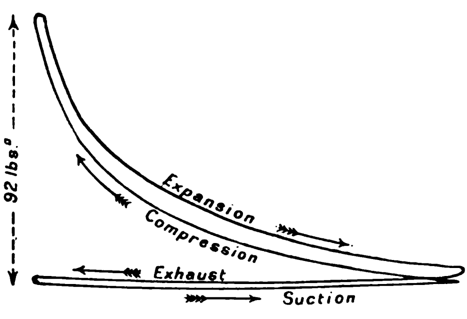


FIG. XIV.—17. Compression Diagram from Engine with hot walls.

shown in Fig. XIV.—15. The influence of wall-temperature on the pressure of gas in the cylinder is shown in Figs. XIV.—16 and 17. In Fig. XIV.—16 the engine is working "cold"—that is, with a copious supply of water. The compressed gas gives out heat to the walls and the expansion curve is in close proximity to the compression curve. In the case of Fig. XIV.—17 the engine is working with a hot cylinder, the water leaving the jacket at about 160° F., the walls giving heat to the compressed gas and the expansion curve rising above the compression curve.

Resistances.—The measurement of the resistances to suction and exhaust can only be made with a very light spring, the limit of

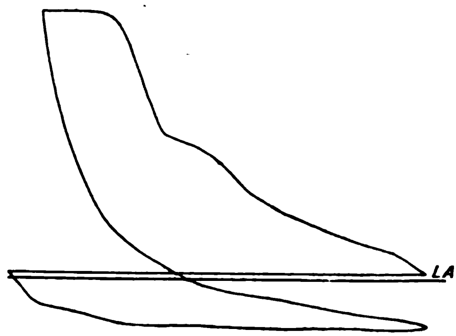


FIG. XIV.—18. Resistance Diagram from throttle-governed Engine, light load.

movement being reached before the end of the compression stroke, so that the explosion will not injure the indicator mechanism.

The amount of suction resistance varies according to whether the engine is fed with town gas or power gas, and, in the latter case, in accordance with the method of governing and the degree of the load. Fig. XIV.—18 is a "no load" suction diagram and Fig. XIV.—19 a

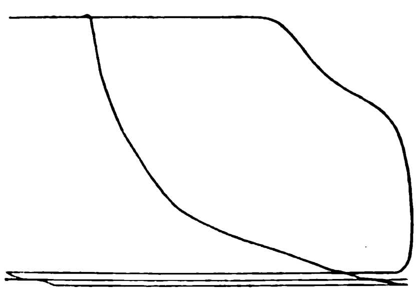


FIG. XIV.—19. Resistance Diagram from throttle-governed Engine, working load.

"working load" suction diagram, both taken from an engine served by a suction gas producer governed by variable admission of the quantity of gas.

Interpretation of the Indicator Diagram.—At the annual meeting of the American Society of Mechanical Engineers, held in New York in

December, 1907, Professor Lucke, of Columbia University, presented a very interesting report upon the flame propagation in explosive mixtures within gas engine cylinders.

He reproduced the results of a series of trials made specially to study the explosive waves occurring during the period of inflammation

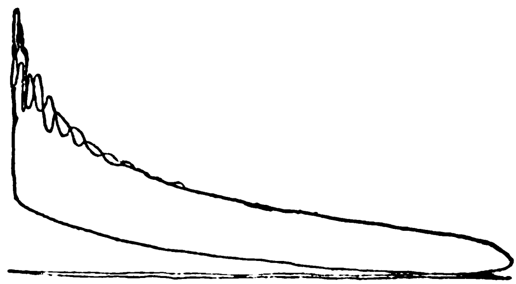


FIG. XIV.—20. Indicator Diagram showing Inertia Effect of Indicator Mechanism.

of the mixture, by means of the indicator, and published a collection of characteristic diagrams in this connection.

Without wishing to detract from the value of Professor Lucke's work, the author discussed several of the deductions that were put forward

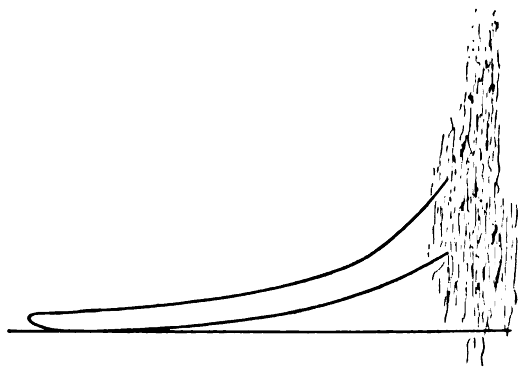


FIG. XIV.—21. Indicator Diagram showing Explosion in Indicator Tube.

with regard to the forms of the indicator diagrams contained in the report, and represented in Figs. XIV.—20 to 24, before the meeting referred to.

As a general rule, unless a special study is made of the diagrams themselves and of the special conditions under which they are taken, it is difficult to distinguish between the irregularities in the curves arising from certain phenomena within the engine cylinders, from defects of erection, or of working, or of the indicators themselves.

For example, with regard to Fig. XIV.—20 the author cannot agree

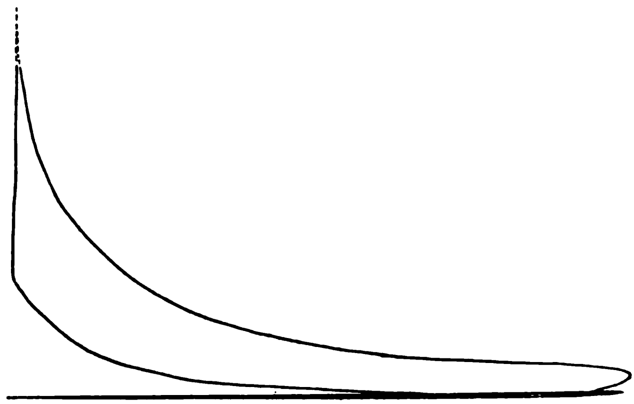


FIG. XIV.—22. Indicator Diagram showing Pencil Vibration.

with Professor Lucke's conclusions. The undulations in the upper part of the expansion curve are not due to waves in the explosive force of the burning mixture but are simply caused by the inertia of the indicator piston and of the parallel motion. The same form is usually

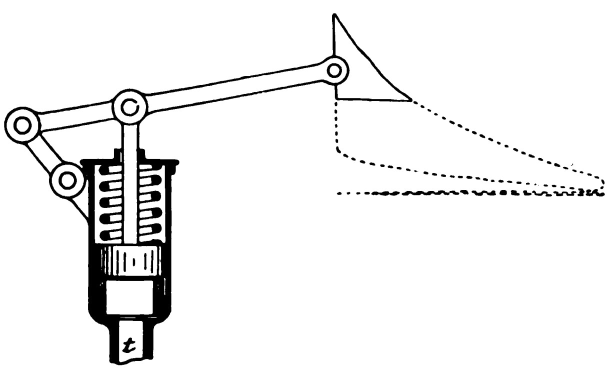


FIG. XIV.—23. Experiment to determine Effect of Inertia of Indicator Mechanism.

observed when too light a spring is employed by reason of the speed at which the engine runs.

Diagram XIV.—21 seems to indicate a different form of wave which would commence with the beginning of the expansion curve such as is generally obtained when there is premature ignition, or when very sudden and very violent explosions are produced either in the engine cylinder or the indicator itself. In the latter case, the phenomena is explained by the very rapid vibrations of the pencil movement.

When violent explosions occur, such as those due to the combined effects of early ignition and of a mixture too rich in hydrogen, for example, the top of the line representing the initial pressure, instead of terminating by a rounded angle, is really prolonged in a dotted line, giving evidence that the pencil was vibrating on the paper, as is indicated in Fig. XIV.—22 taken by the author under these conditions.

To exactly determine the source of these defects and to eliminate them, the author in similar circumstances has used a special arrangement, shown diagrammatically in Fig. XIV.—23, consisting of a short tube, *t*, placed in the indicator cylinder, immediately below the piston, so that it partially compressed the indicator spring. In this way the pencil could only register the higher pressures on the diagram, as shown in full lines, when the explosion occurred. The dotted portion shows the lower part of the diagram that would have been traced in the absence of the tube inserted in the cylinder. The only lines recorded, shown in full lines, are the end of the explosion line, the beginning of expansion, and a line—of no account—parallel to the atmospheric line. The moving parts of the indicator are therefore shielded from inertia effects by reason of their reduced strokes. This procedure enables regular lines to be recorded, exempt from undulations shown in Fig. XIV.—20, 21, and 22, and evidently proves that the alleged waves are not produced within the engine cylinder, but are caused by the effects of inertia of the indicator itself.

Figs. XIV.—24, 25, and 26, presented by Professor Lucke, show the existence of 8 to 10 undulations during the expansion stroke—or during one-half revolution of the engine, which in normal operation would make, say, 200 revolutions per minute. If these undulations really represented explosive waves it must be admitted that the indicator is capable of giving an exact and true record even when fitted to an engine making $2 \times 8 \times 200 = 3{,}200$ or 4,000 revolutions per minute. Specialists in connection with indicator tests know that no instrument exists, of an ordinary construction, that is capable of giving exact indications on a gas engine running at a higher speed than 400 or 500 revolutions per minute, whatever the makers of these instruments may say.

With regard to other erroneous records which may be given by indicators, reference should be made to the apparent scavenging shown by the exhaust line falling under the atmospheric line when a very light spring is used, with a piston stop, to prevent the complete lines

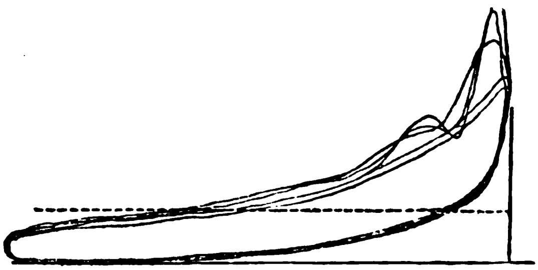


Fig. XIV.—24.

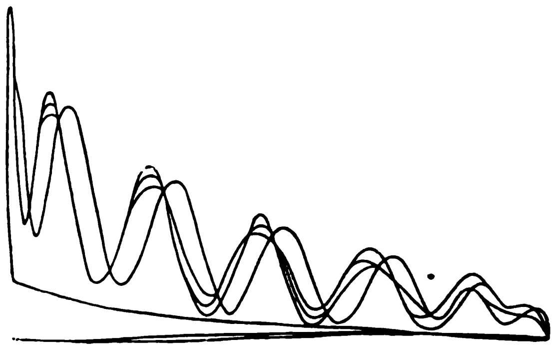


Fig. XIV.—25.

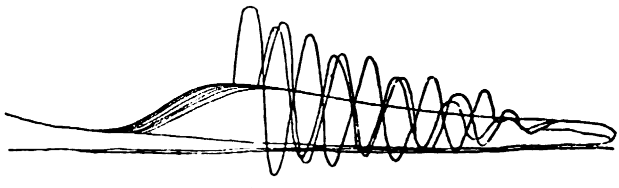


Fig. XIV. 26.
Undulations due to Spring Vibration.

of explosion and expansion being traced, so that only the suction and exhaust pressures are recorded.

From these diagrams many constructors have wrongfully deduced that their engines are scavenged by inertia effects of the gas, whereas the cards show rather an inertia effect of the moving parts of the indicator, giving a drop in the exhaust line below that of the atmosphere (Fig. XIV.—27). As a matter of fact, from the study of diagrams taken during numerous tests conducted by the author since 1890 on all kinds of engines, only in 20 or 25 cases does scavenging about the end of the exhaust really take place. Experience has shown that in order to obtain reliable resistance diagrams the instrument should be fitted with extremely light moving parts, while the spring used should

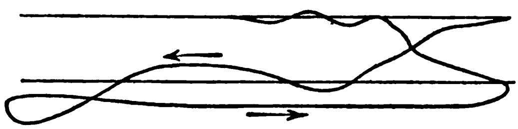


FIG. XIV.—27. Resistance Diagram showing effect of Spring Vibration on Exhaust Curve.

be $\frac{1}{24}$ or $\frac{1}{30}$ of an inch per lb. This spring gives a record sufficiently intelligible to measure the resistance of exhaust and vacuum of suction without being affected by inertia effects of the spring, &c. (see Fig. XIV.—28).

The author does not deny the existence of explosive waves of the nature referred to by Professor Lucke and of which the latter has taken several records during his interesting experiments, but it is necessary

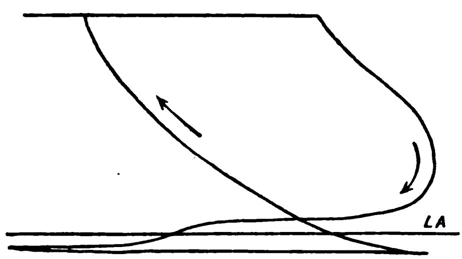


FIG. XIV.—28. Resistance Diagram showing Scavenging by Fluid Inertia.

to guard against placing too much faith in the indicator with respect to some phenomena that they appear to record.

It is evident that some physical conditions, such as the lack of homogeneity in the mixture, might produce wave effects, or, more exactly, some explosive waves will be produced during the propagation of flame in the mixture. Other factors, of a mechanical origin, can also originate them, such as the vibration of the combustion chamber walls or of the back plate of the piston when the explosions are sudden and violent.

The presence of waves usually results from the explosion of very rich mixtures which are predisposed to ignite prematurely, that is before the piston has reached the dead centre and at the instant when it is, therefore, momentarily at rest. If then the vibrations of the enclosing metal synchronise with those which are propagating in the fluid, waves may be recorded in the indicator diagram.

In any case it is to be hoped that Professor Lucke will continue his experiments on larger engines because they will certainly serve to throw a light upon this complex subject.

4. Determination of Speed.

The number of revolutions of the engine shaft per minute enters into the calculations of effective and indicated power and, therefore, must be exactly determined; consequently a chronometer should be used precisely marking the seconds. It is useful, also, to attach a revolution counter to note the total number of revolutions and thus to obtain the average speed during a certain period of which the chronometer can determine the beginning and the end.

In the absence of a counter, the number of revolutions can be similarly noted by observing the movement of one of the valve levers or by the noise of exhaust. When this method is adopted it must be remembered that from putting the seconds hand of a chronometer in motion the counting must start from zero and not from one. Many operators have, in the author's presence, counted the first movement as one, and when the counting is from the half-speed shaft this results in a double error as regards the speed per minute attributed to the engine.

Different instruments, such as tachometers and tachographs, may also be used, but it is unnecessary to enter into the details of their construction.

Variations of speed ought to be noted for each change in the engine's load. In a good engine the variations lie between 2 and 4 per cent. of the number of revolutions when the load suddenly falls from full to quarter and inversely. The relative variations between the working load and no load presents no interest industrially.

The ratio resulting from speed variations is expressed as follows:—

$$\left.\begin{array}{r}\text{Degree of irregularity in}\\ \text{revolutions per minute}\end{array}\right\} = \frac{N\ max. - N\ min.}{\left(\frac{N\ max. + N\ min.}{2}\right)}$$

The cyclic variation, that is to say the variation of angular speed in the same revolution, is expressed in a fraction of the time during which the revolution is made. This is called the coefficient of cyclic irregularity, and the subject has been dealt with in the chapter relating to fly-wheels. The term "cyclic" has no reference to the "cycle" of working.

5. Determination of Temperatures.

Certain temperatures ought to be taken for the verification of the fuel consumption of an engine. For town gas, for example, it is indispensable to reduce the consumption to the standard basis of comparison (0° C. and 760 mm.) or 32° F. temperature and 29·9 inches of mercury pressure.

The temperature T is that of the gas in the meter used to register the consumption. The influence of temperature is relatively important upon the consumption, corresponding to about 1 per cent. for every 5½° F.

The guarantees given for consumption generally specify that the measurements should be in terms of the standard basis, and even in the absence of such stipulation the common custom is to make such correction for all tests.

Other temperatures are also interesting to have recorded from a scientific point of view and for taking into account the working of the complete apparatus, such as:—

For engines: Cooling water at inlet and outlet of all parts provided with water circulation.
Exhaust gases.

For producers: Gas at outlet of generator.
Gas at inlet and outlet of scrubber.
Water in the vaporiser of the generator.

The formula applied to reduce the volume of gas to 0° C. and 760 mm. pressure is as follows:—

$$V_o = V_t \frac{p}{760} \times \frac{273}{273 + t}$$

in which:

V_o = volume of gas in terms of 0° C. and 760 mm.
V_t = ,, ,, at temperature, t, and pressure, p.
p = the atmospheric pressure prevailing at time of test in mm. of mercury.
t = the temperature of the gas measured in C.°.

The same formula using British standards of measurement is:

$$V_o = V_t \frac{p}{29 \cdot 92} \times \frac{491 \cdot 4}{459 \cdot 4 + t}$$

in which:

p = atmospheric pressure in inches of mercury.
t = temperature of gas in F.°.

The temperature of the circulating water can be readily taken when the outlet is visible. For motors of considerable size the different parts, such as cylinder, breech end, exhaust pipe, and also, occasionally, the cylinder covers and pistons, are each provided with separate circulating arrangements. It is then necessary to note the temperature on each circuit both on inlet and outlet.

CO-EFFICIENTS FOR REDUCING VOLUME OF GAS TO STANDARD PRESSURE (760 MM.) AND TEMPERATURE (0° C.).

Millimetres of Mercury.	8°.	9°.	10°.	11°.	12°.	13°.	14°.	15°.	16°.	17°.	18°.	19°.	20°.
740	0·946	0·943	0·940	0·936	0·933	0·930	0·926	0·923	0·920	0·916	0·913	0·911	0·908
741	947	944	941	937	934	931	927	924	921	917	914	912	909
742	949	945	942	938	935	932	928	925	922	918	915	913	910
743	950	946	944	940	937	934	930	927	924	920	917	914	911
744	951	948	945	941	938	935	931	928	925	921	918	915	912
745	952	949	946	942	939	936	932	929	926	922	919	916	913
746	954	950	948	944	941	938	934	931	928	924	921	918	916
747	955	952	949	945	942	939	935	932	929	925	922	919	916
748	956	953	950	946	943	940	936	933	930	926	923	920	917
749	958	954	951	947	944	941	937	934	931	927	924	921	918
750	959	955	952	948	945	942	939	936	933	929	926	923	920
751	960	957	953	949	946	943	940	937	934	930	927	924	921
752	961	958	954	950	947	944	941	938	935	931	928	925	922
753	963	959	956	952	949	946	942	939	936	932	929	926	923
754	964	960	957	953	950	947	943	940	937	933	930	927	924
755	965	962	958	954	951	948	944	941	938	934	931	928	925
756	966	963	960	956	953	950	946	943	940	936	933	930	927
757	968	964	961	957	954	951	947	944	941	937	934	931	928
758	969	966	962	958	955	952	948	945	942	938	935	932	929
759	970	967	964	960	957	954	950	947	944	940	937	934	931
760	972	968	965	961	958	955	951	948	915	941	938	935	932
761	973	969	966	962	959	956	952	949	946	942	939	936	933
762	974	971	967	963	960	957	953	950	947	943	940	937	934
763	975	972	969	965	962	959	955	952	949	945	942	939	936
764	977	973	970	966	963	960	956	953	950	946	943	940	937
765	978	974	972	968	965	962	958	955	952	948	944	941	938
766	979	976	973	969	966	963	959	956	953	949	945	942	939
767	981	977	974	970	967	964	960	957	954	950	946	943	940
768	982	978	975	971	968	965	961	958	955	951	947	944	941
769	983	980	977	972	969	966	962	959	956	952	949	946	943
770	984	981	978	973	970	967	963	960	957	953	950	947	944
771	986	982	979	975	972	968	965	962	958	955	952	948	945
772	987	983	980	976	973	970	966	963	960	956	953	950	947
773	988	985	981	978	974	971	968	964	961	957	954	951	948
774	989	986	982	979	976	972	969	965	962	959	955	952	949
775	991	987	984	980	977	973	970	966	963	960	957	953	950

In order to establish a heat balance-sheet the quantity of water used over a definite period as well as the temperature must be noted. To do this the water must be measured, either by gauge or by weight according to circumstances, for each circuit.

The temperature of the exhaust gas should be measured in the immediate neighbourhood of the exhaust valve, outside the cylinder. As the temperature usually exceeds 650° F. a mercury thermometer cannot be used. The most practicable method is to make use of a calorimeter of the Salleron type. The temperature can also be calculated by the method indicated by the American Society of Mechanical Engineers, by making a chemical analysis of the burnt gas and judging the quantity of air introduced in the explosive mixture.

The temperatures at the various portions of the producer are easily noted by means of thermometers.

6. Determination of Consumption.

The three elements, fuel, oil, and water, are used to ensure the proper working of gas power installations, and the amount consumed of each indicates the relative perfection of the system or apparatus.

Fuel.—The fuel used is either gaseous or liquid. If gaseous, it is furnished either by an entirely independent source, such as from town gas mains, blast furnaces, coke ovens, natural gas, &c., or it is produced by a gas generator coupled to the engine.

In the case of a gas producing plant, either the gas itself or else the solid fuel burnt in the generator is measured, according to the arrangement of the apparatus.

If liquid fuel is used it is contained in a vessel large enough to hold a sufficient quantity to serve the engine during a definite period.

With regard to liquid fuel it is necessary to measure its density, temperature, and calorific value, and a chemical analysis should be made from a sample taken to fairly represent the mass as fed to the engine. The quantity of fuel should be measured, as taken by the engine during a determined period.

To do this, after noting that no leaks exist, the reservoir is filled up to a certain level, which is very carefully noted, or legibly marked. If during the test it becomes necessary to refill the reservoir, the amount added should be weighed or gauged very exactly and poured into the reservoir so as to avoid splashing or waste. At the end of the test the reservoir is again filled until the level exactly reaches the former level, the quantity thus added being similarly weighed or gauged. From these observations the quantity consumed during the trial is determined.

As regards gaseous fuel, whatever the nature of the gas, it is indispensable to determine its calorific power. This varies slightly, as a rule, during the tests with town gas, blast furnace gas, coke oven gas, &c. It may vary between certain limits if the gas is made in a pressure producer to which a gas-holder of some size is connected. It may vary very considerably if the gas is obtained from a suction gas plant. In such a case it is necessary to make a number of calorimetric and chemical analyses, because such results constitute a valuable record of the working of the engine and producer.

Professor Treadwell, of Zurich, several years ago, made some very complete chemical tests upon an installation of a Winterthur engine served by a Dowson pressure plant. Samples of gas were taken at different heights from the bed of fuel within the generator. The height of the fuel bed reached 17·7 inches at the beginning of the test and 18·5 inches at the end. The formation of Dowson gas commenced after 9·8 inches depth of fuel had been made and afterwards remained constant. The ratio of non-saturated hydro-carbons (C_nH_{2n}) increased with the height of the fuel bed, because, in the upper layers dry distillation of the fresh fuel was produced with formation of lighting gas.

One important fact is that, above 10 inches of fire, the Dowson gas remained of a practically constant composition. The results of the volumetric analyses of the unpurified gas were as follows :—

Height above Firebars.	Inches.	4	10	14	18	20
Carbon dioxide . . .	CO_2	1·78	6·85	8·51	8·53	8·05
Non-saturated hydro-carbons	C_nH_{2n}	0·15	0·15	0·30	0·48	0·63
Oxygen	O	18·49	0·49	0·31	0·26	0·18
Carbon monoxide . .	CO	—	26·07	20·80	20·59	23·11
Hydrogen . . .	H_2	0·66	17·0	22·05	19·22	16·85
Methane	CH_4	0·05	0·45	1·29	0·43	—
Nitrogen	N	79·37	48·98	46·74	50·47	51·18
Calorific value (B.Th.U. per cubic foot)		—	140	152	138	139

In the course of a test made in 1906 by Messrs. Hubert and Witz at the Société Cockerill on a 1,400 H.P. engine fed with blast furnace gas, a number of calorimetric analyses were made both by means of a bomb and by a Junkers calorimeter.

It was observed that the (higher) calorific value measured by the bomb (about 100 B.Th.U. per cubic foot) was from 10 to 15 per cent. higher than that determined by the Junkers calorimeter. This power was in close agreement with that obtained from calculations based upon the gas analysis.

The calorific value of a gas can be deduced from the analysis by the following formula :—

$$30{\cdot}5\ CO + 25{\cdot}7\ H_2 + 85{\cdot}1\ CH_4 = \text{Calories per cubic metre;}$$

or $$3{\cdot}42\ CO + 2{\cdot}99\ H_2 + 9{\cdot}55\ CH_4 = \text{B.Th.U. per cubic foot.}$$

Calorimeter.—The calorimetric analysis can be readily made on site by means of a Junkers calorimeter, which has been described, and its method of application explained in many other works.

Locality.	Experimenter.	Date.	Calorific Value. High. B. Th. U. per cubic foot.	Calorific Value. Low. B. Th. U. per cubic foot.
Antwerp . . .	Mathot	1902	662	540
Birmingham . . .	Witz	April, 1902	605	
,, ...	,,	,, 1902	610	
,, ...	,,	,, 1902	617	
,, . . .	Mathot	Sept., 1905	610	
Brussels . . .	,,	March, 1902	630	565
,, ...	,,	Jan., 1903	640	
,, ...	,,	Aug., 1903	595	
,, ...	,,	June, 1904	587	538
,, (Cureghem) .	,,	April, 1904	595	538
,, (Koekelberg) .	,,	Nov., 1903	564	530
,, (Molembeck) .	,,	Jan., 1903	590	
,, ...	,,	Nov., 1903	571	525
,, (St. Josse Tennode)	,,	,, 1902	650	580
Berlin	,,	,, 1901	597	
Budapest . . .	,,	,, 1902	590	
Ghent	,,	Oct., 1904	560	500
Leeds	Grover	——	——	570
Lille	Witz	June, 1900	582	
London	Mathot	,, 1901	680	
Manchester . . .	,,	,, 1906	646	
Mannheim . . .	,,	,, 1906	585	
Mons	Tricot	Dec., 1907	553	
Namur	Mathot	Oct., 1903	643	
New York . . .	,,	,, 1904	703	(water gas)
Ostend . . .	,,	Sept., 1906	590	
Pittsburg . . .	,,	,, 1904	842	(natural gas)
Roubaix . . .	Witz	Jan., 1903	610	
,, ...	,,	,, 1903	650	
Solre-le-Château . .	,,	April, 1889	671	
,, . . .	,,	Feb., 1895	643	
,, . . .	,,	March, 1895	627	
Vol-Saint-Lambert .	Mathot	June, 1903	527	491
Winterthur . . .	,,	,, 1902	595	

When the engine is supplied with gas under pressure, the calorimeter can be fed continuously and, after the working conditions have once been established the movements of the thermometer placed in the water outlet show at once any variation in the heat value of the gas.

When using town gas it is necessary to thoroughly purge the pipes of any gas left from previous tests. The composition of gas which has been kept in the pipes for any length of time becomes modified and stale and its heat value is appreciably diminished.

The list on p. 447 gives some calorimetric observation made by the author or by other experimenters.

When the engine is fed from a suction gas producer, in order to make a test it is necessary to store a certain quantity of gas. For this purpose a bell is arranged over a tank of water of about $1\frac{1}{2}$ or 2 cubic feet capacity, and by connecting this to the gas pipe as near as possible to the outlet from the scrubber the gas is drawn into the bell and afterwards passed, under the pressure due to the weight of the bell itself, into the calorimeter. In practice about half a cubic foot of gas is absorbed in taking note of the heat value, once the working conditions have been established. The water in the tank should be salted or covered with a film of oil to diminish its power of dissolution of the gas. Another method of feeding the calorimeter is by means of a suction pump in the gas pipe.

Quantity of Gas Consumed.—If the gas is under pressure it can be measured either by a meter or by noting the position or displacement of the dome or bell of a gas holder. The precautions mentioned under the heading of "supply" should be observed and records taken of the temperature and pressure of the gas.

When the gas is produced by suction, it is difficult to use any method of measurement and therefore the calorific value and chemical analysis only is noted.

Calorific Value.—With respect to town gas and the other gases containing a strong proportion of hydrogen or hydro-carbons it is important to determine which calorific standard is to be used as a base, that is whether it should be of the "higher" or "lower" standard.

The lower value is equal to the higher value less the latent heat of vaporisation of the water produced by the combustion of the gas. This water can be recovered and measured upon leaving the calorimeter by the tube arranged for the purpose. Note is taken of the quantity of gas burnt to produce the amount of water thus recovered.

For calculating the quantity of heat in calories corresponding to the condensation, Regnault's formula should be used:—

$$606{\cdot}5 + 0{\cdot}305\, t - t = Ca$$

in which t = temperature of water at the outlet of calorimeter in degree C.; $Ca.$ = number of calories for one kilogramme of condensed water.

As indicated in the preceding table the higher values range between 560 and 600 B.Th.U. per cubic foot, and the lower value between 500 and 550.

The maker who gives a guarantee of consumption for a gas of 560 B.Th.U. heat value without any qualification would have an interest in stating that he referred to the "lower" value, whereas the purchaser might hold that he meant "higher" value. If either "effective" heat value or "net" value had been used as a qualification it would clearly apply to the lower value, but, in the absence of any definite specification the author is of the opinion that the guaranteed figure should apply to the "higher" value.

Analyses.—If the determination of the nature and richness of the gas has any serious influence on the consequences of the experiment it is prudent, for checking purposes, to take some samples in a number of flasks for laboratory analysis. It is necessary to take the greatest care to avoid any risk of air entering the vessels, which would entirely falsify the results.

Analyses of power gases can be made by means of the Orsat type of apparatus. Although this is somewhat delicate to manipulate and the absorption of gas by the reagents is somewhat slow and often incomplete, nevertheless by careful operation the results obtained are sufficiently exact to note the working of the gas producer.

Solid Fuel.—A chemical analysis and determination of calorific value of the fuel supplied to a gas producer should be made, whether ordinary coal, anthracite, coke, lignite, peat, or any other kind of low-grade fuel. The experiments should be made upon an average sample, and to secure the latter about 2 or 3 lbs. of coal should be taken haphazard from each fresh charge, and at the end of the test, the different samples should be thoroughly well mixed and a selection made from the whole bulk.

The important constituents to be determined in percentage of volume are as follows: Moisture, H_2O; Carbon, C; Hydrogen, H; Oxygen, O; Sulphur, S; Ash, A.

The calorific power can be calculated approximately from the formula:—

Calories per kilogramme:

$$81\ C + 290\ (H - (\tfrac{1}{8}\ O)\ ) + 255\ S - 6\ H_2O\ ;$$

or, B.Th.U. per lb.:

$$146\ C + 522\ (H - (\tfrac{1}{8}\ O)\ ) + 459\ S - 10{\cdot}8\ H_2O.$$

It is necessary to decide whether the measurement of the fuel consumed should include everything from "cold" or whether from a "flying start."

Measurement from "cold" or from first starting to work should be taken with the apparatus absolutely empty, noting the weight of all kinds of fuel—coal, wood, &c.—that is successively added. At the end of the experiment, the apparatus is again completely emptied and the weight of unconsumed fuel, as well as of cinders or ash, determined, the difference showing exactly the net consumption.

It is necessary to take notice of the time occupied from lighting the fire to the production of gas, and to the setting the engine to work under no load, and also of the successive increases of load until the normal output is reached.

If the observations are to be taken from a "flying start" it is necessary to note the moment when the load upon the engine is reached and has been carried for some time to obtain uniform temperature conditions, &c.; and, after first carefully poking the fire, cleaning the grate, and emptying the ashpit, the generator should be filled to a determined level.

In practice owing to the usual construction of generators it is impossible to observe the level of the fuel inside the apparatus, and, therefore, the generator must be fully charged until no more fuel can be made to pass through the charging valve.

The quantity of fuel added during the test is noted, and, after stopping the engine, the fire should be poked to provoke the settlement of fuel, the grate again cleared, and the ashes removed. The generator should then be charged with a further supply of fuel to reach the same level as at the start, the weight, of course, being noted.

A fuel consumption test should be continued for at least eight or ten hours, otherwise the causes of error, due to refilling, swelling of the fuel within the generator, and the impossibility of obtaining identical temperature conditions at the beginning and end of the trial, &c., would have a disproportionate influence upon the results.

It frequently happens that during the clearing of the grate (which should be effected at regular intervals of three to four hours) some partially burnt fuel falls from the fire with the clinkers. This should be sifted and put back into the generator during the period of test, so that the correctness of the results may be unaffected practically.

The feeding and clearing should be systematically effected at regular intervals to avoid all cause of disturbances during work.

If the gas producer is of the pressure type, with a separate boiler, the fuel consumption of this should be measured separately from that of the generator.

Lubricating Oil.—The oil consumption is very readily noted by completely filling all lubricators at the commencement of the run, and measuring the quantity added to make up the amount used during the trial.

Water.—The consumption of water used for cooling the cylinder and other details of the engine should be determined separately from the amount supplied to the generator.

If the cooling of the engine cylinder is effected on the thermo-syphon principle or cooling towers the amount used is very small, corresponding merely to the waste due to leakages, evaporation, or splashing. There is, therefore, no practical interest in noting the consumption, but it is useful to measure it so as to determine the amount of heat absorbed by the cooling water.

With cooling by towers and pump, and by running the water to waste, the outlet in both is usually visible, and the quantity can be readily measured by permitting the water to flow into a receiving tank for a definite period, and weighing it. A water meter can also be arranged on the circulating pipe.

The precautions necessary for accurately measuring the water consumption have been fully dealt with in the recommendations made by the American Society of Engineers with reference to steam engine and boiler trials, and it is unnecessary to repeat these in the present volume.

With respect to the water used in connection with the gas producer plant, separate measurements should be taken for each service, viz., to the vaporiser, and scrubber or washing apparatus.

It is necessary to notice whether the overflow from the vaporiser is taken to the ashpit, and from thence to the drain, or to the drain direct. In any case the waste should be arranged so that its volume can be measured and deducted from the quantity fed to the vaporiser. Care is necessary to make certain that the same level of water is obtained, both at the beginning and end of the test, in the ashpit as well as in the vaporiser.

7. Inspection and Analyses of Residues.

The term "residues" includes the ash from the gas generator and the exhaust gases from the engine.

The ashes should be examined after the test, and withdrawn from the ashpit, which, at the commencement, was devoid of contents. The weight should be taken, but it is impossible to base any absolute conclusion upon the weight merely.

The weight of the pieces of unburnt coal that may be present in the ashes is deducted from the consumption, but this should not include the dust that may be deposited in the connections. It should be noted, however, that the loss of coal through the grate and retained by the ashpit shows a defect in the construction of the producer.

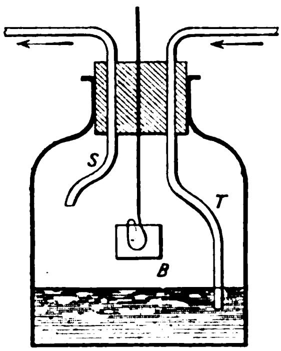


Fig. XIV.—29. Experiment to determine proportion of Carbon Monoxide in Exhaust Gases.

The analysis of exhaust gas is desirable to take into account the working conditions within the engine cylinder, but the requirements for obtaining the necessary samples entail special preparations that are difficult to make in connection with industrial installations.

Imperfect combustion, which causes excessive gas consumption in spite of apparently good valve-setting, is usually due to the lack of homogeneity of the mixture, or an insufficient supply of air to ensure perfect combustion. In the first case the fault is owing to the defective form and arrangement of the valves and combustion chamber and is beyond remedy, being entirely a matter of construction. However, such defects are, to-day, happily rare, thanks to the experience acquired by the largest firms of engine makers.

A faulty mixture can be remedied by the proper adjustments if the exhaust gas analysis reveals the presence of unburnt fuel, such as carbon monoxide. A proportion of more than 1 per cent. of this gas is a sufficient proof that the amount of air in the mixture is not enough for the conversion of the combustible constituents into carbon dioxide.

A practical test of the products of combustion can be made in a very simple manner. A 4-oz. glass bottle (Fig. XIV.—29) is obtained, into which mercury is poured to a depth of about 1 inch. A tube *T* is passed through the cork into the mercury, the other extremity being in communication with the exhaust pipe. A second tube *S* connects the interior of the bottle with the atmosphere. A small sheet of blotting paper, previously soaked in a reagent, is suspended inside the bottle by a piece of wire passed through the cork. The reagent is formed by a solution of equal parts of double chloride of palladium and sodium sufficient to obtain a clear brown liquid. The blotting paper is soaked several times in this solution and allowed to dry between each dipping until the white paper turns brown. The size of the pieces inserted in the bottle should be about ½ inch square.

When connected to the exhaust pipe, the pressure of the gases will force a certain quantity into the bottle at each explosion, the mercury acting as a retaining valve. If the gases contain more than 1 per cent. of carbon monoxide the paper will turn from brown to grey. This is a sign that the amount of air admitted to the cylinder should be increased to ensure perfect combustion.

By means of precise analyses taken from a Winterthur engine served with Dowson gas, Professor Treadwell, of Zurich, obtained the following results from the exhaust gases:—

At no load, 1,000 cubic feet of exhaust gas contained 2·65 cubic feet of carbon monoxide;

At half and full loads, the composition of the exhaust gas was as follows:—

—	—	Half load.	Full load.
		Per cent.	Per cent.
Carbon dioxide	CO_2	9·87	10·12
Oxygen	O	8·77	8·99
Carbon monoxide	CO	0·070	0·05
Methane	CH_4	0·006	0·005
Hydrogen	H	0·013	0·005
Nitrogen	N	81·271	80·830

Thus at full and half loads the combustion was nearly perfect.

8. Working Conditions.

The preceding paragraphs have been concerned with the different operations that are included in a rational gas engine test. It now remains to consider the details as a whole, and to determine which is the most efficient way to examine and co-ordinate the results in order to correctly appreciate the respective influences of the phenomena which are simultaneously produced.

The first rule to observe is that a sufficient number of assistants should be available for each to make the necessary observations without haste or flurry.

The different experimenters should be provided with exact chronometers, which should all be set in agreement at the beginning of the trials.

Each observation should bear a precise indication of the instant at which it was made.

The different operations should be repeated, as frequently as possible, at regular intervals, the length of these intervals being determined to suit the circumstances.

Every incident occurring should be timed by all those who observe their occurrence. The individual records are most useful when the results are being interpreted.

Each assistant should carefully supervise the working of a certain number of details, so as to avoid as much as is possible all chance of an accident which might otherwise bring about a stoppage and thus prejudice the correctness of the observations made. Accidents of this nature are frequent, and it is only on rare occasions that a full load trial can be carried out for eight or ten hours without being forced to interrupt the working of the engine more or less for some moments from one cause or another. In such cases it is imperative to take particular note of the causes of the interruptions and of their duration, and to note the phenomena which accompany them.

The following is a list of observations which it is useful to make during the tests:—

Engine :—Variations of speed; working of the governor; state of bearings and of other frictional surfaces as regards temperature; irregular ignition, early, late, or premature firing; modification of control.

Gas :—Variations of pressure, of temperature, of richness, and of chemical composition.

Generator :—Periods of charging; amount and intervals of the charges and of clearing; influence of these operations upon the quality of the gas and upon the working of the engine.

Site :—Temperature, atmospheric pressure, state of the weather, cleanliness.

By following the instructions given above, the results deduced from the observations made will be correct with a sufficient approximation to satisfy those interested either with regard to the verification of contract specifications or scientific reports.

9. Interpretation of Results.

Having collected the figures relating to the different observations made, in chronological order, the principal results may be calculated concerning: — Indicated H.P.; Brake H.P.; consumptions; and efficiencies.

If any load variations occurred, the average power is determined and account taken of the duration of the respective periods of work.

When the trial has been carried out with all the necessary care, the calculations made from the collected figures nearly always conduce to normal results. If otherwise, and if the results deviate to any great extent from those usually obtained in connection with similar installations it is necessary to suppose either an error in the experiment, or the existence of some peculiar phenomena that have had a marked effect upon the trial.

In the case where such phenomena are not considered sufficient to explain the consequences it is wise not to accept the figures as correct until a new trial has confirmed the first.

Amongst those circumstances that might appreciably change the results of a trial, reference may be made to the influence of atmospheric conditions of temperature and pressure. With town gas engines especially, a low temperature and high barometric pressure perceptibly improves the working and gives a reduced consumption, whilst the same engine will produce less work with a higher consumption, if the atmospheric temperature be high and the pressure low. The differences between the figures taken in the two extreme cases may reach 10 or 15 per cent.

With respect to altitude it is usually calculated that the power of an engine diminishes at the rate of about 3 per cent. for every 1,000 feet elevation above sea level.

When contract specifications are to be verified the experimenter should take note of all the circumstances in such a manner that the consequences of external influences are not attributed to the engine itself.

The figures relating to the brake power and to the fuel consumption constitute the "practical" elements of a test whilst the major portion of the other figures are of a "scientific" character. Amongst the latter are:—The indicated power; compression pressure; initial explosion and mean pressures; the resistances; temperatures of water and of exhaust gases; and the mechanical and thermal efficiencies.

Efficiencies.—As already mentioned, in the paragraph relating to the calculation of the indicated power, the *mechanical efficiency* of an engine is the ratio between the effective power and the corresponding indicated power.

The mechanical equivalent of heat is expressed as:—

$$778 \text{ foot lbs. of work} = 1 \text{ B.Th.U.}$$

or

$$427 \text{ kilogrammetres} = 1 \text{ calorie.}$$

Thus, according to British standards—

$$\text{One H.P.} = \frac{33000}{778} = 42{\cdot}41 \text{ B.Th.U. per minute (or 2545 per hour).}$$

Having obtained, from the figures noted during the trial, the number of B.Th.U. represented by the fuel consumed per effective (or indicated) H.P. hour, C, the *thermal efficiency*, is given by the ratio:—

$$\frac{2545}{C}.$$

The efficiency of a gas producer is the ratio between the total number of B.Th.U. contained in the gas which is produced during a certain time and that of the combustible consumed during the same time.

In both cases the B.Th.U. should be qualified as being either of "higher" or "lower" value.

The efficiency of the producer should be preferably expressed in terms of the raw fuel and not of the "net" (ashes and moisture deducted), because the latter does not accord with practical results.

The results obtained should be presented in such a way that the reader is able to readily grasp the most important conclusions. For this purpose it is suggested that the figures should be collected and grouped in the form of a table, the different circumstances of the operations being referred to in the text of an accompanying detailed report.

The American Society of Mechanical Engineers have proposed a form of table for gas and oil engines tests, which possess the incontestable merit of being excessively complete. But in the author's opinion, it is somewhat diffuse and does not bring out the primary observations, which, in all cases, are of the greatest interest, while it includes some measurements which cannot be verified without completely dismantling the engine, an operation which cannot be undertaken outside a technical college or private experimental shop.

Having been called upon to make a very great number of tests, the author has been forced to give this matter very close study, so as to present the figures summarised in a rational manner. The style he has adopted and recommends is reproduced in the following pages as conforming to the desiderata mentioned.

Effective Power.—In connection with gas engines it is customary to adopt the effective power, or B.H.P. as a base of comparison, whilst with steam engines the indicated power is used. There is no doubt that it is advisable to retain the custom and record all calculations of efficiency in terms of the effective power, at all events, for gas engines of small and moderate outputs.

In the case of large gas engines, the indicated power must be taken into consideration owing to the difficulties involved in measuring the effective work, but it should not be forgotten that the indicated work is liable to be affected from numerous sources of errors.

Influence of the Compression.—Amongst the details of a scientific character that the trials should make clear is the amount of compression given to the mixture and its consequences with regard to the consumption and operation of the engine.

The theory which has been propounded from the tests conducted by Dugald Clerk, Atkinson, Diesel, Banki, Burstall, Allaire, and other learned experimentalists, enables the following principle to be accepted as an axiom: Efficiency increases and consumption decreases with the increase of compression pressures.

Practice, however, imposes a limit to the increase of compression which it is unwise to overstep, and it is necessary to seek means to avoid pre-ignitions, which are the direct consequence of high compression, by impoverishing the mixture, by lowering the gas temperatures, by water injection, or by energetic cooling of the cylinder and combustion chamber.

No. of test.	Type of engine.	B.H.P.	Pressure, lbs. per square inch.		B.Th.U. per B.H.P. per hour.
			Compression.	Initial pressure.	
4	Cockerill	215	152	237	12,600 blast furnace gas
23	Cie. Berlin Anhalt	64·5	120	242	11,700 producer ,,
36	Soest	54·5	108	282	10,080 Dowson ,,
44	Tangye	66·5	71	284	10,370 town ,,
56	Deutz	35·5	138	314	8,140 ,, ,,
59	Niel	46·0	170	—	9,900 ,, ,,
77	Schmitz	28·8	166·5	363	10,000 ,, ,,
124	Catteau	17·5	185	—	8,500 ,, ,,
15	Winterthur	104·8	165	276	12,400 anthracite
31	Benz	77·3	192	412	13,200 ,,
41	Deutz	65·1	170	385	10,800 ,,
74	Winterthur	31·2	114	370	9,890 ,,
79	Winterthur	31·2	152	292	12,700 ,,
114	Diesel	19·9	—	—	10,800 petroleum

According to the arrangements adopted to secure high compression, the results with respect to consumption are more or less favourable, and it is impossible to discern the effects of high compression solely, from the published results of gas engine tests, for this reason.

From the test results given elsewhere the table on p. 457 has been compiled showing the relative heat consumption, with particulars of compression and initial explosion pressures.

From the figures it will be seen that no very precise deductions are possible with regard to the influence of compression pressures, but as gas engines are now made it does not seem to be absolutely indispensable to use high compression—that is to say exceeding 140 to 170 lbs. per square inch—in order to obtain what must be deemed a satisfactory consumption. The trials that the author has made particularly with a view to study the consequences of raising the compression, more or less, has necessarily resulted in great gaps showing no direct relation. It is therefore useful that experimenters should carefully take observations bearing upon the practical determination of what the influence of the compression may be.

Summary of Results of a Gas Engine Test.

I.—Description.

1. Test made by
2. On an engine installed at
3. Owner's name
4. To determine
5. Circumstances of the trial
6. Prevailing temperature
7. Atmospheric pressure
8. Date of test
9. Type of engine and serial number
10. Method of ignition
11. Class of engine
12. Name of maker

II.—Principal Data.

A.—*Engine.*

13. Diameter of piston $D =$ inches
14. Stroke ,, $L =$ inches
15. Diameter of front piston rod = inches
16. ,, back ,, = inches
17. Effective area of piston $A =$ square inches
18. Number of impulses per revolution $K =$
19. ,, ,, revolutions per minute $N =$
20. Normal B.H.P. =
21. Modulus of absolute power $\dfrac{A \times L}{792000} =$

22. Modulus of speed $\frac{K\ N}{B.h.p.} =$

B.—*Producer.*

23. Type
24. Purifiers and washers

	No load.	Constant load.	Maximum load.
III.—Work.			
25. Duration of test . . . . hrs. mins.			
26. Effective power in H.P. . . . . . B.H.P.			
27. Revolutions per minute — maximum . . .			
minimum . . .			
average . . .			
C.—*Diagrams.*			
28. Average compression . . . lbs. per sq. in.			
29. Average initial explosion pressure . . .			
30. ,, final expansion ,, . . .			
31. ,, of mean ,, . . .			
32. Mean negative pressure during suction . .			
33. Indicated H.P. . . . . *Pi*			
IV.—Fuel.—This section should be in duplicate for an installation of engine and producer, observations then being taken of the coal served to the generator and of the gas supplied to the engine.			
Physical and Chemical Properties.			
34. Nature			
35. Source of supply			
36. Dimensions			
37. Chemical composition			
38. Calorific value, higher or lower, reduced to standard of 32° F. and 29·92 inches Bar . . .			
39. Hourly consumption, gross			
40. ,, ,, net			
41. B.Th.U. consumed per hour			
V.—Water.—This section should be repeated for each separately water-cooled part.			
Engine.			
42. Average temperature of cooling water at inlet .			
43. ,, ,, ,, ,, outlet .			
44. Quantity of water used per hour			
45. Number of corresponding B.Th.U. . . .			
Producer.			
46. Quantity of water used in vaporiser per hour .			
47. Average temperature of vaporisation . . .			
48. Quantity used for washing gas per hour .			
49. Average inlet temperature to both apparatus .			
50. ,, outlet ,, from ,, ,, .			

	No load.	Constant load.	Maximum load.
VI.—Gas.			
51. Temperature at generator outlet			
52. ,, entry of scrubber			
53. ,, outlet ,,			
54. ,, inlet to engine			
55. Ratio gas to air in mixture per cent.			
56. Analysis of exhaust gas			
57. Hourly quantity of exhaust gas			
58. Number of corresponding B.Th.U.			
VII.—Efficiencies and Consumption.			
59. Ratio $\frac{\text{Brake H.P.}}{\text{Indicated H.P.}}$			
60. Fuel consumption per brake H.P. hour			
61. Lubricating oil ,, ,, ,, ,, ,,			
62. Cooling water ,, ,, ,, ,, ,,			
63. Vaporising ,, ,, ,, ,, ,,			
64. Washing ,, ,, ,, ,, ,,			
65. Volume of exhaust gas ,, ,, ,,			
66. B.Th.U. in ,, ,, ,, ,, ,,			
VIII.—Heat Balance Sheet.			
67. B.Th.U. supplied per brake H.P. hour			
68. Percentage absorbed in useful work			
69. ,, ,, ,, mechanical friction			
70. ,, lost in cooling water			
71. ,, ,, ,, exhaust and radiation			
72. ,, ,, ,, incomplete combustion			

The following are the chief points to be observed in order that the tests may be readily and impartially conducted :—

1. All tests to verify the clauses of a contract specification should only be made after the interested parties have been invited to assist in the presence of each other.

2. The experimenter should authorise the constructor to check and, if necessary, to adjust the installation before proceeding.

3. The experimenter should take into account the influence of climatic conditions upon the results.

4. When a guarantee has been given with respect to the consumption of gas of a stipulated calorific value, with no further qualification, the figures should be prepared upon the "higher" heat value.

5. Unless there is any stipulation to the contrary in writing, the makers' guarantees should be deemed to be fulfilled if the figures do

not vary more than 5 per cent. from the specified terms or quantities. This latitude should be considered as an allowance on account of the "coefficient of ability" on the part of the experimenter.

6. For gas engines of small and moderate powers the figures for consumption and efficiency should be preferably computed upon the "brake H.P." basis.

CHAPTER XV

INDICATOR DIAGRAMS AND EXPLOSION RECORDS: RESULTS OF TRIALS

The "hit-and-miss" system of governing has enabled some very valuable information to be gained, explanatory of some of the

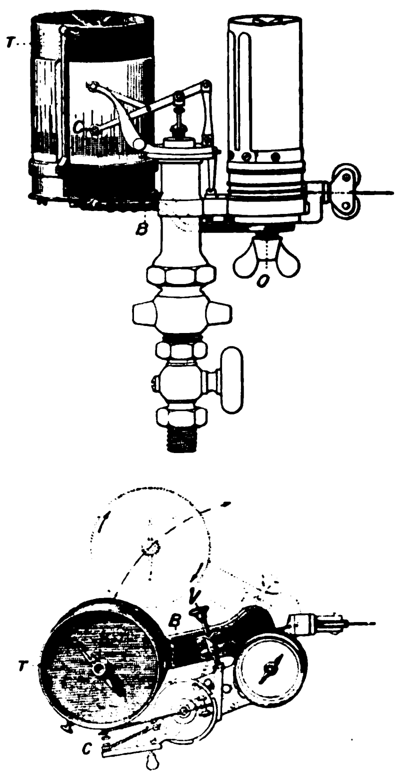


FIG. XV.—1. Mathot Explosion Recorder.

phenomena connected with the combustion of explosive mixtures in gas engine cylinders. Such information is of the greatest service both to the practical engineer and those engaged in research work, but in order to properly appreciate the importance of the phenomena, not only with regard to the pressure variations within the cylinders during a complete working cycle, but also in a series of consecutive cycles, a special form of apparatus is advisable and indeed essential.

For this purpose the author has designed an "explosion recorder" consisting of a paper drum rotated by clockwork upon a separate spindle, the whole capable of attachment to any type of pencil indicator, as illustrated in Fig. XV.—1. Thus superimposed diagrams can be obtained from the same instrument in relation to positions of the engine piston throughout one or more cycles in the ordinary manner, and also a sequence of cycles can be recorded upon a long strip of paper, the characteristic features of each being thereby presented in a form which permits the elucidation of many problems.

The diagrams thus produced represent the succession of explosions and intermission of cut-out strokes, as well as the proportion of one to the other according to the action of the governor at varying loads.

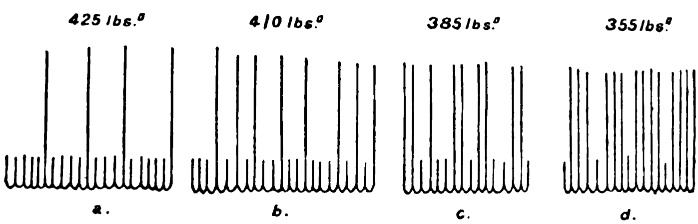


FIG. XV.—2. Explosion Records of Varying Loads.

By proper interpretation, the phenomena thus graphically recorded can be recognised as being the consequence of faulty construction, defective governing or unsatisfactory erection.

The record reproduced in Fig. XV.—2 was taken from a modern 30 H.P. Tangye gas engine. The longer lines represent initial explosion pressures and the shorter compression pressures of air only during cut-out cycles. In the section marked *a* the engine was unloaded and the extreme sensibility of the governor is proved by the regular occurrence of gas admissions and consequent explosions after every four or five cut-outs. It will be seen that the average initial explosion pressure in this section *a* reached about 425 lbs. per square inch, but fell successively to 410, 385, and 355 lbs. per square inch in the sections marked *b*, *c*, and *d*, which refer respectively to loads of 18, 22, and 30 H.P.

The gradual diminution of the initial pressures in proportion to the increase of load appears to have some relation to the corresponding increase of cylinder temperature. In fact, the reduction in weight of mixture admitted is proportionate to the increase in temperature. Apart from the necessity of adjusting the water circulation to suit the load, it appears advisable, therefore, to protect the inlet valve as much as possible from excessive heating.

The graphic record reproduced in Fig. XV.—3 supports this conclusion and shows that the compression pressure itself is affected by the heating of the pure air admitted during cut-out strokes. The record

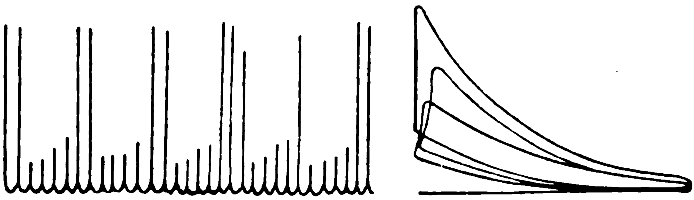


FIG. XV.—3. Explosion Record, restricted water circulation.

was taken from an American gas engine, the inlet valves and cylinder of which only regained normal temperature by the absorption of the excess of heat by the air admitted during cut-out strokes. The accompanying ordinary reciprocating diagram shows that the extreme variations occurring correspond to mean pressures of 57 to 85 lbs. per square inch.

The diagram and record in Fig. XV.—4 were taken from an engine

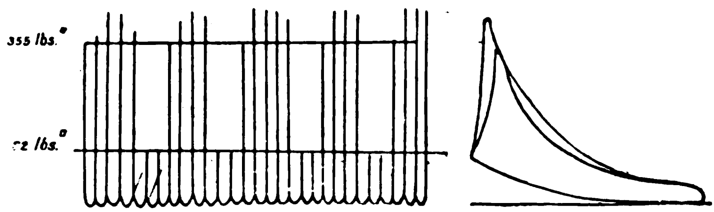


FIG. XV.—4. Explosion Record, weak mixture, after cut-out stroke.

in which the gas valve is separated from the mixture valve by a somewhat long passage (Fig. XV.—5). After each cut-out the next admission of gas and air was impoverished by the quantity of gas corresponding to the contents of the passage, as compared with the proportions taken in when several admissions of mixture followed each other. The first explosion after a cut-out is therefore weaker, and the diagram shows signs of retarded ignition.

The contrary effect is produced in other engines having the gas and mixture valves sufficiently close to prevent a dead space, but presenting, on the other hand, a combustion chamber of a shape less favourable to the proper expulsion of burnt gases, or with an insufficient area of exhaust port. In such a case, Fig. XV.—6 shows that the first explosion which follows a cut-out is stronger than the others, and

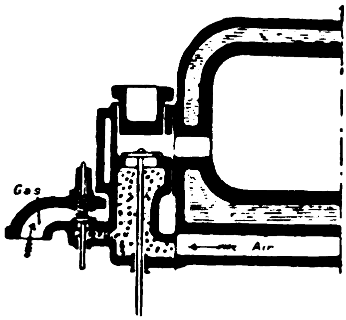


FIG. XV.—5. Improperly designed Gas Passage.

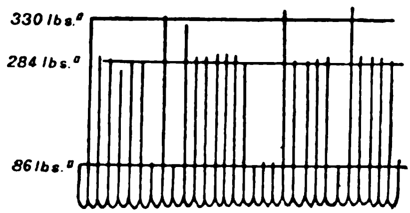


FIG. XV.—6. Explosion Record, rich mixture, after cut-out stroke.

is even more so when two or three consecutive cut-outs occur, because the amount of air thus admitted and expelled from the cylinder has swept out the products of combustion which had before been present to impoverish the mixture.

The next record (Fig. XV.—7) is somewhat similar to Fig. XV.—6, but in this case the explosion pressures gradually decrease during a

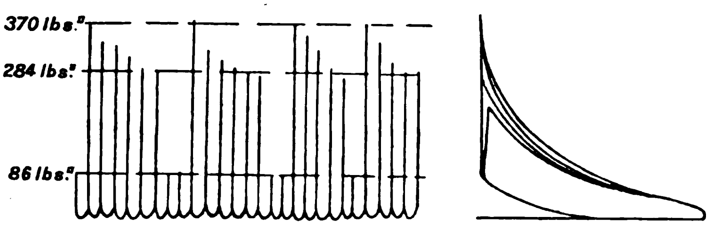


FIG. XV.—7. Explosion Record and Diagrams, effect of long Exhaust Pipe.

series of power strokes. It was taken from an engine fitted with an extremely long exhaust pipe, while the exhaust valve remained open during a large portion of the suction stroke. The result was that, during the first part of the suction of the charge, some of the exhaust gas was drawn back and the amount thus re-entering the cylinder was increased in proportion to the quantity of burnt products in the exhaust pipe. After a cut-out, the engine drew back the air discharged during the preceding stroke, and the harmful effect of this on the mixture was less than that occasioned by the burnt gases.

The observation seems to conform to the third of the rules formulated several years ago by Mr. F. Grover, of Leeds, as follows :—

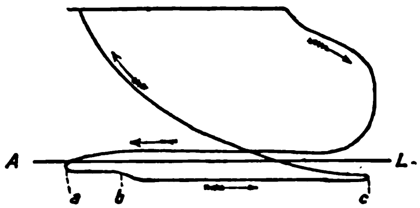


FIG. XV.—8. Re-aspiration of Burnt Products during Suction Stroke.

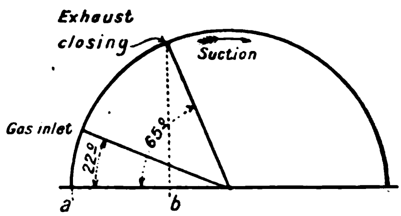


FIG. XIV.—9. Diagram of improper Valve Settings.

1. Higher pressures are attained when gas residuals take the place of an excess of air.

2. When the volume of the products of combustion do not exceed 58 per cent. of the mixture, it is explosive if the air present be not less than 5·5 times the volume of town gas.

3. The instant of explosion is greatly retarded when the excess of air is replaced by products of combustion.

In support of this theory reference may be made to a report made

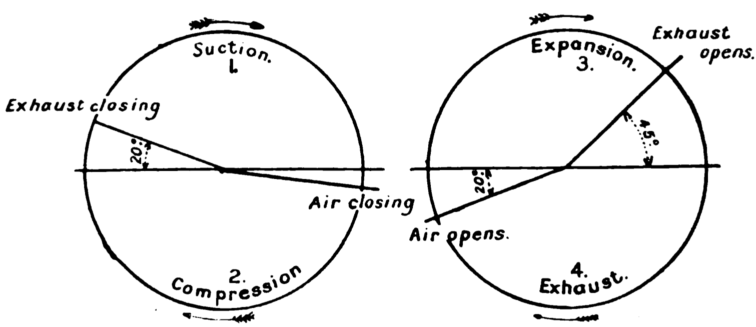


FIG. XV.—10. Diagram of Valve Setting to encourage Scavenging by Fluid Inertia.

by the author upon a 20 H.P. Dudbridge gas engine, fed with town gas, the resistance diagrams (Fig. XV.—8) taken from which plainly confirmed the consequences of imperfect valve adjustment. In this engine, the exhaust valve remained open 65° after the dead centre at the beginning of the suction stroke, whilst the inlet valve opened before the closure of the exhaust (Fig. XV.—9). In this case, there was undoubtedly re-aspiration of burnt products through the exhaust valve. The low consumption noted during the test can be explained in two ways. Either it confirms Grover's theory, or, if the theory of stratification be admitted, it is probable that the burnt gas remained close to the piston during the suction and compression strokes, forming a cushion of inert fluid, while the explosive charge filled the combustion chamber.

The portion *a*—*b* of the suction line (Fig. XV.—8), shows a minimum of depression corresponding to the re-entry of the burnt gases, whilst the admission line proper commences only at *b* and finishes at the end of the piston stroke at *c*.

Whichever explanation may be chosen to account for the economical result obtained, it partly confirms the value which the majority of English makers attach to the system of scavenging, such action being automatically induced by the air and exhaust valves being open at

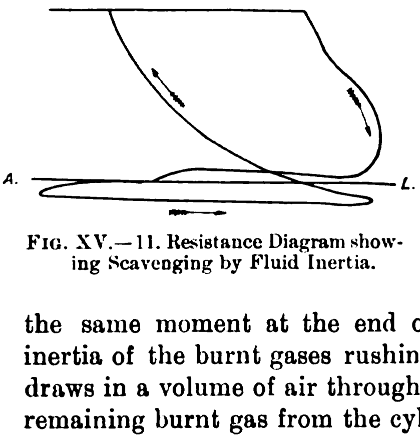


FIG. XV.—11. Resistance Diagram showing Scavenging by Fluid Inertia.

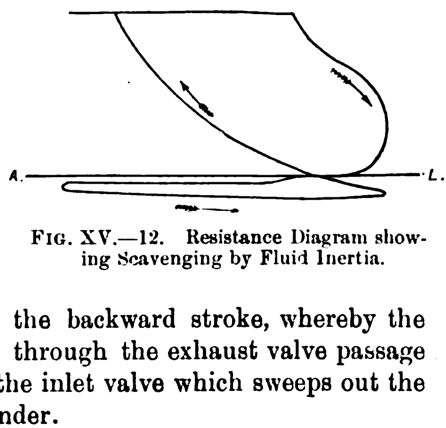


FIG. XV.—12. Resistance Diagram showing Scavenging by Fluid Inertia.

the same moment at the end of the backward stroke, whereby the inertia of the burnt gases rushing through the exhaust valve passage draws in a volume of air through the inlet valve which sweeps out the remaining burnt gas from the cylinder.

The diagram of the valve settings (Fig. XV.—10), shows this arrangement, and it will be seen that the period during which both the air and exhaust valves are open simultaneously is represented by two angles, each of 20°.

As a matter of fact, however, in nearly 450 tests of all kinds of engines made by the author in Europe and America, he has come across perhaps a score of instances in which such scavenging has really been accomplished, as shown in the resistance diagrams (Figs. XV.—11 and 12).

In Fig. XV.—11 the depression is only to be seen at the end of the backward stroke of the piston, whilst in Fig. XV.—12 it is to be observed at an earlier period of the exhaust stroke.

In Fig. XV.—13, the exhaust curve shows variations due to the undulatory movements of the gas within the exhaust box and connections, and also the partial vacuum apparent at the end of the stroke by which scavenging is induced.

Exhaust boxes, which are always fitted to gas engines, impose the

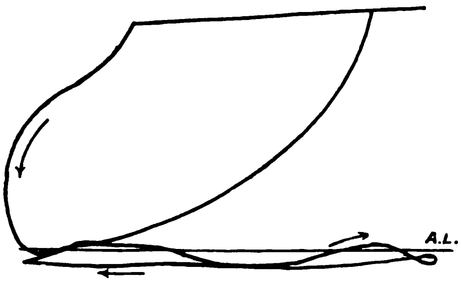


FIG. XV.—13. Resistance Diagram showing undulation of Exhaust Curve due to connection.

first obstacle to the inertia of the gas escaping from the cylinder. The length of the pipe and the bends form the second source of resistance. Scavenging "by inertia" only takes place in the exceptional cases where the exhaust pipes are short and straight, and devoid of large boxes or silencers which involve expansion, loss of energy, and therefore loss of velocity, of the exhaust gases.

The existence of "inertia" scavenging, however, cannot be denied,

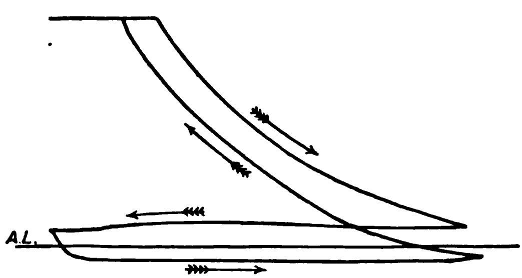


FIG. XV.—14. Resistance Diagram, air charge on Cut-out Cycles.

as it is attested by the resistance diagrams taken from engines governed by "hit-and-miss."

Fig. XV.—14 is a diagram taken during a cut-out stroke taking a charge of air only, and Fig. XV.—15 is one taken with an explosive charge. The spring employed is the same in both cases. Towards the end of the exhaust stroke it will be seen that Fig. XV.—14 shows more resistance than Fig. XV.—15. This is due to the fact that with air only, the escaping fluid has no energy, and is simply pushed out by the returning piston. In Fig. XV.—15 on the other hand, the burnt gas escapes at a pressure at release of more than 30 lbs. per

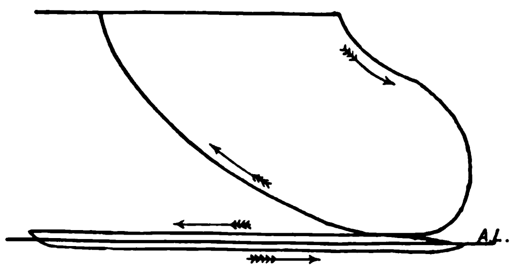


FIG. XV.—15. Resistance Diagram, Power Cycle.

square inch, and reaches the highest speed of evacuation towards the half-stroke of the piston when the exhaust valve is still wide open. Towards the end of the stroke the gases escaping in the pipe impart some of their acquired speed to the residuals within the cylinder, and even if the partial vacuum desired by the partisans of "inertia"

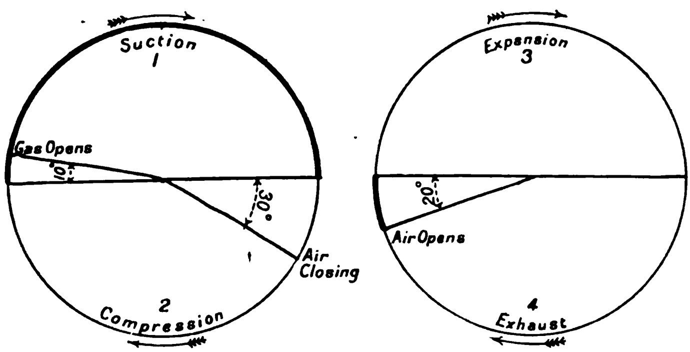
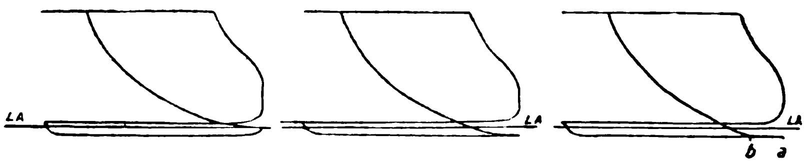


FIG. XV.—16. Diagram of Valve Setting showing "lead" and "lag" of Air Inlet Valve.

scavenging is not produced, the cylinder is, at all events, more completely emptied of heated products of combustion.

The phenomena connected with fluid inertia may also be observed during the suction stroke, and advantage is taken of the fact to more completely fill the cylinder during the period of induction.

Engines working with pressure gas or suction gas require the settings of the gas and mixture valves adjusted accordingly. In all cases, the speed of the gas entering the cylinder proceeds from the linear speed of the piston, and this is at a maximum towards the middle of the stroke, falling away to nothing at each end. But it is necessary to arrange for the valves to open and close smoothly, gradually, and in such a manner that the area for the passage of the fluid reaches a minimum at the ends of the stroke. In order that the cylinder may be completely filled during the suction stroke, therefore, it is desirable to close the inlet valve *after* the outer dead centre, so that the inertia of the incoming charge continues to fill the cylinder although the piston has commenced its backward stroke. The diagram of this valve setting is shown in Fig. XV.—16, and upon the resistance diagram (Fig. XV.—17) it corresponds to a diminution of the negative pressure at the right hand end of the suction line and a well marked variation of the compression curve at its commencement. Fig. XV.—18 shows the representative compression curve actually obtained from



FIGS. XV.—17. 18. 19.
Variations in Suction Curve due to varied setting of Inlet Valve.

cylinders in which the inlet of mixture finishes *at* the outer dead centre.

Late closing of the inlet valve does not permit the end in view, however, if the resistance to suction is strong enough to counteract the subtle effect of inertia. The proof of this is to be seen from Fig. XV.—19, which was taken from the same engine as Fig. XV.—18, but with delayed closure of the air valve, so that the compression curve, instead of starting from the beginning of the inward stroke, is curtailed, owing to the commencement being delayed by the portion of the stroke from *a* to *b*.

The gas valve should be closed before the mixture inlet valve at a point near the dead centre, so that the gas contained in the space between the gas valve and the inlet valve may be swept into the cylinder with greater certainty by the current of air.

The "hit-and-miss" system of governing also allows some thermal phenomena to be observed and controlled which are not without interest, as, for instance, the influence of high temperature, in the cylinder and its ports, upon the efficiency of the engine.

Some experiments made upon a small English engine by varying the flow of cooling water have already been alluded to on p. 179. During each experiment the author applied a brake and noted the

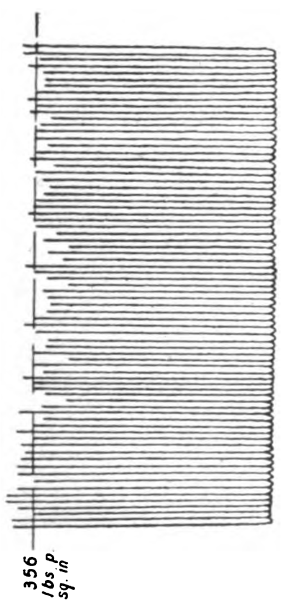


FIG. XV.—20. Explosion Record of Engine with *cold* jacket developing 10·65 B.H.P.

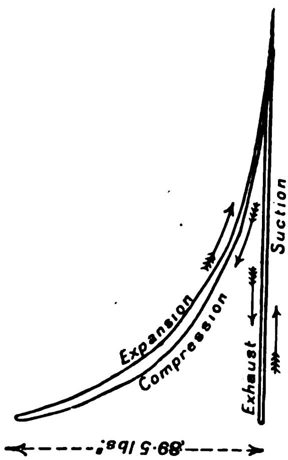


FIG. XV.—21. Compression Diagram under conditions of Fig. 20.

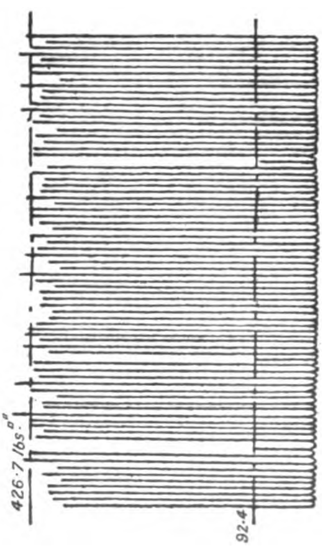


FIG. XV.—22. Explosion Record of same Engine with *hot* jacket developing 11·24 B.H.P.

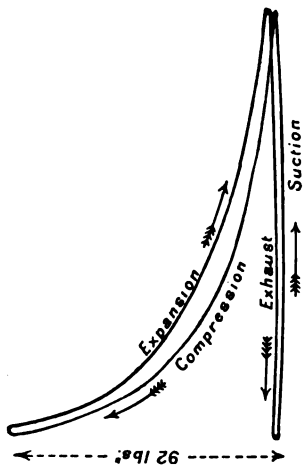


FIG. XV.—23. Compression Diagram under conditions of FIG. 22.

gas consumption, and, at the same time, took a number of indicator diagrams and explosion records. With a copious supply of water to give a well-cooled jacket, the power developed was 10·65 H.P., the engine under these conditions giving out its maximum power with no cut-outs (Fig. XV.—20). The gas consumption was 19·8 cubic feet per H.P. hour (at standard temperature and pressure—32° F. and 29·9 inches of mercury), and the compression pressure shown by the diagram (Fig. XV.—21) was 89·5 lbs. per square inch.

With moderate cooling, the water flowing from the outlet of the jacket at a temperature of about 140° F., the power developed by the engine was 11·24 H.P., and the explosion record (Fig. XV.—22) showed a margin of 2 per cent. of explosions in hand. The gas consumption fell to 18·5 cubic feet per H.P. hour, and the compression pressure reached 92 lbs. per square inch (Fig. XV.—23).

With a comparatively slight variation in the amount of water

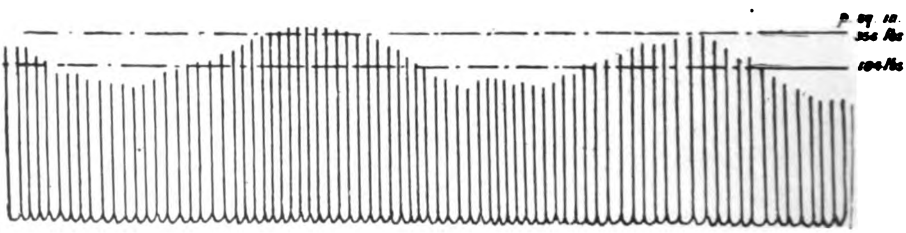


FIG. XV.—24. Explosion Record from small Engine drawing gas from main also serving a larger Engine.

circulated, it will be seen that the compression varied 4 per cent., and the corresponding power nearly 6 per cent.

The gas pressure, and that of the mixture, also has an influence that should not be overlooked. A regular and continuous supply of gas should be assured by suitable arrangements to avoid the effect of pressure fluctuations, however slight in themselves. The explosion record in Fig. XV.—24 points the moral. It was taken, during the 1900 Paris Exhibition, from a Campbell engine erected about 60 or 70 yards away from the large 600 H.P. Cockerill engine, which, by the force of circumstances, had to be served with town gas from the same main that supplied all the other engines exhibited in operation. The record taken under light load conditions shows well-marked undulations in the initial explosion pressures in synchronism with the number of charges taken by the Cockerill engine, which, being of such large dimensions, created a drop in pressure in the service main at each suction stroke.

From this it will be evident that for engines working with suction producer gas, for example, a pressure regulator should be fitted along the supply connections when several engines are served by the same generator. An engine fed from its own generator should have a reservoir or expansion chamber fitted at a suitable position to damp down pressure variations, whether increases or decreases, and

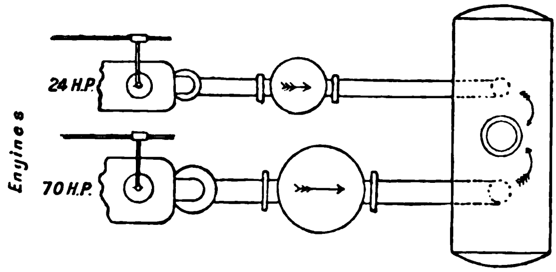


FIG. XV.—25. Engines exhausting to a Chamber common to both.

whether connected to a producer plant, blast furnace or any other form of generator.

Variations of pressure or resistance to the exhaust gases also deserve to be considered in connection with the good working of an engine. The free escape of the burnt products must be assisted by the areas and

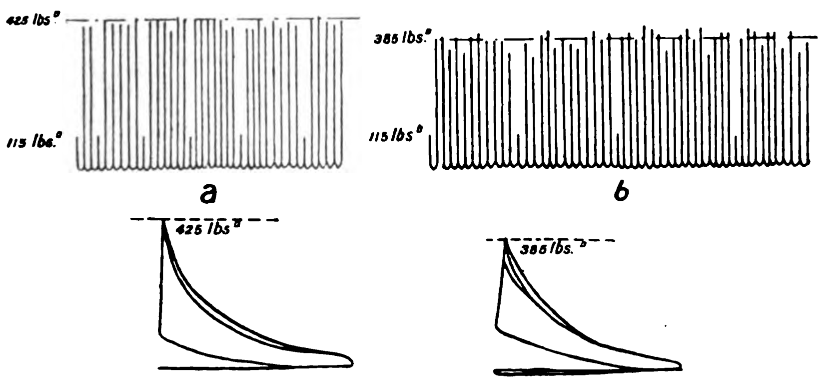


FIG. XV.—26. Records and Diagrams showing variations due to combined exhaust connections.

contours of the exhaust pipe and not be hindered by combining the exhausts from several engines in one pipe or in the same vessel.

In this connection reference may be made to two engines, one of 70 and the other of 24 H.P., the exhaust pipes of which were combined from the first exhaust silencer, as shown in Fig. XV.—25. In spite of large areas and of the general arrangement, which apparently protected the

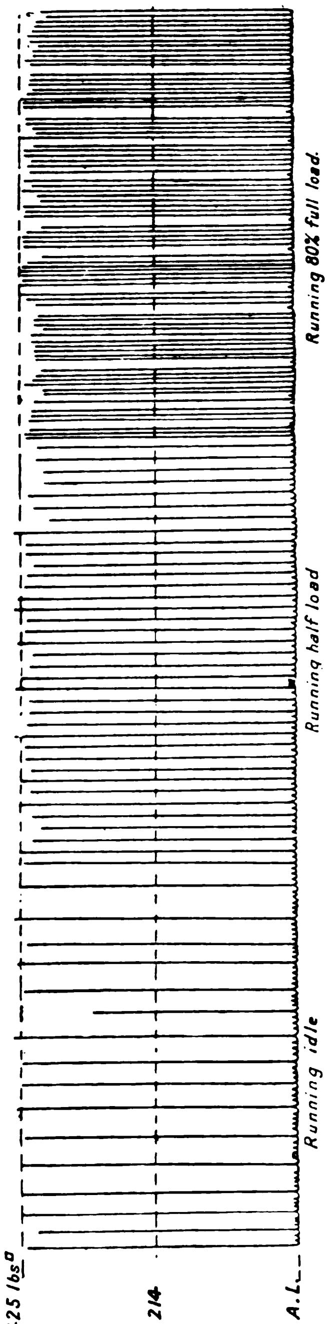


FIG. XV.—27. Explosion Record from " Banki " Benzene Engine.

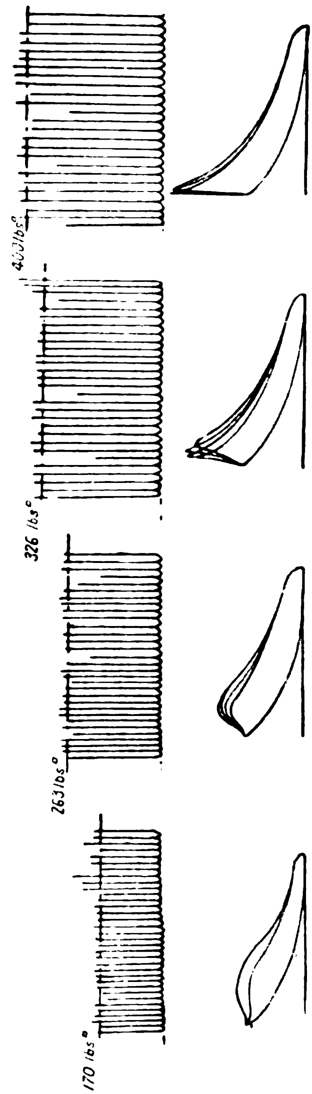


FIG. XV.—28. Records and Diagrams from Engine governed by " conical " Gas Cam.

engines from any disturbances that might arise from opposing discharges, the diagrams taken from the 70 H.P. engine and reproduced in Fig. XV.—26 are instructive. The section marked *a* corresponded to the same engine at the same load as the section *b*—between 55 and 60 H.P. — but in the former (*a*) the 70 H.P. engine worked by itself, whilst in the latter (*b*) the two engines worked together.

In the gas supply pipe which fed the two engines, the pressure remained constant and the supply was scarcely affected when the two charges were taken simultaneously. The section *a* shows great regularity of initial pressures varying between 400 and 425 lbs. per square inch, "cut-outs" occurring every 6 to 8 cycles. The section *b* shows irregularity of explosions, and the mean height has fallen to between 370 and 400 lbs. per square inch, while "cut-out" strokes occur at longer intervals to produce the same output of power. The explosions have gained in quantity what they lost in power on account of the exhaust from one opposing the exhaust from the other.

Fig. XV.—27 relates to a test made by the author on a 35 H.P. Banki engine in 1899, at the works of *Ganz & Tarsa*, at Budapest.

In support of the learned deductions already made by Professor E. Meyer from his personal experiments, the author believes it to be interesting to point out the regularity obtained by inertia governors on the "hit-and-miss" principle. The Banki engine referred to worked with spirit and was governed by the exhaust. The "cut-out" strokes, therefore, do not appear as a compression line in the diagram (Fig. XV.—27), but by a slight deflection about the atmospheric line due to the low back pressure.

The number of explosions is practically in proportion to the indicated H.P. of the engines. The initial pressures are unusually uniform, remaining constant between 400 and 425 lbs. per square inch at all loads. This is in consequence of effective scavenging of the cylinder and of the uniform temperature maintained in the walls by the water injection introduced with the benzine.

The leading dimensions of the engine were :—

Diameter of cylinder . .	250 mm. (nearly 10 inches)
Stroke of piston . . .	400 mm. (15·8 inches)
Revolutions per minute . .	214 to 217
Density of benzine at 15° C. .	0·700
Consumption at 25·7 B.H.P. .	0·51 lbs. per B.H.P. hour
19·8 ,, .	0·54 ,, ,,
13·93 ,, .	0·60 ,, ,,
6·72 ,, .	0·86 ,, ,,
light ,, .	3·125 ,, per hour.

From the preceding examples it will be seen that from the tests made upon "hit-and-miss" governed engines, an abundance of useful lessons may be learnt, but which can be deduced with much more difficulty from engines governed upon up-to-date principles.

The explosion records reproduced in Fig. XV.—28, with their corresponding diagrams, point out the imperfections of governing by a conical cam. They were taken from a 50 H.P. German engine at 15,

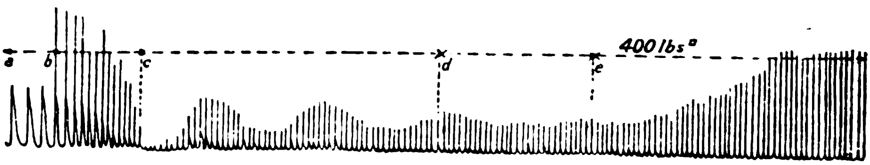


FIG. XV.—29. Explosion Record from throttle-governed Engine.

25, 35, and 50 H.P. respectively. The great irregularity in the initial explosion pressures, especially in the first three sections, are partly caused by the lack of homogeneity in the mixture and also by the lack of steadiness of the governing device. The conical cam usually produces harmful reactions on the governing mechanism.

The sensibility of the governing device adopted by the Gasmotoren Fabrik Deutz, and referred to on p. 220, is clearly shown by the

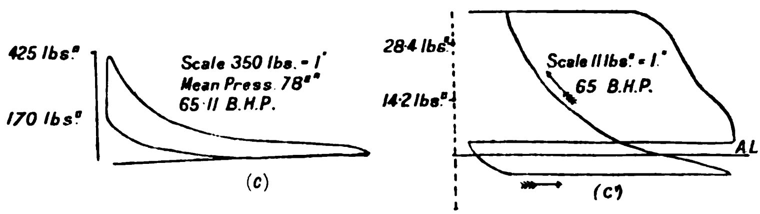


FIG. XV.—30. "Power" and "Resistance" Diagrams, full load.

explosion record (Fig. XV.—29) and the diagrams (Fig. XV.—30 and 31) taken from a "G 9" engine, from a report on which the following interesting figures are abstracted.

The explosion record represents a period of 120 seconds and is composed of five different sections, since the starting up and putting under load only occupied two minutes:—

The portion from *a* to *b* shows the period of starting up by compressed air, consisting of three admissions reaching about 140 lbs. per square inch of pressure and sufficing to start the engine and to produce the first admissions of explosive charges. The ignition of these charges create explosion pressures of from 400 to 430 lbs. per square inch.

The engine having thus exceeded its normal rate of speed, the governor comes automatically into action and gradually reduces the admission until—at *c*—the lowest point of the vertical lines descends below the atmospheric line. This demonstrates the degree of vacuum occasioned by the rarefication of the mixture admitted to the cylinder. From *c* to *d*—approximating to about 40 seconds—the governor oscillates, and finally, from *d* to *e*, takes up a position corresponding to "no load" with explosions from 115 to 140 lbs. per square inch initial pressure. This accounts for 80 seconds. From the point *e* the load is gradually applied by means of a brake until fully loaded, for which

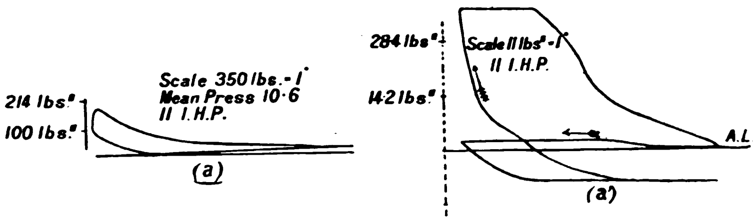


FIG. XV.—31. "Power" and "Resistance" Diagrams, light load.

condition the explosions uniformly reached the initial pressure of about 400 lbs. per square inch.

The following table gives the details of a test made by the author at Cologne, March 15th, 1904, upon a 60 H.P. Gasmotoren Fabrik Deutz gas engine, type "G 9," with suction gas:—

Diameter of cylinder	420 mm. (16·6 inches)
Stroke of piston	480 mm. (19·0 inches)

Full Load.

1. Revolutions per minute	188·66 mean
2. Brake H.P.	65·11 (metric)
3. Average compression pressure	171 lbs. per sq. in.
4. Initial explosion pressure	385 ,, ,,
5. Final expansion pressure	24·2 ,, ,,
6. Negative pressure (max.) during suction stroke	4·27 ,, ,,
7. Average of mean pressures	78·4 ,, ,,
8. Indicated H.P.	77 (metric)

Fuel.

9. Description, lean coal, 10—20 mm. (¼ to ½ inch)
10. Source of supply—Colliery at Zeche-Morsbach at Aix-la-Chapelle
11. Fuel analysis—

Carbon	83·22 per cent.
Hydrogen	3·31 ,,
Nitrogen and Oxygen	3·01 ,,
Sulphur	0·44 ,,
Ash	7·33 ,,
Moisture	2·69 ,,

12. Calorific value, gross . . . . . 13,950 B.Th.U. per lb

Gas.

13. Analysis—

Carbon dioxide	6·60 per cent.
Oxygen	0·30 ,,
Hydrogen	18·90 ,,
Methane	0·57 ,,
Carbon monoxide	24·30 ,,
Nitrogen	49·33 ,,

14. Calorific value of gas at constant volume, water vapour condensed to 15° C. and reduced to 0° C. and 760 mm. mercury=140 B.Th.U. per cubic foot.

Temperatures.

15. Cooling water, entering at 13° C. (55° F.) = 43° C. (109° F.) at outlet of combustion chamber.
16. Cooling water entering at 13° C. (55° F.) = 53° C. (127° F.) at outlet of cylinder.
17. Water in vaporiser = 84° C. (182·5° F.).

Efficiency.

18. Mechanical efficiency . . . . . . 84·6 per cent.
19. Consumption of raw coal per B.H.P. hour . 0·79 lbs.
20. Thermal efficiency per B.H.P. on gross coal supplied to generator . . . . . 24·3 per cent.

Half Load.

1. Revolutions per minute . . . . . 195·5 average
2. Corresponding effective load . . . . . 33·83 B.H.P. metric

3.	Average compression pressure . . .	121·0 lbs. per sq. in.
4.	Average initial explosion pressure . .	250·0 ,, ,,
5.	Final explosion pressure	20·0 ,, ,,
6.	Negative pressure (max.) during suction stroke	6·5 ,, ,,
7.	Average of mean pressures	44·5 ,, ,,
8.	Indicated H.P.	45·0 metric.
9.	Speed variation full load to half load . .	3·5 per cent.

Efficiency.

10.	Consumption of raw coal per B.H.P. hour .	1·15 lbs.

No Load.

1.	Revolutions per minute . . .	199·0 average.
2.	Average compression pressure . .	92·45 lbs. per square inch.
3.	Initial explosion pressure . . .	213·3 ,, ,,
4.	Final expansion pressure . . .	0·0 ,, ,,
5.	Depression during suction stroke .	8·5 ,, ,,
6.	Average of mean pressures . .	10·8 ,, ,,
7.	Indicated H.P.	11·0 metric.
8.	Speed, variation, full load to no load .	5·2 per cent.

It will be noticed that at all loads the diagrams always show the ignition at the dead centre, that is to say that the best use is made of the charges at constant ratio without necessitating any variation of ignition during the tests. The explosion pressure remained practically in proportion to the compression. Hence the remarkable low consumption at half load.

The diagram Fig. XV.—32 is from a 1,000 H.P. two-cycle Oechelhäuser engine fed with blast furnace gas. The mean pressure at 612 H.P., as shown by the diagram, is 46·8 lbs. per square inch only. The initial explosion pressure is 348 lbs. per square inch. It will be seen that the exhaust, blast of air, and inlet of charge occupies but one-eighth of the total stroke of the two pistons.

The superimposed diagrams and the tachogram reproduced in Fig. XV.—33 is from a 200 H.P. Gasmotoren Fabrik Deutz double-acting, four-cycle engine served by a suction gas producer. The indicator diagram shows the variations of compression pressures and corresponding explosion pressures. The tachometer diagram depicts the sudden changes of load between no load and maximum, causing speed variations of plus and minus 3·5 per cent. Fig. XV.—34 is a reproduction of an explosion record taken during starting up the same engine. It shows the different phases of compressed air starting, first

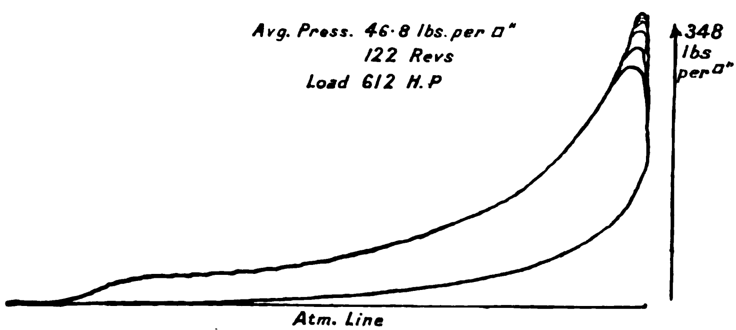


FIG. XV.—32. Diagram from 1,000 H.P. Oechelhäuser Engine, Blast Furnace Gas.

explosions with full compression, no admissions, period of governor oscillation, and normal "no load" after 90 seconds.

From the diagrams Fig. XV.—35 taken from the two ends of one of

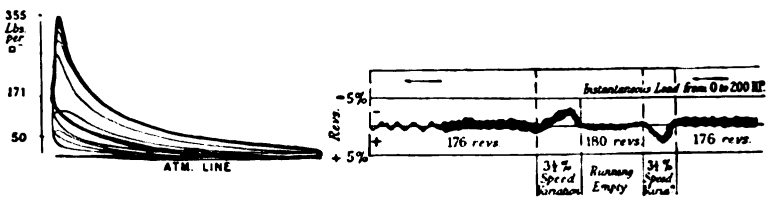


FIG. XV.—33. Diagrams and Tachograph from 200 H.P. double-acting Engine.

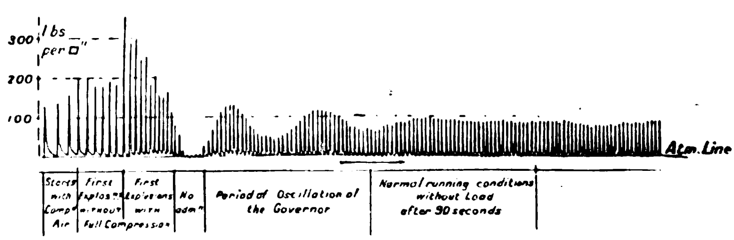


FIG. XV.—34. Explosion Record-starting of 200 H.P. double-acting Engine.

the two cylinders of a Nürnberg 1,000 H.P. engine, it will be seen that high mean pressures of 86·5 and 92·5 lbs. per square inch are obtained even with blast furnace gas of only about 100 B.Th.U. per cubic foot. Such mean pressures as these, however, are somewhat rarely obtained, so that it is customary to calculate the diameter of cylinder and stroke of piston of these engines upon a mean pressure not exceeding 70 lbs. per square inch, as has been previously mentioned.

The diagrams in Fig. XV.—36 were taken from a Crossley tandem

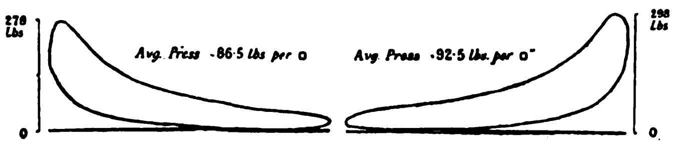


FIG. XV.—35. Diagrams from opposite ends of Cylinder of 1,000 H.P. Nürnberg Engine.

gas engine governed by the mechanism described in a previous chapter (p. 254). The tests were superintended by Dr. J. T. Nicholson, Professor of Mechanical Engineering in the University of Manchester, and were particularly directed to decide the speed variation between

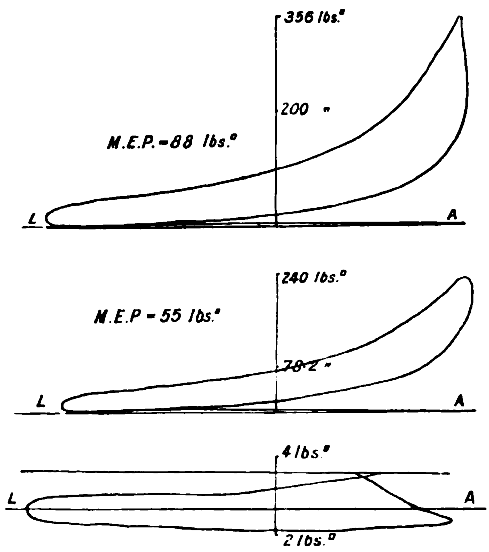


FIG. XV.—36. Diagrams from 600 H.P. Crossley Tandem Gas Engine.

full load and no load. These speeds were respectively 119·4 and 121·4 revolutions per minute when the load was instantaneously dropped from 600 to about 50 H.P. The total speed variation was therefore $1\frac{2}{3}$ per cent. of the mean speed. The full load was then again suddenly applied and taken off several times in succession. The speed of the engine never varied more than the above percentage.

The system of governing must therefore be considered to be extremely sensitive. The engine was fed from a Crossley bituminous fuel gas producer with gas of the following composition:—

Carbon dioxide . . .	11·4	per cent. by volume	
Oxygen	1·0	,,	,,
Carbon monoxide . . .	15·1	,,	,,
Hydrogen	24·3	,,	,,
Methane	3·5	,,	,,
Nitrogen (difference) . .	47·7	,,	,,

The calorific value on the lower scale was 156·5 B.Th.U. per cubic foot at 0° C. (32° F.) and 760 mm. (29·9 inches) of mercury. The calorimetric observations by means of a Junker's calorimeter showed 149 B.Th.U. on the lower scale at atmospheric pressure. The engine was running on a brake load equivalent to 559 H.P. and the consumption of gas (at 0° C. and 760 mm.) was 51·94 cubic feet per B.H.P. hour. The heat supplied was, therefore, 8,128 B.Th.U. per B.H.P. hour, and the thermal efficiency on the B.H.P. was consequently about 31 per cent.

The following indicator cards and explosion records have been taken by Messrs. Hal Williams and Bridges, of London, on a 30 H.P. single-acting suction gas engine, which was constructed in England from designs made under the supervision of the author according to the principles he advocates in the present volume.

The variable admission of a constant mixture is obtained under the control of the governor causing a variable lift of the main inlet valve.

The gas valve is combined with the latter in view of making a self-contained arrangement and that the combustion chamber may be of a shape most favourable to quick propagation of the flame in the mixture.

Gas and air inlet valves, as well as the exhaust valve, are operated by one common cam.

The test of a new engine of which the piston had not yet been ground, gave the following results:—

Diameter of cylinder	12 inches.
Stroke of piston	19 ,,
Revolutions per minute	214·4
Brake H.P.	35·6
Indicated H.P.	43·3
Mechanical efficiency	82·25 per cent.

After working for some time the same engine with an extra heavy fly-wheel developed, as maximum power, 39 to 40 B.H.P.

In Fig. XV.—37 is shown a series of diagrams taken during the

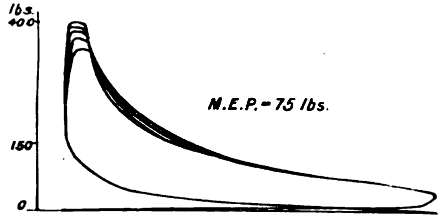


FIG. XV.—37. Full-load Diagrams from Engine designed by the Author.

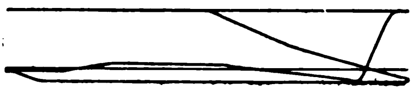


FIG. XV.—38. Resistance Diagram from same Engine.

trial. The mean pressure reached 75 lbs. per square inch. It will be noticed that the diagrams, instead of terminating at a point (and showing a sharp explosion which could increase the mean pressure only to a very slight extent) have a horizontal line which denotes combustion at constant pressure for a certain portion of the stroke, thus producing a marked increase to the area of the diagrams.

Fig. XV.—38, taken with a light spring, shows that the exhaust

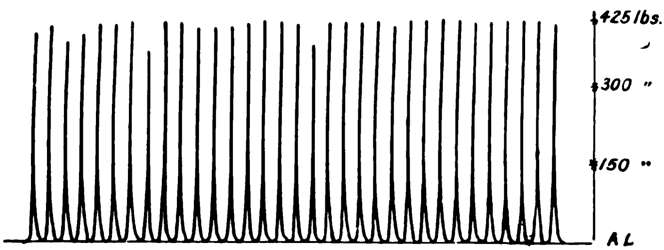


FIG. XV.—39. Explosion Record from same Engine.

pressure at the end of the stroke fell below atmospheric pressure. Scavenging of the burnt products was therefore effected. Fig. XV.—39 shows the extreme regularity of initial explosion pressures. In the majority of engines such regularity does not exist.

CHAPTER XVI

DIMENSIONS, CLASSIFICATIONS, AND TESTS OF ENGINES

The published figures relating to gas engine tests have not hitherto been grouped or classified with a view to the deduction of certain rules, more or less empirical, concerning the calculations connected with gas engine design. The author has selected a method of classification which permits a real comparison of similar engines. The following are the principal headings upon which such classification is based.

The Indicated Power of a four-cycle gas engine, giving one impulse per cycle, is obtained by the formula:—

$$(1) \qquad I.h.p. = P_m \times \frac{2\,L}{12} \times \frac{A}{33000} \times \frac{N}{4} \times K$$

$$= \frac{P_m \times L \times A \times N}{792000} \times K$$

in which $I.h.p.$ = Indicated H.P.
P_m = Mean pressure in lbs. per square inch.
L = Stroke of piston in inches.
A = Area of cylinder (less than area of piston rod if existing).
N = Number of revolutions.
K = Number of impulses per cycle.

For a single-cylinder, single-acting, four-cycle engine $K = 1$.
,, ,, ,, ,, ,, two-cycle ,, $K = 2$.

The Effective Power, $B.h.p.$, is equal to the indicated power multiplied by the mechanical efficiency R, or:

$$B.h.p. = I.h.p. \times R$$

$$(2) \qquad B.h.p. = \frac{P_m \times L \times A \times N}{792000} \times K \times R.$$

Assuming that for a four-cycle single-acting engine $K = 1$; that the number of revolutions per minute $N = 1$; and that the mean pressure $P_m = 1$; the formula (2) becomes:

(3) $$B.h.p. = R = \frac{L \times A}{792000}$$

and assuming further, that the efficiency $R = 1$ we have:

(4) $$M = \frac{L \times A}{792000}.$$

This expression may be called the *modulus of* ABSOLUTE *power* of an engine cylinder and be thus defined:

The modulus of absolute power of an engine cylinder is the power that would theoretically be developed per working face of piston, the engine making but one revolution per minute under a mean pressure equal to unity and an efficiency also equal to unity.

The *modulus of* ACTUAL *power* of the same cylinder will be equal to the *modulus of* ABSOLUTE *power* multiplied by the mechanical efficiency, R, of the engine:

(5) $$m = RM.$$

It will be seen that in the expression of *modulus of* ABSOLUTE *power* only invariable quantities and known features of the engine are concerned. These expressions M and m, therefore, enable precise comparisons to be made between two engines. The general formula (2) thus becomes:

(6) $$B.h.p. = K \times R \times M \times N \times P_m,$$

K, the number of impulses per cycle, and M, the *modulus of* ABSOLUTE *power* are the invariable data; RN and P_m are the variable data.

Theoretically, $B.h.p.$ increases in proportion to the number of revolutions N of the engine. Practically, N cannot exceed a certain value for a given engine, the limit depending upon the speed of flame propagation for the mixture used; upon the construction of the engine; upon the area and lift of valves; upon conditions of governing, &c.

The mean pressure, P_m, is a function of the calorific value of the gas used; upon the terminal compression pressure of the mixture; and upon the more or less favourable conditions under which combustion proceeds.

P_m remains within certain limits which vary according to the type of engine.

The formula (6) can be replaced by:

$$(7) \qquad 1 = \frac{K\,N}{B.h.p.} \times R \times M \times P_m$$

and the expression $\frac{K\,N}{B.h.p.}$ can be termed the *modulus of speed*, n, of the engine and be defined as follows:

The *modulus of speed* of an engine is the number of revolutions per minute at which it should run to develop 1 brake H.P.

The formula (7) would finally be written:

$$(8) \quad 1 = n \times M \times R \times P_m.$$

This very simple expression enables a rational classification of internal combustion engines to be made, and it can thus be defined:

The product of the four factors—modulus of absolute power, modulus of speed, mechanical efficiency, mean pressure—is considered as being equal to unity.

Briefly stated, the *modulus of* ABSOLUTE *power* is given by the formula:

$$M = \frac{L \times A.}{792000}$$

The *modulus of speed* is given by the expression:

$$n = \frac{K\,N}{B.h.p.}$$

The terms:—

K = number of impulses per cycle,
N = ,, ,, revolutions per minute, and
$B.h.p.$ = effective H.P.,

are known, either from a brake test or from figures provided by the makers.

Referring to formula (8) the value of R (mechanical efficiency) is usually between 70 and 85 per cent.

Engines whose mechanical efficiency under full load falls below 70 per cent. must be considered poor, while those in which the efficiency exceeds 85 per cent. are rare.

The mean pressure is frequently more than 60 lbs. per square inch, but is usually lower than 90 lbs., especially with producer gas.

The product of R and P_m varies between the limits of

$$0{\cdot}70 \times 60 = 42{\cdot}0 \text{ and}$$
$$0{\cdot}85 \times 90 = 76{\cdot}5$$

The power of every engine corresponding to the usual working condition can be expressed by an equation between the limits of

$$1 = 42 \times n \times M \quad \text{and,} \quad 1 = 76{\cdot}5 \times n \times M.$$

From the figures obtained in tests made by the author a comparative table has been prepared by taking as a basis the modulus of power of the various engines and arranging them in order of decreasing values. The modulus of speed has been calculated for each case, these increasing in proportion to the decrease of modulus of power.

Sometimes the increase is not regular, but this is explained by the trials not having been carried out in the same manner. In the majority of instances the brake H.P. recorded is not the maximum obtainable from the engine, while some of the tests refer to a remote period and to out-of-date engines.

The table comprises 138 different tests relating to engines of more than 8 to 10 H.P. By its aid, comparisons can be made between engines of practically the same modulus of power, and those in which the explosive mixture is best utilised can be differentiated. For instance :—

(1.) Nos. 14 to 18. The double-acting Deutz engine, No. 18, takes first place with a lower modulus of power than the others. One H.P. is generated for every 1·35 revolutions per minute, whilst for the Winterthur engine, No. 16, tested in 1906, 1 H.P. was developed for every 1·84 revolutions.

(2.) Nos. 23 to 31. The Benz engine, No. 31, with a modulus of power 0·0608 takes first place, before No. 23 with 0·0783. The Benz engine gave out 1 H.P. for 2·12 revolutions per minute and the mean pressure reached 87 lbs. per square inch.

These comparisons, however, must not be considered as being absolutely reliable because, as has already been mentioned, the majority of the tests are incomplete and they have not been conducted at the maximum output of power. Nevertheless, different conclusions can be arrived at from the figures given in the table, as follows :—

Mean Pressures.—These reach from 85 to 92·5 lbs. per square inch even with producer gas. In the Güldner engine, No. 113, tested by Schrötter, it exceeds 130 lbs. per square inch.

Fuel.—The fuel consumptions per B.H.P. hour range between the following limits :—

0·77 lb. to 1·00 lb. of ANTHRACITE of 13,500 to 14,400 B.Th.U. per lb.

14·0 cubic feet	to	17·5 cubic feet	of	TOWN GAS of 560 to 620 B.Th.U. per cubic foot (higher value).
70·0 ,,	,,	88·0 ,,	,,	PRODUCER GAS of 125 to 145 B.Th.U. per cubic foot (higher value).
88·0 ,,	,,	105·0 ,,	,,	BLAST FURNACE GAS of 100 to 112 B.Th.U. per cubic foot.
0·55 lb.	to	1·1 lb.	of	PETROLEUM of 18,900 to 19,800 B.Th.U. per lb.

In the petroleum explosion engines, the conditions of combustion are usually very irregular, and this explains the very marked differences that are noted.

It is useful to tabulate the minimum figures for fuel consumption given in the table,[1] as follows:—

No.	Maker.	Type.	B.H.P.	Fuel.	Consumption per B.H.P. hour.
18	Deutz .	double-acting four-cycle	200	Anthracite. 14,600 B.Th.U. per b.	0·715 lb.
61	Winterthur	single-acting four-cycle	40	Anthracite.	0·694 lb.
124	Catteau .	,,	17	Town Gas. 650 B.Th.U. per c. ft.	13·0 c. ft.
36	Soest .	,,	54	Producer Gas. 153 B.Th.U. per c. ft.	66·0 c. ft.
13	Diesel .	,,	560	Petroleum.	0·407 lb.
114	Diesel .	,,	20	Petroleum. 19,800 B.Th.U. per c. ft.	0·545 lb.
112	Banki .	,,	25	Benzine.	0·51 lb.

Lubricating Oil, and Water, Consumption.—Very few tests have been made in this connection. With regard to lubricating oil the 75 H.P. Delamarre engine, No. 7, consumed ·008 lbs. per H.P. hour. For water, the 53 H.P. Stockport engine, No. 39, consumed 5·3 gallons at the outlet temperature of 113° F.

The water leaving the cylinder is usually at a temperature of between 105 and 120° F., except in the Diesel engine, No. 114, from which it was 71° F.

Formerly, for town gas engines with relatively low compression pressures, the outlet temperature was 140 to 150° F.

[1] The test made April, 1909, of Tangye T-type engine is not included in the tables (see p. 49). The consumption noted was 0·665 lbs. of anthracite per B.h.p. hour at 81 B.h.p. output.

Temperature of Exhaust Gas.—This is between 570 and 930° F., only falling below 570° F. for engines with long expansion, such as the Charon or Sargent or the old types of gas engines like the Hugon engine.

Waste Heat.—This averages from 25 to 30 per cent. with respect to water circulation, and from 40 to 50 per cent. for exhaust gases.

Pressures.—Compression pressures have been carried to 170 lbs. and even 200 lbs. per square inch in some recently built engines designed for producer gas.

Examination of the table shows that the consumption varies inversely to the compression pressure. This, however, is not without exception.

The initial explosion pressures are between 285 and 430 lbs. per square inch for either producer gas or town gas.

Thermal Efficiency.—The thermal efficiency per brake H.P. ranges between 18 and 24 per cent. with regard to anthracite, for the combination of engine and producer, and between 25 and 32 per cent. with respect to gas.

From the preceding considerations it will be recognised that, in power and consumption tests on gas engines, observations should always be made as to the maximum indicated and brake H.P. obtained; the compression pressures and the corresponding explosion pressures at the particular degree of load; the amount and temperatures of water in circulation; and of exhaust gases. A thermal balance-sheet of the trial can then be drawn out.

It is only from experiments made upon the same basis and upon the same methods that it will be possible to obtain comparable results and thus to determine the laws yet unknown which apply to the working of internal combustion engines.

According to Mr. Dugald Clerk (Report of the Committee of the Institution of Civil Engineers upon the Limits of Thermal Efficiency in Internal Combustion Engines, 1907) the efficiency obtained per indicated H.P. with well-designed gas engines of powers varying from 6 to 60 H.P., reached to 50 to 70 per cent. of the ideal theoretical thermal efficiency.

Experiments conducted by the Committee gave results summarised below:

Heat distribution.	Engines.		
	6 H.P.	24 H.P.	60 H.P.
Exhaust gas	41·1	37·1	39·9
Water circulation and external radiation	27·1	29·6	25·4
Indicated work	31·8	33·3	37·7
	100	100	100
Mechanical efficiency	0·84	0·85	0·86

Mr. Dugald Clerk had already conducted tests to determine the heat distribution, and had obtained the following figures:

Lost, during explosion and expansion	16·1
Lost, to exhaust	49·3
Converted to indicated work	34·6
	100

Notes in connection with the following table.

(a) In cases where the mechanical efficiency has not been mentioned by the operator, an assumption of 80 per cent. has been made, and, according to circumstances, the indicated H.P. or brake H.P. has been filled in accordingly, as also the corresponding mean pressure.

(b) *Abbreviations:*

2nd column—	*D.*	Diameter of cylinder in inches.
	d.	„ „ piston rod in inches.
	A.	Effective area of cylinder in square inches.
	L.	Stroke of piston in inches.
	p.	Diameter of pumps.
4th column—	*N.*	Revolutions per minute.
	I.h.p.	Indicated H.P.
	B.h.p.	Effective H.P.
	K.	Number of impulses per cycle of two revolutions.
	R.	Mechanical efficiency.
9th column—	*C.*	Compression pressure in lbs. per square inch.
	I.	Initial explosion pressure „ „ „ „

Translator's note.—Throughout the following table conversions from metric to British standards of measurement have been made except with regard to I.h.p. and B.h.p., which, being but slightly dissimilar, have been reproduced as in the original. In some cases conversion from British to metric standards for use in the original, and re-conversion from metric to British in this translation, may have introduced minor errors that have remained undetected during revision.

No.	Type of Engine, Date of Test, Operator.	Dimensions.	Modulus of Absolute Power $M = \frac{L \times A}{792000}$	Details of Tests.	Modulus of Speed $n = \frac{KN}{B.h.p.}$	Mean Pressure $Pm = \frac{1}{R} \times \frac{1}{Mn}$ lbs. per square inch.
	1	2	3	4	5	6
1	Cockerill (Simplex), 21/3/1900. Hubert.	*D* 51·18 *A* 2,057 *L* 55·07	1·431	*N* 94·37 *I.h.p.* 786·16 *B.h.p.* 575·0 *K* 1·0 *R* 0·73	0·164 88·5 per cent. of explosions.	$\frac{1}{0{\cdot}73} \times \frac{1}{1{\cdot}43 \times 0{\cdot}164}$ $= \frac{1}{0{\cdot}1674} = 59{\cdot}7$ according to diagrams = 99
2	Borsig (Oechelhäuser), 10/10/1904. From Junge.	*D* 26·575 *A* 554·66 *L* 37·5 front, 37·3 back 74·8 total.	0·523	*N* 107·0 *I.h.p.* 839·0 *K* 1·0 *I.h.p.* of pumps 83·1 *I.h.p.* of blower 626·6 Friction of engine 117·3	—	from diagrams = 72·5
3	Koerting, 1/7/1904. ?	*D* 31·0 *d* 8·1 *A* 703·2 *L* 55·12 *Gas pump* *p* 2·756 *Air pump* *p* 3·138 Stroke 42·52	0·4894	*N* 80·0 *I.h.p.* 890·0 *B.h.p.* 703·8 *K* 4·0 *R* 0·791	0·455	$\frac{1}{0{\cdot}791 \times 0{\cdot}455 \times 0{\cdot}4894}$ $= 56{\cdot}8$
4	Cockerill (Simplex), 1901. Hubert.	*D* 33·465 *A* 880·0 *L* 39·37	0·437	*N* 99·74 *I.h.p.* 246·89 *B.h.p.* 215·31 *K* 1·0 *R* 0·87	0·464 89·3 per cent. of explosions.	$\frac{1}{{\cdot}87 \times {\cdot}437 \times {\cdot}464} = 56{\cdot}7$ according to diagrams = 27·2
5	Cockerill (Simplex), 20/7/1898. Witz.	*D* 31·5 *A* 779·0 *L* 39·37	0·387	*N* 105·20 *I.h.p.* 226·45 *B.h.p.* 181·16 *K* 1·0 *R* 0·80	0·58 89·3 per cent. explosions.	55·0

Consumption per B.H.P. hour.	Details.			Notes.	No.
	Temperatures, deg. Fah.	Pressures, lbs. per square inch.	Thermal Efficiency.		
7	8	9	10	11	
Blast furnace gas, 124 cub. ft. at 109·5 B.Th.U. per cub. ft., higher value. = 13,650 B.Th.U. Water, 15·2 gallons.	Exhaust 950 Water 91·5	C. 135	0·183	—	1
Coke-oven gas.	—	—	—	*Air pump* Stroke, 19·68 ; Dia. front piston, 35·43 ; Dia. back piston, 27·56. *Gas pump* Stroke, 19·68 ; Dia. piston, 23·2.	2
Anthracite 0·78 lbs.	—	—	—	Work absorbed by pumps = 91·9 H.P., $\frac{91{\cdot}9}{890} = 0{\cdot}103$. Work absorbed by mechanism 890 − 91·9 − 703·8 = 94·3 H.P. $\frac{94{\cdot}3}{890} = 0{\cdot}106$. 1·0 − 0·103 − 0·106 = 0·791.	3
Blast furnace gas, 120·5 cub. ft. at 104·3 B.Th.U. per cub. ft. = 12,600 B.Th.U.	Exhaust gas 1,100	C. 152 I. 237	0·200	—	4
Blast furnace gas, 117 cub. ft. at 110 B.Th.U. per cub. ft. = 12,960 B.Th.U.	—	—	0·193	Scrubbers absorbing 6·6 gallons per H.P.	5

.	Type of Engine, Date of Test, Operator.	Dimensions.	Modulus of Absolute Power $M=\frac{L\times A}{792000}$	Details of Tests.	Modulus of Speed $n=\frac{KN}{B.h.p.}$	Mean Pressure $Pm=\frac{1}{R}\times\frac{1}{Mn}$ lbs. per square inch.
	1	2	3	4	5	6
6	Crossley, 7–8/3/1906. Mathot (241).	D 25·0 A 490·9 L 36·0	0·223	N 104·0 $I.h.p.$ 651·0 $B.h.p.$ 515·0 K 4·0 R 0·79	0·80	70·0
7	Delamarre, 12/9/1890. Witz.	D 22·64 A 402·0 L 37·4	0·190	N 100·8 $I.h.p.$ 109·9 $B.h.p.$ 75·86 K 1·0 R 0·69	1·32	57·0
8	Cockerill, 25/5/1901. François.	D 23·622 A 438·0 L 31·5	0·1743	N 156·0 $B.h.p.$ 107·0 K 1·0	1·46 85 per cent. explosions.	
				N 156·0 $I.h.p.$ 184·4 $B.h.p.$ 139·5 K 1·0 R 0·76	1·12	66·5
9	Letombe (Triplex), 28/12/1901. Witz.	D 23·622 A 438·0 L 31·5	0·1743	N 131·01 $I.h.p.$ 368·1 $B.h.p.$ 294·51 K 3·0 R 0·80	1·33	53·5
10	Société de Mech. Ind. d'Anzin, 9–10/5/1908. Gaz de Rennes.	D 24·4 A 470·5 L 31·5	0·1861	N 165·0 $I.h.p.$ 375·93 $B.h.p.$ 309·75 K 2·0 R 0·80	1·10	60·5
11	Ehrhardt & Sehmer, 1906. Saarbruck.	D 24·40 d 6·69 A 433·0 L 29·52	0·1614	N 150·0 $I.h.p.$ 723·0 $B.h.p.$ 600·0 K 4·0 R 0·83	1·00	74·0
12	National Gas Engine Co., 26/2/1907. Hal Williams & Bridges.	D 22·5 A 397·6 L 30·0	0·1509	N 169·88 $I.h.p.$ 218·7 $B.h.p.$ 184·0 K 1·0 R 0·841	0·92 81·7 per cent. explosions.	85 lbs., according to diagram = 88 lbs.

Consumption per B.H.P. hour	Details.			Notes	No.
	Temperatures, deg. Fah	Pressures, lbs. per square inch.	Thermal Efficiency.		
7	8	9	10	11	
Producer gas, 67 cub. ft. at 142 B.Th.U. per cub. ft. = 9,500 B.Th.U.	—	—	0·264	Two-cylinder engine, tandem double-acting.	6
Coke and coal, 1·345 lbs. Producer gas, 96 cub. ft. at 168 B.Th.U. per cub. ft. = 16,170 B.Th.U. Water, 15·15 gallons. Oil, 0·00845 lbs.	Water, 109 Gas, 820	C. 85·5	0·158	Dowson Producer.	7
Blast furnace gas, 117·8 cub. ft. at 110 B.Th.U. per cub. ft. = 13,360 B.Th.U.	—	C. 107 I. 314	0·194	Pressure at end of expansion, 42·7 lbs. per square inch.	8
Anthracite, 0·82 lbs. Producer gas, 76 cub. ft. at 152 B.Th.U. per cub. ft. (high) = 11,800 B.Th.U.	Water, 131 to 149 Gas, 660	C. 118	0·218	—	9
Mixture of coke and anthracite about half-and-half, 1·032 lbs. gross, 0·79 lbs. cinders, deducted.	—	—	—	Single-acting, tandem-engine.	10
Coke oven gas, 17·8 cub. ft. at 460 B.Th.U. per cub. ft. = 8,100 B.Th.U.	—	—	0·310	—	11
Coal, 0·91 lbs. Water, 3·64 gallons.	—	—	—	—	12

No.	Type of Engine. Date of Test. Operator.	Dimensions.	Modulus of Absolute Power $M = \frac{L \times A}{792000}$	Details of Tests.	Modulus of Speed $n = \frac{KN}{B.h.p.}$	Mean Pressure $Pm = \frac{1}{R} \times \frac{1}{Mn}$ lbs. per square inch.
	1	2	3	4	5	6
13	Diesel, 7/2/1905. Michael Longridge.	D 22·0 A 380·1 L 29·52	0·1415	N 152·8 $I.h.p.$ 698·7 $B.h.p.$ 562·5 K 3·0 R 0·805	0·81	106·5 lbs., according to diagram = 96 lbs.
14	Winterthur, 9/3/1905. Mathot.	D 20·47 A 330·0 L 29·92	0·1244	N 157·9 $I.h.p.$ 139·3 $B.h.p.$ 111·5 K 1·0 R 0·80	1·41	70·5 lbs.
15	Winterthur, 1902. Allaire.	D 20·47 A 330·0 L 29·92	0·1244	N 160·0 $I.h.p.$ 131·0 $B.h.p.$ 104·83 K 1·0 R 0·80	1·52	65·5 according to diagrams = 70 lbs.
		Same.		N 160 $I.h.p.$ 260 $B.h.p.$ 208·15 K 2·0 R 0·80	1·53	65·0
16	Winterthur, 9/4/1896. Meyer.	Same.	0·1244	N 137·7 $I.h.p.$ 178·3 $B.h.p.$ 149·8 K 2·0 R 0·84	1·84	54·0
17	Otto-Deutz, 5/4/1905. Mathot (248).	D 21·26 A 354·5 L 27·56	0·1235	N 180·0 $I.h.p.$ 274·7 $B.h.p.$ 214·5 K 2·0 R 0·78	1·68	60·8 Diagram— right-hand cylinder 7[illegible] left-hand „ 6[illegible]
				Full load. N 180·0 $I.h.p.$ 301·2 $B.h.p.$ 241·02 K 2·0 R 0·80	1·50	67·5
18	Otto-Deutz, 15/3/1904. Witz and Mathot	D 21·26 A 354·5 L 27·56	0·1235	N 150·2 $I.h.p.$ 278·5 $B.h.p.$ 222·83 K 2·0 R 0·80	1·35	74·0

Consumption per B.H.P. hour.	Details.			Notes.	No.
	Temperatures, deg. Fah.	Pressures, lbs. per square inch.	Thermal Efficiency.		
7	8	9	10	11	
Petroleum, 0·407 lbs.	Water, 71 increase. Gas, 650	C. 975 (in pump). I. 530.	—	—	13
—	—	—	—	With suction gas producer.	14
Anthracite, 0·868 lbs. at 14,200 B.Th.U. per lb. = 12,350 B.Th.U.	—	C. 165. I. 276.	0·205	With pressure producer.	15
Anthracite, 0·877 lbs. at 14,200 B.Th.U. per lb. = 12,400 B.Th.U.	—	—	0·206	—	
Coke, 1·63 lbs. at 13,300 B.Th.U. per lb. = 21,600 B.Th.U. Gas, 96 cub. ft. at 135 B.Th.U. per c. ft. = 13,000 B.Th.U. Water, 5·75 gallons.	Water, 98·5	—	0·193 per I.H.P. 0·162 per B.H.P.	Heat lost to water, 0·259. " " exhaust, 0·548.	16
—	—	—	—	Engine with twin cylinders.	17
Anthracite, 0·715 lbs. at 14,600 B.Th.U. per lb. = 10,600 B.Th.U. Producer gas, 139 B.Th.U. per cub. ft. Water, 7·75 gallons.	—	—	0·244	Double-acting engine with suction gas producer. Water consumption included generator and engine.	18

No.	Type of Engine. Date of Test. Operator.	Dimensions.	Modulus of Absolute Power $M = \frac{L \times A}{792000}$	Details of Tests.	Modulus of Speed $n = \frac{KN}{B.h.p.}$	Mean Pressure $Pm = \frac{1}{R} \times \frac{1}{Mn}$ lbs. per square inch.
	1	2	3	4	5	6
19	Stockport, 26/2/1904. Musker.	*D* 20·0 *A* 314·16 *L* 27·0	0·1071	*N* 182·0 *I.h.p.* 141·0 *B.h.p.* 121·0 *K* 1·0 *R* 0·86	1·5 91 per cent. explosions.	71·0
20	Simplex, 7/3/1904. Bourdon.	*D* 18·89 *A* 280·0 *L* 27·56	0·0975	*N* 138·6 *I.h.p.* 78·2 *B.h.p.* 62·55 *K* 1·0 *R* 0·80	2·21	57·0
21	Otto-Deutz, 6/7/1907. Mathot (261).	*D* 19·68 *A* 304·0 *L* 24·4	0·09376	*N* 180·0 *I.h.p.* 134·93 *B.h.p.* 106·08 *K* 1·0 *R* 0·78 *N* 180·0 *I.h.p.* 144·3 *B.h.p.* 115·45 *K* 1·0 *R* 0·80	1·70 Average load. 1·55	79·5 85·0
22	Stockport, 26/2/1904. Musker.	*D* 18·0 *A* 254·5 *L* 27·0	0·08668	*N* 180·0 *I.h.p.* 113·0 *B.h.p.* 96·0 *K* 1·0 *R* 0·85	1·87 85 per cent. explosions.	71·5
23	Cie. Berlin Anhalt, 6/10/1898. Witz.	*D* 16·92 *A* 225·0 *L* 27·56	0·0783	*N* 160·6 *I.h.p.* 80·5 *B.h.p.* 64·45 *K* 1·0 *R* 0·80	2·49 91 per cent. explosions.	63·5
24	Benier, 3/4/1901. Witz.	*D* 17·32 *A* 235·7 *L* 25·6	0·0734	*N* 126·6 *I.h.p.* 52·8 *B.h.p.* 42·24 *K* 2·0 *R* 0·80	6·0	27·0
25	Charon, 4/9/1898. Fischener.	*D* 16·93 *A* 225·0 *L* 24·8	0·0701	*N* 153·8 *I.h.p.* 48·5 *B.h.p.* 37·19 *K* 1·0 *R* 0·80	4·13	42·4

Consumption per B.H.P. hour.	Details.			Notes.	No.
	Temperatures, deg. Fah.	Pressures, lbs. per square inch.	Thermal Efficiency.		
7	8	9	10	11	
Coal, 1·4 lbs.	—	—	—	Wilson producer.	19
Coal, 1·325 = 13,000 B.Th.U.	—	—	—	—	20
Anthracite, 0·735 lbs. Oil = 0·00396.	—	C. 156·5 I. 442·0	—	—	21
Coal, 1·4 lbs.	—	—	—	Wilson producer.	22
Producer gas, 100 cub. ft. at 117·7 B.Th.U. per cub. ft. = 11,700 B.Th.U.	—	C. 120 I. 242	0·215		23
Coal, 1·52 lbs.	—	—	—	–	24
Anthracite, 0·962 lbs. at 14,900 B.Th.U. per lb. = 14,400 B.Th.U.	—	—	0·176	Taylor producer.	25

No.	Type of Engine. Date of Test. Operator.	Dimensions.	Modulus of Absolute Power $M = \frac{L \times A}{792000}$	Details of Tests.	Modulus of Speed $n = \frac{KN}{B.h.p.}$	Mean Pressure $Pm = \frac{1}{R} \times \frac{1}{Mn}$ lbs. per square inch.
	1	2	3	4	5	6
26	Crossley, 22/6/1894. Witz.	*D* 16·93 *A* 225·0 *L* 23·89	0·0678	*N* 164·36 *I.h.p.* 77·46 *B.h.p.* 60·18 *K* 1·0 *R* 0·78	2·74 90 per cent. explosions.	68·0
27	Same.	*D* 16·97 *A* 226·0 *L* 23·82	0·0678	*N* 164·36 *I.h.p.* 68·77 *B.h.p.* 56·75 *K* 1·0 *R* 0·83	2·87	62·0
28	Crossley, 1895. Meyer.	*D* 16·9 *A* 224·0 *L* 23·9	0·0675	*N* 165·4 *I.h.p.* 63·7 *B.h.p.* 57·6 *K* 1·0 *R* 0·90	2·87 93·5 per cent. explosions.	56·5 according to diagrams } 59·5
29	Schmitz, 20/4/1903. Mathot (133).	*D* 16·14 *A* 204·2 *L* 25·98	0·0670	*N* 190·0 *I.h.p.* 92·3 *B.h.p.* 73·9 *K* 1·0 *R* 0·80	2·57	71·5 according to diagrams } 75·4
30	Schmitz, 25/2/1904. Mathot (181).	Same.	0·0670	*N* 192·0 *I.h.p.* 93·0 *B.h.p.* 72·23 *K* 1·0 *R* 0·78	2·65	71·6
31	Benz, 10/2/1905. Mathot (211).	*D* 16·73 *A* 220·0 *L* 22·0	0·0608	*N* 182·0 *I.h.p.* 97·8 *B.h.p.* 85·5 *K* 1·0 *R* 0·875	2·12	87·0
				N 183·2 *I.h.p.* 94·8 *B.h.p.* 77·3 *K* 1·0 *R* 0·818	2·37	79·5
32a	Schmitz, 5/7/1906. Mathot (240).	*D* 16·18 *A* 204·5 *L* 23·62	0·0608	*N* 82·0 *I.h.p.* 79·4 *B.h.p.* 62·5 *K* 1·0 *R* 0·787	2·91	71·0

Consumption per B.H.P. hour.	Details.			Notes.	No.
	Temperatures, deg. Fah.	Pressures, lbs. per square inch.	Thermal Efficiency.		
7	8	9	10	11	
—	—	—	—	Dowson producer.	26
—	—	—	—	—	27
Coal, 1·36 lbs. Producer gas, 90 cub. ft.	—	—	—	—	28
Anthracite, 1·095 lbs.	—	C. 188 I. 285	—	—	29
Anthracite, 1·02 lbs. Producer gas, 157 B.Th.U. per cub. ft. (high).	—	C. 160 I. 228	—	—	30
—	—	C. 206 I. 490	—	—	31
Anthracite, 0·9 lb. at 14,600 B.Th.U. per lb. = 13,200 B.Th.U.	—	C. 192 I. 412	0·190	—	
Anthracite, 0·744.	—	C. 176 I. 373	—	—	32a

No.	Type of Engine. Date of Test. Operator.	Dimensions.	Modulus of Absolute Power $M=\frac{L \times A}{792000}$	Details of Tests.	Modulus of Speed $n = \frac{KN}{B.h.p.}$	Mean Pressure $Pm = \frac{1}{R} \times \frac{1}{M_n}$ lbs. per square inch
	1	2	3	4	5	6
32b	Schmitz 10/2/1906. Mathot (211).	Same.	0·0608	*N* 180·0 *I.h.p.* 84·6 *B.h.p.* 67·7 *K* 1·0 *R* 0·80	2·66 Max. load.	76·0
33	Schmitz, 28/5/1906. Mathot (244).	*D* 16·14 *A* 204·4 *L* 23·62	0·0608	*N* 180·0 *I.h.p.* 72·6 *B.h.p.* 59·2 *K* 1·0 *R* 0·80	3·04	66·5
34	Tangye, 14/9/1903. Mathot (156).	*D* 16·0 *A* 201·0 *L* 23·0	0·0585	*N* 170·0 *I.h.p.* 64·0 *B.h.p.* 50·0 *K* 1·0 *R* 0·78	3·40	63·5 according to diagrams } 66·0
35	Baechtold, 20/6/1906. Mathot (240).	*D* 15·75 *A* 194·8 *L* 23·62	0·058	*N* 174·0 *I.h.p.* 65·9 *B.h.p.* 52·72 *K* 1·0 *R* 0·80	3·30	70·0
36	Soest, 11/2/1904. Mathot (177).	*D* 15·75 *A* 194·8 *L* 22·83	0·056	*N* 181·8 *I.h.p.* 72·2 *B.h.p.* 54·5 *K* 1·0 *R* 0·755	3·33	70·0
37	Charon, 5/3/1901. Mathot (50).	*D* 15·0 *A* 176·7 *L* 23·8	0·0527	*N* 140·0 *I.h.p.* 26·5 *B.h.p.* 19·1 *K* 1·0 *R* 0·72	7·33	35·56
38	Schleicher, 1892. Spangler.	*D* 14·5 *A* 165·1 *L* 25·0	0·052	*N* 160·0 *I.h.p.* 127·9 *B.h.p.* 92·5 *K* 2·0 *R* 0·724	3·45	75·0
39	Stockport, 14/6/1901. Mathot (51).	*D* 15·5 *A* 188·7 *L* 22·0	0·0524	*N* 212·0 *I.h.p.* 63·0 *B.h.p.* 53·0 *K* 1·0 *R* 0·84	4·0 85 per cent. explosions.	55·5 according to diagrams } 70·0

Consumption per B.H.P. hour.	Details. Temperatures, deg. Fah.	Pressures, lbs. per square inch.	Thermal Efficiency.	Notes.	No.
7	8	9	10	11	
					32b
Anthracite, 0·946 lb.	—	—	—		33
Anthracite, 1·18 lbs.	—	C. 106 I. 298	—	—	34
Anthracite, 0·765 lb. Gas, 145 B.Th.U. per cub. ft.	—	C. 156 I. 295	—	—	35
Dowson Gas, 66 cub. ft. at 153 B.Th.U. per cub. ft. (high) = 10,080 B.Th.U.	—	C. 108 I. 282	0·250	—	36
—	—	C. 85 I. 114	—	Taylor Producer.	37
Anthracite, 1·32 lbs.	—	C. 78	—	—	38
Anthracite, 1·02 lbs., at 13,060 B.Th.U. per lb. = 13,200 B.Th.U.	Water, 113. 5·3 gallons.	I. 314	0·185	—	39

No.	Type of Engine. Date of Test. Operator.	Dimensions.	Modulus of Absolute Power $M = \frac{L \times A}{792000}$	Details of Tests.	Modulus of Speed $n = \frac{KN}{B.h.p.}$	Mean Pressure $Pm = \frac{1}{R} \times \frac{1}{Mn}$ lbs. per square inch.
	1	2	3	4	5	6
40	Charon, 5/3/1895. Witz.	D 14·96 A 176·0 L 23·62	0·0523	N 152·23 $I.h.p.$ 77·36 $B.h.p.$ 61·89 K 2·0 R 0·80	4·92	48·0
41	Deutz, 15/3/1904. Mathot (187).	D 16·53 A 214·3 L 18·89	0·0511	N 188·7 $I.h.p.$ 77·0 $B.h.p.$ 65·1 K 1·0 R 0·846	2·9	78·0
42	Winterthur, 14/10/1907. Mathot (262).	D 14·567 A 166·8 L 23·62	0·0496	N 190·0 $I.h.p.$ 60·5 $B.h.p.$ 48·4 K 1·0 R 0·80	3·92	63·5
43	Charon, ?/5/1897. Rancial.	D 14·173 A 158·0 L 23·62	0·047	N 154·0 $I.h.p.$ 77·06 $B.h.p.$ 61·65 K 2·0 R 0·80	4·98	52·6
44	Tangye, 16/1/1903. Mathot (125).	D 14·5 A 165·1 L 22·0	0·0458	N 194·9 $I.h.p.$ 83·1 $B.h.p.$ 66·5 K 1·0 R 0·80	2·93	91·0
45	Deutz, ?/7/1903. Mathot (150).	D 14·173 A 158·0 L 22·87	0·0455	N 192·0 $I.h.p.$ 41·1 $B.h.p.$ 32·3 K 1·0 R 0·82	5·94	55·5
				N 192·0 $I.h.p.$ 49·05 $B.h.p.$ 40·27 K 1·0 R 0·82	4·76	—
46	Tangye, 23/2/1904. Mathot (180).	D 14·5 A 165·1 L 22·0	0·0458	N 170·8 $I.h.p.$ 48·6 $B.h.p.$ 38·9 K 1·0 R 0·80	4·39 85 per cent explosions	78·0

Consumption per B.H.P. hour.	Details.			Notes.	No.
	Temperatures, deg. Fah.	Pressures, lbs. per square inch.	Thermal Efficiency.		
7	8	9	10	11	
Town gas, 14·6 cub. ft., at 630 B.Th.U. per cub. ft. (high) = 9,300 B.Th.U.	Water, 192 Gas, 427	—	0·26	—	40
Anthracite, 0·825 lb., at 13,600 B.Th.U. per lb. = 10,800 B.Th.U.	Water, Breech end, 109 Cylinder, 127	C. 170 I. 385	0·243	Final expansion pressure, 24 lbs. per square inch	41
Anthracite, 0·805 lb.	—	—	—	—	42
Town gas, 16·55 cub. ft. Water, 5·35 gallons.	—	—	—	—	43
Town gas, 16·2 cub. ft., at 640 B.Th.U. per cub. ft. (high) = 10,370 B.Th.U.	—	C. 71 I. 284	0·245	—	44
Anthracite, 1·075 lbs.	—	—		—	45
Producer gas, at 173 B.Th.U. per cub. ft. (high).	—	C. 149 I. 270	—	—	
Anthracite, 1·0 lb. Producer gas, at 129·5 B.Th.U. per cub. ft.	—	C. 74 I. 214	—	—	46

No.	Type of Engine. Date of Test. Operator.	Dimensions.	Modulus of Absolute Power $M = \frac{L \times A}{792000}$	Details of Tests.	Modulus of Speed $n = \frac{KN}{B.h.p.}$	Mean Pressure $P_m = \frac{1}{R} \times \frac{1}{Mn}$ lbs. per square inch.
	1	2	3	4	5	6
47	Charon, 19/4/1894. Allaire.	*D* 13·78 *A* 149·0 *L* 23·62	0·0444	*N* 151·0 *I.h.p.* 58·3 *B.h.p.* 53·15 *K* 2·0 *R* 0·91	5·68	42·8
48	Fetu, 10/9/1903. Mathot (155).	*D* 13·78 *A* 149·0 *L* 22·0	0·0414	*N* 200·0 *I.h.p.* 46·0 *B.h.p.* 35·07 *K* 1·0 *R* 0·80	5·70 95 per cent. explosions.	52·5
49	Otto-Deutz, 22/5/1905. Mathot (223).	*D* 15·75 *A* 194·8 *L* 16·5	0·0406	*N* 187·7 *I.h.p.* 62·3 *B.h.p.* 49·85 *K* 1·0 *R* 0·80	3·76	81·0
				N 190·0 *I.h.p.* 40·73 *B.h.p.* 28·28 *K* 1·0 *R* 0·70	6·71 Half load.	51·7
50	Crossley, 25/11/1904. Burstall.	*D* 13·97 *A* 153·0 *L* 21·0	0·0406	*N* 166·0 *I.h.p.* 60·5 *B.h.p.* 49·7 *K* 1·0 *R* 0·82	3·34 81 per cent. explosions.	88·5 according to diagrams } 91·4
51	Schmitz, 18/4/1904. Mathot (104).	*D* 13·78 *A* 149·0 *L* 21·26	0·040	*N* 200·0 *B.h.p.* 44·5 *K* 1·0 *R* 0·80	4·5	68·25
52	Schmitz, 29/10/1903. Mathot (161).	*D* 13·78 *A* 149·0 *L* 21·26	0·040	*N* 192·0 *I.h.p.* 58·2 *B.h.p.* 46·6 *K* 1·0 *R* 0·80	4·12	74·6
53	Stockport, 30/3/1902. Mathot.	*D* 14·0 *A* 154·0 *L* 20·27	0·0395	*N* 190·0 *I.h.p.* 65·8 *B.h.p.* 56·0 *K* 1·0 *R* 0·85	3·39	82·0

Consumption per B.H.P. hour.	Details.			Notes.	No.
	Temperatures, deg. Fah.	Pressures, lbs. per square inch.	Thermal Efficiency.		
7	8	9	10	11	
Town gas, 16·95 cub. ft. at 66° F.	—	—	—	—	47
Anthracite, 1·107 lbs. at 14,400 B.Th.U. per lb. = 17,600 B. Th.U.	—	C. 134 I. 248	0·144	—	48
Anthracite, 0·716 lb. Producer gas, 118 B.Th. U. per cub. ft. Oil, 0·0057 lb. Anthracite, 1·0 lb. Producer gas, 120 B.Th. U. per cub. ft.	— — —	C. 185 I. 344 —	— —	— —	49
Town gas, 14·43 cub. ft. at 578 B.Th.U. per cub. ft. = 8,350 B.Th.U. Water injected in cylinder, 0·131 lb. Cooking water, 2·56 gallons.	Water, 77·52, increase. Gas, 718.	—	0·308	Water injection. Heat lost to water, 0·29. " " exhaust, 0·336.	50
Anthracite, 0·89 lb. Producer gas at 139 B.Th.U. per cub. ft.	—	—	—	—	51
Anthracite, 0·89 lb. at 14,600 B.Th.U. per lb. = 13,000 B. Th.U.	—	C. 134 I. 342	0·193	Vaporiser, 0·04 gallons per lb. of coal.	52
—	—	—	—	—	53

No.	Type of Engine. Date of Test. Operator.	Dimensions.	Modulus of Absolute Power $M = \frac{L \times A}{792000}$	Details of Tests.	Modulus of Speed $n = \frac{KN}{B.h.p.}$	Mean Pressure $Pm = \frac{1}{R} \times \frac{1}{Mn}$ lbs. per square inch.
	1	2	3	4	5	6
54	Stockport, 12/12/1901. Mathot.	*D* *A* *L* } Same.	0·0395	*N* 190·0 *I.h.p.* 64·7 *B.h.p.* 55·0 *K* 1·0 *R* 0·85	3·45 91 per cent. explosions.	85·3
55	Otto-Deutz, 18/9/1903. Mathot (157).	*D* 13·0 *A* 132·73 *L* 22·83	0·0381	*N* 196·0 *I.h.p.* 54·3 *B.h.p.* 43·5 *K* 1·0 *R* 0·80	4·5	72·0
56	Otto-Deutz, 4/3/1904. Mathot (182).	*D* *A* *L* } Same.	0·0381	*N* 186·6 *B.h.p.* 35·5 *K* 1·0 *R* 0·80	5·25	61·5
				N 188·0 *B.h.p.* 43·4 *K* 1·0 *R* 0·80	4·33	75·0
57	Crossley, 27/10/1903. Witz.	*D* 13·19 *A* 136·5 *L* 20·9	0·036	*N* 163·36 *I.h.p.* 43·0 *B.h.p.* 38·05 *K* 1·0 *R* 0·884	4·29 90 per cent. explosions.	72·5 according to diagrams } 77·5
58	Schmitz, 26/8/1902. Mathot (92)	*D* 13·0 *A* 132·73 *L* 21·26	0·0355	*N* 195·0 *I.h.p.* 60·4 *B.h.p.* 48·3 *K* 1·0 *R* 0·80	4·04 98 per cent. explosions.	86·0
59	Niel, 10/11/1901. Witz.	*D* 13·78 *A* 149·0 *L* 19·0	0·0355	*N* 213·4 *I.h.p.* 53·54 *B.h.p.* 46·02 *K* 1·0 *R* 0·86	4·63	70·0
60	Stockport, 31/10/1902. Mathot (95).	*D* 14·0 *A* 154·0 *L* 20·23	0·0352	*N* 210·0 *I.h.p.* 65·0 *B.h.p.* 51·0 *K* 1·0 *R* 0·78	4·12 93 per cent. explosions.	58·5

Consumption per B.H.P. hour.	Details.			Notes.	No.
	Temperatures, deg. Fah.	Pressures, lbs. per square inch.	Thermal Efficiency.		
7	8	9	10	11	
Town gas at 32° F. 15·8 cub. ft.	—	C. 86·75 I. 295	—	—	54
Town gas, 16·6 cub. ft.	—	C. 136·5 I. 284	—	—	55
Anthracite, 1·11 lbs. Producer gas, 126·5 B.Th.U. per cub. ft.	—	C. 138 I. 314	0·310	—	56
Town gas, 15·1 cub. ft. at 540 B.Th.U. per cub. ft. (low) = 8,140 B.Th.U.	—	—	0·308	—	
Town gas at 32° F. 21·45 cub. ft. at 565 B.Th.U. per cub. ft. = 12,100 B.Th.U.	—	—	0·207	—	57
Town gas, 18·0 cub. ft.	—	C. 121 I. 284	—	—	58
Town gas, 15·5 cub. ft., at 63.5 B.Th.U. per cub. ft. = 9,900 B.Th.U.	Water, 159	C. 170	0·255	Heat lost to water 0·216 ,, ,, friction 0·141	59
Town gas at 32° F. 16·0 cub. ft.	—	C. 92·5 I. 340	—	—	60

No.	Type of Engine. Date of Test. Operator.	Dimensions.	Modulus of Absolute Power $M=\frac{L\times A}{792000}$	Details of Tests.	Modulus of Speed $n=\frac{KN}{B.h.p.}$	Mean Pressure $Pm=\frac{1}{R}\times\frac{1}{Mn}$ lbs. per square inch.
	1	2	3	4	5	6
61	Winterthur, 22/3/1905. Mathot (215).	*D* 13·78 *A* 149·0 *L* 20·75	0·0352	*N* 192·0 *I.h.p.* 55·45 *B.h.p.* 43·53 *K* 1·0 *R* 0·78	4·41	81·0
				N 192·0 *I.h.p.* 59·8 *B.h.p.* 47·88 *K* 1·0 *R* 0·80	4·01	87·7
62	Testing-shop, Onghena Works, ?/3/1908. Fornier.	*D* 12·6 *A* 124·5 *L* 20·86	0·0328	*N* 199·0 *I.h.p.* 58·12 *B.h.p.* 47·3 *K* 1·0 *R* 0·81	4·20	88·5
	Couckelaere Electricity Works (figures given by makers).			*N* 200·3 *I.h.p* 55·45 *B.h.p.* 44·63 *K* 1·0 *R* 0·80	4·48 Max. load = *B.h.p.* 49·3.	84·5
63	Tangye, 12/7/1906. Mathot (246).	*D* 12·5 *A* 122·7 *L* 21·0	0·0326	*N* 180·0 *I.h.p.* 34·95 *B.h.p.* 27·96 *K* 1·0 *R* 0·80	6·47	58·25
64	Winterthur (Soc. d'Anzin), 22/10/1906. Mathot (252).	*D* 12·8 *A* 128·5 *L* 19·68	0·0318	*N* 194·0 *I.h.p.* 37·6 *B.h.p.* 30·05 *K* 1·0 *R* 0·80	6·45	60·0
65	Stockport, 6/4/1900. Mathot (33).	*D* 13·0 *A* 132·7 *L* 19·0	0·0318	*N* 220·0 *I.h.p.* 63·7 *B.h.p.* 50·2 *K* 1·0 *R* 0·78	4·38 94 per cent. explosions.	91·0
66	Otto-Deutz, 16/6/1904. Mathot (200).	*D* 13·78 *A* 149·0 *L* 16·5	0·0311	*N* 202·0 *I.h.p.* 38·2 *B.h.p.* 31·5 *K* 1·0 *R* 0·82	6·41	61·0

Consumption per B.H.P. hour.	Details.			Notes.	No.
	Temperatures, deg. Fah.	Pressures, lbs. per square inch.	Thermal Efficiency.		
7	8	9	10	11	
Anthracite, 0·694 lbs. Producer gas, at 145 B.Th.U. per cub. ft. Analysis:— CO_2 7·4 O 0·5 CO 24·0 H 15·7 N 52·4	—	C. 142 I. 356	—	—	61
Town gas at 32° F., 29·9 inches mercury, 20 cub. ft. at 555 B.Th.U. per cub. ft. = 11,150 B.Th.U. Anthracite, 0·81 lb.	—	—	—	—	62
Anthracite, 1·11 lbs.	—	—	—	—	63
Anthracite, 0·98 lb. Producer gas at 134·5 B.Th.U. per cub. ft.	—	—	—	—	64
Town gas, 15·6 cub. ft.	—	C. 78 I. 400	—	—	65
Anthracite, 0·935 lb. Producer gas at 158 B.Th.U. per cub. ft. (high).	—	C. 148 I. 384	—	—	66

No.	Type of Engine, Date of Test. Operator.	Dimensions.	Modulus of Absolute Power $M=\frac{L \times A}{792000}$	Details of Tests.	Modulus of Speed $n=\frac{KN}{B.h.p.}$	Mean Pressure $Pm = \frac{1}{R} \times \frac{1}{Mn}$ lbs. per square inch.
	1	2	3	4	5	6
67	Tangye, 18/3/1902. Mathot (76).	*D* 12·5 *A* 122·7 *L* 19·84	0·0306	*N* 210·0 *I.h.p.* 54·8 *B.h.p.* 45·5 *K* 1·0 *R* 0·83	4·61 95 per cent. explosions.	84·0 according to diagrams } 88·2
68	Soest, 30/3/1905. Mathot (216).	*D* 12·25 *A* 117·86 *L* 20·5	0·0305	*N* 190·0 *I.h.p.* 47·96 *B.h.p.* 38·37 *K* 1·0 *R* 0·80	4·95	82·0
69	Bollinckx, 6/4/1907. Mathot (258).	*D* 11·81 *A* 109·5 *L* 20·67	0·0286	*N* 196·0 *I.h.p.* 37·79 *B.h.p.* 31·54 *K* 1·0 *R* 0·85	6·21	68·6 according to diagram } 69·6
				N 184·2 *I.h.p.* 41·68 *B.h.p.* 35·43 *K* 1·0 *R* 0·85	5·20 Max. load.	78·5
70	Diesel, ?/9/1900. Meyer.	*D* 11·81 *A* 109·5 *L* 18·18 *p* 1·89 } air *l* 3·07 } pump	0·028	*N* 177·4 *I.h.p.* 48·2 *B.h.p.* 39·45 *K* 1·0 *R* 0·81	4·5	95·5 according to diagrams } 106·3
				N 181·1 *I.h.p.* 39·52 *B.h.p.* 30·17 *K* 1·0 *R* 0·763	6·0	77·3 according to diagrams } 85·0
				N 183·3 *I.h.p.* 25·02 *B.h.p.* 15·26 *K* 1·0 *R* 0·61	12·0	48·2 according to diagrams } 53·3
71	Robey, 14/1/1902. Mathot (73).	*D* 12·32 *A* 119·0 *L* 18·11	0·0272	*N* 177·0 *I.h.p.* 41·7 *B.h.p.* 35·5 *K* 1·0 *R* 0·85	5·0 96 per cent. explosions.	85·0

Consumption per B.H.P. hour.	Details.			Notes.	No.
	Temperatures, deg. Fah.	Pressures, lbs. per square inch.	Thermal Efficiency.		
7	8	9	10	11	
Town gas, 16·6 cub. ft.	—	C. 71 I. 299	—	—	67
Anthracite, 0·895 lb.	—	—	—	—	68
Anthracite, 0·92 lb. Producer gas at 126·5 B.Th.U. per cub. ft. Oil, 0·009 lb. Anthracite, 0·834 lb.	— Water, 132	C. 142 I. 290 C. 142 I. 356	—	Water evaporated per lb. of fuel, 0·071 gallon (·7 lbs.).	69
American petroleum, 0·796 density, at 63·5° F. 0·473 lb. 0·446 lb. 0·567 lb.	—	—	—	—	70
Town gas, 23 cub. ft.	—	C. 59·7 I. 270	—	--	71

No.	Type of Engine. Date of Test. Operator.	Dimensions.	Modulus of Absolute Power $M = \frac{L \times A}{792000}$	Details of Tests.	Modulus of Speed $n = \frac{KN}{B.h.p.}$	Mean Pressure, $Pm = \frac{1}{R} \times \frac{1}{Mn}$ lbs. per square inch.
	1	2	3	4	5	6
72	Electric Construction Co., 8/5/1905. Hal Williams & Bridges.	*D* 12·0 *A* 113·1 *L* 19·0	0·0271	*N* 214·4 *I.h.p.* 43·4 *B.h.p.* 35·6 *K* 1·0 *R* 0·82	6·02	73 5
73	Niel, 22/1/1896. Stapfer.	*D* 11·81 *A* 109·5 *L* 18·9	0·0261	*N* 178·0 *I.h.p.* 27·5 *B.h.p.* 22·0 *K* 1·0 *R* 0·80	8·09 93 per cent. explosions.	58·5
74	Winterthur, 3/11/1902. Mathot (96).	*D* 12·2 *A* 116·8 *L* 17·7	0·0261	*N* 221·0 *I.h.p.* 39·0 *B.h.p.* 31·2 *K* 1·0 *R* 0·80	7·08 82 per cent. explosions.	66·7 according to diagrams } 82·7
75	Deutz, 22/2/1904. Mathot (179).	*D* 12·6 *A* 124·5 *L* 16·53	0·0260	*N* 196·3 *I.h.p.* 32·9 *B.h.p.* 26·3 *K* 1·0 *R* 0·80	7·46	63·5
76	Stockport, 15/12/1902. Mathot (100).	*D* 12·0 *A* 113·1 *L* 18·0	0·0257	*N* 196·0 *I.h.p.* 42·3 *B.h.p.* 34·7 *K* 1·0 *R* 0·82	5·64	82 5
77	Schmitz, 7/11/1903. Mathot (162).	*D* 11·85 *A* 110·0 *L* 18·0	0·025	*N* 200·0 *I.h.p.* 34·4 *B.h.p.* 26·25 *K* 1·0 *R* 0·79 *N* 200·0 *B.h.p.* 28·76	6·95	71·5
78	Fetu, 10/10/1901. Mathot (61).	*D* 11·0 *A* 95·0 *L* 20·6	0·0247	*N* 200·0 *I.h.p.* 31·2 *B.h.p.* 25·0 *K* 1·0 *R* 0·80	8·0	65·5

Consumption per B.H.P. hour.	Details.			Notes.	No.
	Temperatures, deg. Fah.	Pressures, lbs. per square inch.	Thermal Efficiency.		
7	8	9	10	11	
Anthracite, 0·805 lb.	—	—	—	—	72
Coal, 1·705 lbs.	—	—	—	—	73
Anthracite, 0·7 lb., at 14,100 B.Th.U. per lb. = 9,870 B.Th.U. Producer gas at 137 B.Th.U. per cub. ft.	—	C. 114 I. 370	0·253	—	74
Producer gas, at 136·5 B.Th.U. per cub. ft.	—	C. 132 I. 210	—	—	75
Town gas, 18 cub. ft.	—	C. 99·5 I. 387	—	—	76
Anthracite, 1·055 lbs. Town gas, 17·7 cub. ft. at 565 B.Th.U. per cub. ft. = 10,000 B.Th.U.	—	C. 159 I. 306 C. 166·5 I. 363	— 0·250	—	77
Town gas, 25·15 cub. ft.	—	C. 71 I. 220	-	—	78

No.	Type of Engine. Date of Test. Operator.	Dimensions.	Modulus of Absolute Power $M = \frac{L \times A}{792000}$	Details of Tests.	Modulus of Speed $n = \frac{KN}{B.h.p.}$	Mean Pressure $Pm = \frac{1}{R} \times \frac{1}{Mn}$ lbs. per square inch.
	1	2	3	4	5	6
79	Winterthur, ?/10/1903. Mathot (160).	D 11·8 A 109·3 L 17·7	0·0245	N 200·0 $I.h.p.$ 38·8 $B.h.p.$ 31·2 K 1·0 R 0·81	6·41	77·5 according to diagram } 78·25
80	Benier, ?/12/1894. Witz.	D 11·8 A 109·3 L 17·3	0·0239	N 147·18 $B.h.p.$ 14·79 K 2·0	20·02	
				N 150·79 $I.h.p.$ 27·6 $B.h.p.$ 14·59 K 2·0 R 0·53	20·66	37·7
81	Koerting, 9/11/1903. Mathot (164).	D 10·82 A 92·0 L 20·5	0·0237	N 190·0 $I.h.p.$ 27·0 $B.h.p.$ 21·6 K 1·0 R 0·80	8·79	59·0
82	Tangye, 29/9/1905. Mathot (234).	D 11·0 A 95·0 L 20·0	0·024	N 186·05 $I.h.p.$ 36·8 $B.h.p.$ 29·87 K 1·0 R 0·81	6·22	81·5
83	Charon, 15/2/1898. Lemerle.	D 11·41 A 102·0 L 18·1	0·0234	N 164·5 $I.h.p.$ 17·4 $B.h.p.$ 13·95 K 1·0 R 0·80	11·79	44·7
84	Charon. 17/3/1902. Mathot (75).	D 11·0 A 95·0 L 19·0	0·0228	N 179·0 $I.h.p.$ 15·7 $B.h.p.$ 13·0 K 1·0 R 0·83	13·77	37·8

Consumption per B.H.P. hour.	Details.			Notes.	No.
	Temperatures, deg. Fah.	Pressures, lbs. per square inch.	Thermal Efficiency.		
7	8	9	10	11	
Anthracite, 0·92 lb., at 13,700 B.Th.U. per lb. = 12,700 B.Th.U.	—	C. 152 I. 292	0·201	—	79
Coke, 1,655 lbs., at 12,250 B.Th.U. per lb. = 20,030 B.Th.U. Producer gas at 146·7 B.Th.U. per cub. ft. (high). Oil, 0·0286 lb.	—	—	0·123	—	80
Anthracite, 1·57 lbs., at 14,400 B.Th.U. per lb. = 22,650 B.Th.U. Producer gas at 129·0 B.Th.U. per cub. ft. (high). Water, 12·5 gallons.	Water, 140 Gas, 570	C. 64·5 I. 103·3	0·110	Heat lost to water, 0·265	
Anthracite, 0·805 lb. at 13,450 B.Th.U. per lb. = 10,850 B.Th.U. Oil, 0·0064 lb. Water, 10·2 gallons.	—	C. 57 I. 356	0·230	—	81
Anthracite, 0·7 lb. = 10,500 B.Th.U. Producer gas at 146 B.Th.U. per cub. ft.	Water, 122 Gas, 960	C. 156·5 I. 340	0·238	Water consumption, per lb. of coal, 0·09 gallon (0·9 lbs.). Heat lost to water, 0·339 " " exhaust radiation and generator } 0·420	82
Riché gas, 28·5 cub. ft.	—	—	—	—	83
Anthracite, 0·95 lb. Producer gas at 136 B.Th.U. per cub. ft.	—	C. 105 I. 310	—	—	84

No.	Type of Engine. Date of Test. Operator.	Dimensions.	Modulus of Absolute Power $M=\frac{L \times A}{792000}$	Details of Tests.	Modulus of Speed $n = \frac{KN}{B.h.p.}$	Mean Pressure $Pm = \frac{1}{R} \times \frac{1}{Mn}$ lbs. per square inch.
	1	2	3	4	5	6
85	Winterthur, ?/11/1901. ?	*D* 11·41 *A* 102·0 *L* 17·7	0·0228	*N* 197·5 *I.h.p.* 29·5 *B.h.p.* 23·8 *K* 1·0 *R* 0·80	8·3	65·0
86	Forward, 10/1/1901. Mathot (77).	*D* 10·7 *A* 90·0 *L* 20·11	0·0228	*N* 196·0 *I.h.p.* 45·6 *B.h.p.* 38·7 *K* 1·0 *R* 0·85	5·06	99·5
87	Forward, 1/8/1900. Mathot (37).	*D*, *A*, *L* } Same.	0·0228	*N* 190·0 *I.h.p.* 37·1 *B.h.p.* 29·8 *K* 1·0 *R* 0·80	6·37	89·0
88	Mennig, ?/11/1907. Mathot.	*D* 11·0 *A* 95·0 *L* 19·0	0·0228	*N* 186·0 *I.h.p.* 35·1 *B.h.p.* 28·1 *K* 1·0 *R* 0·80	6·62 85 per cent. explosions.	81·0
				N 195·0 *I.h.p.* 36·6 *B.h.p.* 30·01 *K* 1·0 *R* 0·82	6·5 90 per cent. explosions.	81·0
89	Schmitz, 15/4/1904. Mathot (194).	*D* 11·0 *A* 95·0 *L* 19·0	0·0228	*N* 201·7 *I.h.p.* 29·4 *B.h.p.* 24·4 *K* 1·0 *R* 0·83	8·26	62·5
90	Charon, 3/7/1902. Mathot.	*D* 11·0 *A* 95·0 *L* 19·0	0·0228	*N* 184·0 *I.h.p.* 19·7 *B.h.p.* 15·8 *K* 1·0 *R* 0·80	11·65	46·5
91	Dudbridge, 25/5/1903. Mathot (139).	*D* 11·0 *A* 95·0 *L* 18·6	0·0225	*N* 181·0 *I.h.p.* 40·2 *B.h.p.* 34·2 *K* 1·0 *R* 0·85	5·29 99 per cent. explosions.	97·0

Consumption per B.H.P. hour.	Details.			Notes.	No.
	Temperatures, deg. Fah.	Pressures, lbs. per square inch.	Thermal Efficiency.		
7	8	9	10	11	
Alcohol + 50 per cent. Benzine ; 0·78 lb.	Water, 162	—	—	Magneto ignition.	85
Town gas, 17·5 cub. ft. at 630 (high), 595 (low), B.Th.U. per cub. ft. = 11,050 B.Th.U.	—	—	0·227	—	86
Town gas, 18·25 cub. ft.	—	C. 71 I. 356	—	—	87
Town gas, 15·5 cub. ft. at 32° F.; at 555 B.Th.U. per cub. ft. = 8,570 B.Th.U.	—	C. 92·5	0·295	—	88
Benzol, 0·5 lb. at 17,950 B.Th.U. per lb. = 9,000 B.Th.U.			0·280		
Anthracite, 0·99 lb. Producer gas at 154 B.Th.U. per cub. ft. (high).	—	C. 138 I. 250	—	—	89
—	—	—	—	—	90
Town gas, 18·25 cub. ft. at 590 (high) B.Th.U. per cub. ft. = 10,750 B.Th.U.	—	C. 71 I. 500	0·233	—	91

No.	Type of Engine. Date of Test. Operator.	Dimensions.	Modulus of Absolute Power $M = \frac{L \times A}{792000}$	Details of Tests.	Modulus of Speed $n = \frac{KN}{B.h.p.}$	Mean Pressure $P_m = \frac{1}{R} \times \frac{1}{M_a}$ lbs. per square inch.
	1	2	3	4	5	6
92	Dudbridge. 25/1/1903. Mathot.	*D* 11·0 *A* 95·0 *L* 18·6	0 0225	*N* 196·0 *I.h.p.* 40·8 *B.h.p.* 34·7 *K* 1·0 *R* 0·85	5·65	90·0
93	Robey, 5/4/1900. Mathot (32).	*D* 11·0 *A* 95·0 *L* 18·0	0·0216	*N* 185·0 *I.h.p.* 20·0 *B.h.p.* 16·0 *K* 1·0 *R* 0·80	11·56	49·2
94	Charon, 9/6/1892. Modelski.	*D* 11·0 *A* 95·0 *L* 17·7	0·02125	*N* 142·5 *I.h.p.* 30·33 *B.h.p.* 27·83 *K* 2·0 *R* 0·92	10·24	49·0
95	Hugon. 7/2/1866. Tresca.	*D* 13·0 *A* 132·7 *L* 12·5	0·0211	*N* 53·0 *I.h.p.* 2·57 *B.h.p.* 2·07 *K* 1·0 *R* 0·80	26·08	22·8 according to diagram } 23·1
96	Crossley, 3/12/1893.	*D* 11·5 *A* 103·87 *L* 16·0	0·021	*N* 164·0 *I.h.p.* 20·16 *B.h.p.* 15·22 *K* 1·0 *R* 0·755	10·77 78 per cent. explosions.	58·0 according to diagram } 73·0
97	Winterthur, 30/10/1903. Mathot (159).	*D* 11·0 *A* 95·0 *L* 16·93	0·0204	*N* 204·0 *I.h.p.* 28·1 *B.h.p.* 22·5 *K* 1·0 *R* 0·80	9·06	66·8
98	Globe, 23/2/1901. Mathot (46).	*D* 10·0 *A* 78·54 *L* 20·0	0·0198	*N* 180·0 *I.h.p.* 20·8 *B.h.p.* 16·6 *K* 1·0 *R* 0·80	10·84	57·5
99	Altmann, 1894. Schöttler.	*D* 11·0 *A* 95·0 *L* 15·75	0·0189	*N* 200·0 *I.h.p.* 15·1 *B.h.p.* 12·1 *K* 1·0 *R* 0·80	16·52	38·5

Consumption per B.H.P. hour.	Details.			Notes.	No.
	Temperatures, deg. Fah.	Pressures, lbs. per square inch.	Thermal Efficiency.		
7	8	9	10	11	
Town gas, 15·4 cub. ft.	—	—	—	—	92
Town gas, 22·5 cub. ft.	—	C 57 I. 142	—	—	93
Anthracite, 1·32 lbs. Town gas, 16·2 cub. ft.	—	—	—	—	94
Town gas, 92 cub. ft.	Water, 107·5 Gas, 367·0	— I. 42·7	—	—	95
Coal, 1·865 lbs.	—	—	—	—	96
Mixed gas, 74 cub. ft. at 169 B.Th.U. per cub. ft. = 12,400 B.Th.U.	—	—	0·202	—	97
Petroleum at 810 density = 0·75 lb.	—	C. 28·5 I. 228·0	—	—	98
Petroleum, 0·95 lb.	—	—	—	—	99

No.	Type of Engine Date of Test. Operator. 1	Dimensions. 2	Modulus of Absolute Power $M = \frac{L \times A}{792000}$ 3	Details of Tests. 4	Modulus of Speed $n = \frac{KN}{B.h.p.}$ 5	Mean Pressure $Pm = \frac{1}{R} \times \frac{1}{Mn}$ lbs. per square inch. 6
100	Winterthur, 4/6/1902. Mathot (85).	*D* 10·6 *A* 88·7 *L* 16·	0·0189	*N* 200·0 *I.h.p.* 27·9 *B.h.p.* 22·9 *K* 1·0 *R* 0·82	8·73 96 per cent. explosions.	72·5 according to diagrams 75·4
101	Tangye, 16/1/1903. Mathot (124).	*D* 10·0 *A* 78·54 *L* 19·0	0·01885	*N* 190·6 *I.h.p.* 31·4 *B.h.p.* 27·7 *K* 1·0 *R* 0·88	6·89	86·5
102	Tangye, 23/2/1902. Mathot (74).	*D* 10·0 *A* 78·54 *L* 19·0	0·01885	*N* 218·0 *I.h.p.* 34·7 *B.h.p.* 29·5 *K* 1·0 *R* 0·85	7·39 94 per cent. explosions.	84·0
103	Tangye, 7/4/1902. Witz.	*D* 10·0 *A* 78·54 *L* 19·0	0·01885	*N* 201·5 *I.h.p.* 36·7 *B.h.p.* 32·16 *K* 1·0 *R* 0·87	6·26 99 per cent. explosions.	96·7
104	Tangye, 26/9/1905. Mathot (232).	*D* 10·0 *A* 78·54 *L* 19·0	0·01885	*N* 197·1 *I.h.p.* 23·8 *B.h.p.* 20·1 *K* 1·0 *R* 0·838	9·85 84·9 per cent. explosions.	63·6 according to diagrams 75·0
				N 197·6 *I.h.p.* 14·9 *B.h.p.* 10·9 *K* 1·0 *R* 0·727	18·1 Half load. 54·5 per cent. explosions.	40·0 according to diagrams 73·0
105	Tangye, 31/1/1905. Mathot.	*D* 10·82 *A* 92·0 *L* 20·5	0·0238	*N* 190·0 *I.h.p.* 27·0 *B.h.p.* 21·6 *K* 1·0 *R* 0·80	8·74	76·0
106	Tangye, 5/10/1901. Mathot.	*D* 10·0 *A* 78·54 *L* 18·0	0·01785	*N* 180·0 *I.h.p.* 20·5 *B.h.p.* 16·4 *K* 1·0 *R* 0·80	10·97	63·0

Consumption per B.H.P. hour.	Details.			Notes.	No.
	Temperatures, deg. Fah.	Pressures, lbs. per square inch.	Thermal Efficiency.		
7	8	9	10	11	
Town gas, 13·6 cub. ft. at 595 B.Th.U. per cub. ft. = 8,100 B.Th.U.	—	—	0·309	No load consumption, 92·5 cub. ft. „ explosions, 22 per cent.	100
Town gas, 17·8 cub. ft. at 640 (high) B.Th.U. per cub. ft. = 11,450 B.Th.U.	—	C. 85·5 I. 313·0	0·219	—	101
Town gas, 15·4 cub. ft.	—	C. 78·0 I. 306·0	—		102
Town gas, 17·2 cub. ft., 605 B.Th.U. per cub. ft. (high) = 10,400 B.Th.U.	Water, 149 Gas, 970	C. 100·0 I. 374·0	0·241	Heat lost to water, 0·353 „ „ exhaust, 0·230 „ „ friction, 0·136 „ „ radiation, 0·042	103
Anthracite, 0·8 lb. = 11,000 B.Th.U. Anthracite, 0·92 lb. = 13,050 B.Th.U.	Water, 115 Water, 95	—	0·231 0·192	0·138 gallons, full load } Water evaporated per lb. of fuel. 0·152 „ half load } Heat lost to water, 0·304, full; 0·217, half. Heat lost to exhaust } „ „ radiation } 0·467, full, 0·590, half. „ „ generator }	104
—	—	—	—	—	105
Town gas, 21·5 cub. ft.	—	—	—	—	106

No.	Type of Engine. Date of Test. Operator.	Dimensions.	Modulus of Absolute Power $M = \frac{L \times A}{792000}$	Details of Tests.	Modulus of Speed $n = \frac{KN}{B.h.p.}$	Mean Pressure $P_m = \frac{1}{K} \times \frac{1}{Mn}$ lbs. per square inch.
	1	2	3	4	5	6
107	Stockport, 30/7/1906. Mathot.	*D* 10·0 *A* 78·54 *L* 18·0	0·01785	*N* 220·0 *I.h.p.* 29·4 *B.h.p.* 23·32 *K* 1·0 *R* 0·80	9·43 87 per cent. explosions.	73·0
				N 220·0 *I.h.p.* 33·8 *B.h.p.* 27·72 *K* 1·0 *R* 0·82	7·93	85·0
108	National, 7/8/1903. Mathot (152).	*D* 10·0 *A* 78·54 *L* 18·0	0·01785	*N* 180·0 *I.h.p.* 24·0 *B.h.p.* 18·36 *K* 1·0 *R* 0·76	9·8 98 per cent. explosions.	84·0
109	Schmitz, 28/5/1907. Mathot (259).	*D* 10·23 *A* 82·5 *L* 16·5	0·0171	*N* 206·0 *I.h.p.* 22·25 *B.h.p.* 17·3 *K* 1·0 *R* 0·78	11·9	62·25
				N 205·0 *I.h.p.* 24·75 *B.h.p.* 19·8 *K* 1·0 *R* 0·80	10·35 Max. load.	69·5
110	Stockport, 11/4/1904. Mathot.	*D* 9·75 *A* 74·66 *L* 17·91	0·0169	*N* 208·0 *I.h.p.* 31·2 *B.h.p.* 25·0 *K* 1·0 *R* 0·80	8·32	87·5
111	Dudbridge, 13/1/1903. Mathot (127).	*D* 9·75 *A* 74·66 *L* 17·5	0·0166	*N* 180·0 *I.h.p.* 26·3 *B.h.p.* 22·4 *K* 1·0 *R* 0·85	8·03	88·0
112	Banki, 1899. Mathot.	*D* 9·84 *A* 76·0 *L* 15·75	0·0151	*N* 214·0 *I.h.p.* 32·12 *B.h.p.* 25·7 *K* 1·0 *R* 0·80	8·36 82 per cent. explosions.	98·25 according to diagrams

Consumption per B.H.P. hour.	Details.			Notes.	No.
	Temperatures, deg. Fah.	Pressures, lbs. per square inch.	Thermal Efficiency.		
7	8	9	10	11	
Anthracite, 0·82 lb. Producer gas at 153·5 B.Th.U. per cub. ft. Town gas, 17·7 cub. ft.	—	—	—	—	107
Anthracite, 1·06 lbs. at 13,900 B.Th.U. per lb. = 14,750 B.Th.U. Producer gas at 120 B.Th.U. per cub. ft.	—	C. 86·5 I. 292·0	0·17	—	108
Anthracite, 1·06 lbs.	—	—	—	—	109
Town gas, 18·4 cub. ft.	—	—	—	—	110
Town gas, 16·0 cub. ft.	—	C. 87·0 I. 364·0	—	—	111
Benzine, 0·700 density, 0·508 lb.	—	C. 284·0 I. 370·0		Benzine consumption: 19·8 B.H.P. = 0·537 lb.; 13·93 " = 0·60 "; 6·72 " = 0·86 "; no load = 3·125 "	112

No.	Type of Engine. Date of Test. Operator.	Dimensions.	Modulus of Absolute Power $M = \frac{L \times A}{792000}$	Details of Tests.	Modulus of Speed $n = \frac{KN}{B.h.p.}$	Mean Pressure $Pm = \frac{1}{R} \times \frac{1}{Mn}$ lbs. per square inch.
	1	2	3	4	5	6
113	Güldner, ? Schrötter.	*D* 9·84 *A* 76·0 *L* 15·75	0·0151	*N* 210·7 *I.h.p.* 35·9 *K* 1·0		110·0
				N 214·5 *I.h.p.* 37·7	Max. load.	114·5
				N 213·9 *I.h.p.* 21·0	Half load.	64·0
				N 220·8 *I.h.p.* 44·8 *B.h.p.* 35·6 *K* 1·0 *R* 0·80	6·2	132 (?) deduced from figures given by Schrötter, but extremely high.
114	Diesel, 17/2/1897. Schrötter.	*D* 9·84 *A* 76·0 *L* 15·67	0·01505	*N* 171·8 *I.h.p.* 27·85 *B.h.p.* 19·87 *K* 1·0 *R* 0·72	8·64	105·0
115	Hornsby, 21/6/1894. Capper.	*D* 10·0 *A* 78·54 *L* 15·0	0·0149	*N* 230·0 *I.h.p.* 10·2 *B.h.p.* 8·47 *K* 1·0 *R* 0·83	27·15	29·5
116	Otto-Deutz, 1895. Meyer.	*D* 10·63 *A* 88·5 *L* 14·5	0·0162	*N* 177·0 *I.h.p.* 17·21 *B.h.p.* 14·56 *K* 1·0 *R* 0·85	12·15 99 per cent. explosions.	65·0 according to diagrams } 65·0
117	Brouhot & Co., 1902.	*D* 9·45 *A* 70·0 *L* 15·75	0·0139	*N* 203·0 *I.h.p.* 20·4 *B.h.p.* 16·34 *K* 1·0 *R* 0·80	12·42	68·0
118	Acme, 1904. Mathot.	*D* 9·5 *A* 70·88 *L* 16·0	0·0144	*N* 204·0 *I.h.p.* 10·3 *B.h.p.* 8·3 *K* 1·0 *R* 0·80	24·57	35·0

Consumption per B.H.P. hour.	Details.			Notes.	No.
	Temperatures, deg. Fah.	Pressures, lbs. per square inch.	Thermal Efficiency.		
7	8	9	10	11	
Town gas 10·5 cub. ft. at 560 B.Th.U. per cub. ft. = 5,920 B.Th.U. per I.H.P., 11·5 cub. ft. = 6,470 B.Th.U. per I.H.P., 13·25 cub. ft. = 7,500 B.Th.U. per I.H.P. Anthracite 0·735 lbs. (half load) 1·11 lbs.	—	C. 114 C. 156	0·427 0·390 0·339	Calculations made on I.H.P.: Heat lost to cooling water = 0·332 „ „ exhaust = 0·241	113
Petroleum 0·789 density 0·545 lb. at 19,800 B.Th.U. per lb., 10,800 B.Th.U. Water, 14·25 gallons.	Water, 72 Gas, 760	—	0·232	Heat lost to water = 0·390. „ „ exhaust = 0·273.	114
Petroleum, 0·824 density, 0·537 lb. per I.H.P., at 18,550 B.Th.U. per lb., 9,970 B.Th.U. per I.H.P.	—	—	0·251	Per I.H.P.	115
Petroleum, 0·82 lb.	Gas, 1,300	—	0·196	Heat lost to water, 0·354 „ „ exhaust, 0·450	116
Alcohol + 50 per cent. benzine = 0·512 lb.	—	—	—	No load consumption, 5·7 lbs.	117
—	—	—	—	—	118

No.	Type of Engine. Date of Test. Operator.	Dimensions.	Modulus of Absolute Power $M = \frac{L \times A}{792000}$	Details of Tests.	Modulus of Speed $n = \frac{KN}{B.h.p.}$	Mean Pressure $Pm = \frac{1}{R} \times \frac{1}{Mn}$ lbs. per square inch.
	1	2	3	4	5	6
119	Société de Vierzon, 1897.	*D* 9·85 *A* 76·0 *L* 15·0	0·0144	*N* 219·0 *I.h.p.* 11·2 *B.h.p.* 9·0 *K* 1·0 *R* 0·80	24·33	35·25
120	Westinghouse, 15/1/1907. Mathot (255).	*D* 11·0 *A* 95·0 *L* 12·0	0·0144	*N* 296·0 *I.h.p.* 87·75 *B.h.p.* 70·2 *K* 3·0 *R* 0·80	12·8	67·0
121	Brouhot, 1901.	*D* 9·5 *A* 70·88 *L* 15·75	0·0141	*N* 180·7 *I.h.p.* 20·1 *B.h.p.* 16·11 *K* 1·0 *R* 0·80	11·21	79·0
122	Charon, 7/4/1904. Coste.	*D* 9·0 *A* 63·6 *L* 16·5	0·01325	*N* 160·0 *I.h.p.* 25·1 *B.h.p.* 20·09 *K* 1·0 *R* 0·80	8·0	114·5
123	Bollinckx, 12/8/1905. Mathot (228).	*D* 9·0 *A* 63·6 *L* 16·5	0·01325	*N* 216·0 *I.h.p.* 18·87 *B.h.p.* 15·1 *K* 1·0 *R* 0·80	14·3	64·0
124	Catteau, 1903. Witz.	*D* 9·0 *A* 63·6 *L* 18·0	0·01445	*N* 206·7 *I.h.p.* 21·0 *B.h.p.* 17·48 *K* 1·0 *R* 0·82	11·82	75·7
125	Crossley, 1892. Capper.	*D* 8·5 *A* 56·745 *L* 18·0	0·0129	*N* 162·5 *I.h.p.* 13·32 *B.h.p.* 11·33 *K* 1·0 *R* 0·80	14·34	67·5 according to diagrams } 72·5
126	Crossley, 1895. Capper.	*D* 8·5 *A* 56·745 *L* 18·0	0·0129	*N* 172·1 *I.h.p.* 12·97 *B.h.p.* 11·62 *K* 1·0 *R* 0·80	14·81	65·0

Consumption per B.H.P. hour.	Details.			Notes.	No.
	Temperatures, deg. Fah.	Pressures, lbs. per square inch.	Thermal Efficiency.		
7	8	9	10	11	
Petroleum, 0·885 lbs.	—	—	—	—	119
—	—	—	—	Three-cylinder vertical engine.	120
Alcohol + 50 per cent. benzine = 0·84 lb.	—	—	—	Tube ignition.	121
Town gas, 17·4 cub. ft.	—	—	—	—	122
—	—	C. 142 I. 370	—	—	123
Town gas, 13 cub. ft. at 650 B.Th.U. per cub. ft. = 8,500 B.Th.U.	—	C. 185	0·207	Electric ignition.	124
Town gas, 24 cub. ft.	Water, 140	I. 226	0·228	Heat lost to water, 0·389 " " exhaust, 0·405	125
Town gas, 25·8 cub. ft. at 600 B.Th.U. per cub.ft.(low), = 15,500 B.Th.U.	—	—	0·161	Heat lost to water, 0·443 (including friction). Heat lost to exhaust, 0·347	126

No.	Type of Engine. Date of Test. Operator.	Dimensions.	Modulus of Absolute Power $M=\frac{L \times A}{792000}$	Details of Tests.	Modulus of Speed $n = \frac{KN}{B.h.p.}$	Mean Pressure $Pm = \frac{1}{R} \times \frac{1}{Mn}$ lbs. per square inch.
	1	2	3	4	5	6
127	Lenoir, 24/5/1890. Hirsch.	*D* 9·0 *A* 63·6 *L* 15·75	0·01265	*N* 160·0 *I.h.p.* 18·8 *B.h.p.* 16·13 *K* 2·0 *R* 0·80	19·83	48·7
128	Altmann, 1894. Schöttler.	*D* 9·0 *A* 63·6 *L* 15·75	0·01265	*N* 180·0 *I.h.p.* 10·2 *B.h.p.* 8·16 *K* 1·0 *R* 0·80	22·18	43·5
129	Midland, 10/11/1901. Mathot.	*D* 9·0 *A* 63·6 *L* 16·0	0·01285	*N* 220·0 *I.h.p.* 16·3 *B.h.p.* 13·0 *K* 1·0 *R* 0·80	16·92	57·5
130	Stockport, 10/5/1902. Mathot (82).	*D* 9·0 *A* 63·6 *L* 15·0	0·0120	*N* 202·0 *I.h.p.* 23·3 *B.h.p.* 18·6 *K* 1·0 *R* 0·80	10·86 88 per cent. explosions.	93·0
131	Hille, ? /7/1897. ?	*D* 8·66 *A* 59·0 *L* 15·75	0·0117	*N* 240·0 *I.h.p.* 11·4 *B.h.p.* 9·1 *K* 1·0 *R* 0·80	26·37	40·0
132	Gorton, 2/3/1901. Mathot (47).	*D* 9·5 *A* 70·88 *L* 13·0	0·0116	*N* 300·0 *I.h.p.* 22·8 *B.h.p.* 18·3 *K* 1·0 *R* 0·80	16·39	65·0
133	Bollinckx, 20/6/1904. Mathot (202).	*D* 8·25 *A* 53·45 *L* 16·5	0·0111	*N* 216·0 *B.h.p.* 16·06 *K* 1·0	13·44	—
	17/12/1903. Mathot (178).	Same.		*N* 218·0 *I.h.p.* 16·07 *B.h.p.* 12·2 *K* 1·0 *R* 0·73	17·87 90·7 per cent. explosions.	67·5 according to diagrams } 74·0

Consumption per B.H.P. hour.	Details.			Notes.	No.
	Temperatures, deg. Fah.	Pressures, lbs. per square inch.	Thermal Efficiency.		
7	8	9	10	11	
Town gas, 21·2 cub. ft. at 596 B.Th.U. per cub. ft., 12,620 B.Th.U.	—	C. 35·6 I. 171	0·198	—	127
Petroleum, 0·797 density, 0·83 lb. at 19,350 B.Th.U. per lb. = 16,150 B.Th.U.	—	—	0·155	—	128
Town gas, 22·4 cub. ft.	—	—	—	—	129
Town gas at 32° F., 20·25 cub. ft.	—	C. 121 I. 435	—	—	130
Petroleum, 1·25 lbs.	—	—	—	—	131
—	—	C. 51 I. 200	—	—	132
Town gas, 18·7 cub. ft. at 587 (high), 538 (low), B.Th.U. per cub. ft. = 11,000 B.Th.U. Anthracite, 0·99 lb. Producer gas, 127 (high) B.Th.U. per cub. ft.	—	— C. 150 I. 343	0·230	—	133

No.	Type of Engine. Date of Test. Operator. 1	Dimensions. 2	Modulus of Absolute Power $M = \frac{L \times A}{792000}$ 3	Details of Tests. 4	Modulus of Speed $n = \frac{KN}{B.h.p.}$ 5	Mean Pressure $Pm = \frac{1}{R} \times \frac{1}{Mn}$ lbs. per square inch. 6
134	Tangye, 17/1/1902. Mathot (71).	*D* 8·0 *A* 50·26 *L* 17·0	0·0108	*N* 200·0 *I.h.p.* 17·6 *B.h.p.* 15·1 *K* 1·0 *R* 0·85	13·24 95 per cent. explosions.	81·0
135	Tangye, 28/7/1903. Mathot (151).	*D* 8·0 *A* 50·26 *L* 17·0	0·0108	*N* 200·0 *I.h.p.* 17·8 *B.h.p.* 14·3 *K* 1·0 *R* 0·80	13·98	81·5
136	Tangye, 15/4/1903. Mathot (132).	*D* 8·0 *A* 50·26 *L* 17·0	0·0108	*N* 216·0 *I.h.p.* 15·4 *B.h.p.* 14·0 *K* 1·0 *R* 0·85	15·43 89·2 per cent. explosions.	71·0 according to diagrams } 76·7
137	Tangye, 6/3/1907. Mathot (257).	*D* 8·0 *A* 50·26 *L* 17·0	0·0108	*N* 212·0 *I.h.p.* 20·12 *B.h.p.* 16·10 *K* 1·0 *R* 0·82	13·16 89 per cent. explosions.	85·0
138	Tangye, 26/9/1905. Mathot (230).	*D* 7·0 *A* 38·48 *L* 16·0	0·00777	*N* 225·1 *I.h.p.* 11·25 *B.h.p.* 9·02 *K* 1·0 *R* 0·801	24·9 95·8 per cent. explosions.	71·0 according to diagrams } 74·0
				N 226·6 *I.h.p.* 7·6 *B.h.p.* 5·0 *K* 1·0 *R* 0·658	45·32 half load 65·6 per cent. explosions.	72·5 according to diagrams } 47·25

Consumption per B.H.P. hour.	Details.			Notes.	No.
	Temperatures, deg. Fah.	Pressures, lbs. per square inch.	Thermal Efficiency.		
7	8	9	10	11	
Town gas, 15·5 cub. ft.	—	C. 74 I. 284	—	—	134
Town gas, 17·5 cub. ft.	—	C. 87 I. 377		—	135
Town gas, 20·25 cub. ft.	—	C. 75 I. 334	—	—	136
Town gas, 18·3 cub. ft. at 8·2 *B.h.p.* = 23·0 cub. ft.	—	C. 122 I. 450	—	—	137
Anthracite, 0·855 lb. = 12,200 B.Th.U.	Water, 115	C. 150 I. 350	0·206	0·169 gallons of water evaporated per lb. of coal at full load. 0·160 gallons of water evaporated per lb. of coal at half load.	138
Anthracite, 0·945 lb. = 13,450 B.Th.U.	Water, 94		0·187	Heat lost. — Full load. — Half load. To water . . 0·277 — 0·373 „ exhaust . } „ radiation } 0·516 — 0·440 „ generator . }	

APPENDIX I

BIBLIOGRAPHY

The following is a list of the more important books published in different countries dealing with Internal Combustion Engines:—

Author.	Title.	Date of Publication.	Published by
Allen, Horace .	Gas and Oil Engines .	1907	Scientific Publishing Co., Ltd., Manchester.
,, ,, .	Modern Power Gas Producer Practice	1908	Technical Publishing Co., Ltd., London.
Donkin, Bryan .	Gas, Oil and Air Engines .	1896	C. Griffin & Co., Ltd., London.
Carpenter and Diederichs	Internal Combustion Engines	1908	D. Van Nostrand Co., New York.
Clerk, Dugald .	Gas, Oil and Petrol Engines, Vol. I.	1909	Longmans, Green & Co., London.
Goldingham .	Design and Construction of Oil Engines	1900	E. & H. Spon, Ltd., London.
Grover . .	Modern Gas and Oil Engines	1902	Technical Publishing Co., Ltd., London.
Guldner . .	Calcul et Construction des Moteurs à Combustion	1905	Beranger, Paris.
Haeder, Hermann	Die Gasmotoren . .	1902	Schwann, Duisburg à Rhein.
Hiscox . .	Gas, Gasoline and Oil Engines	1901	Norman Henley & Co., New York.
Hutton, F. R.	The Gas Engine . .	1908	Wiley & Sons, New York.
Jones, F. R. .	The Gas Engine . .	1909	Wiley & Sons, New York.
Lucke, C. E. .	Gas Engine Design . .	1905	Van Nostrand Co., New York.
Marchis, M. L. .	Leçons sur la production et l'utilisation des gaz pauvres	1905—06	Dunod, Paris.
Mathot, R. E. .	Manuel pratique des Moteurs-à-gaz et Gazogenes	1904	Beranger, Paris.
,, ,, .	Moteurs à Combustion Interne et Machines à Vapeur	1907	Beranger, Paris
Moreau, G. . .	Théorie des Moteurs á gaz	1902	Beranger, Paris.
Parsell & Weed .	Gas Engine Construction .	1900	Norman Henley & Co., New York.

Author.	Title.	Date of Publication.	Published by
Poole, C. P.	The Gas Engine	1909	Hill Publishing Co., New York.
Richard, G.	Moteurs à gaz et à petrole	1892—98	Dunod, Paris.
Steen	Gas, Petroleum, en Benzine Motoren	1898	Sijlhoff, Leiden.
Tookey, W. A.	Gas Engines	1903	
	Oil Engines	1904	
	Gas Producers for Power Purposes	1905—08	Percival Marshall & Co , London.
	Gas Engine Manual	1908	
Vermand	Les moteurs à gaz et à pétrole	1894	Ganthier-Villars, Paris.
Vigreux, Ch. et Milandre	Moteurs à gaz Théorie et Pratique		E. Bernard, Paris.
Wimperis, H. E.	The Internal Combustion Engine	1909	A. Constable & Co., London.
Witz, E.	Traité théorique et pratique des Moteurs à Gaz et à Pétrole	1903	Bernard et Cie., Paris.
Wyer, S. S.	Producer Gas and Gas Producers	1907	Hill Publishing Co., New York.

For several years past few works have appeared relating to large gas engines, in France particularly. The question of the utilisation of these large engines has scarcely been touched upon except in the form of magazine articles.

The treatises upon large gas engines by the better known authors are given alphabetically as follows :—

BONTE, A.—Vortrag über der Fortschitten von heutigen gross gasmachinen.

BURSTALL, F. W.—The recent designs of large English Gas Engines. March to April, 1906. *Uhlands Zeitschriften.*

BURSTALL, F. W.—Third report to the Gas Engine Research Committee. *Proceedings, I.M.E.* (London), 1908.

CLARKE, H. ADE.—The Diesel Engine. *I.M.E.* (London), July, 1903.

CLERK, DUGALD.—Limits of Thermal Efficiency in Internal Combustion Motors.

On the specific Heat of, Heat Flow from, and other phenomena of, the working Fluid in the cylinder of the Internal Combustion Engine. *Proceedings of the Royal Society* (London), 1906.

Recent Progress in Large Gas Engines. February, 1905. *Iron and Coal Trades Review* (London).

DESCHAMPS, T.—Les grandes moteurs á gaz (*Revue de Mecanique*).

DIEDERICHS, H.—Some notes on Gas Engines. *Gas and Oil Power* (London). 1907.

FERNALD, R. H.—Report on trials of Gas Engines and Producers, 1902. *Geological Survey.* (New York.)

GREINER, LEON.—Production economique de la force matrice dans les Usines. *Metallurgiques* (Le Souvier. Paris), 1907.

HOPKINSON, B.—The effect of mixture strengths and scavenging upon Thermal Efficiency of Internal Combustion Engines. *Proceedings, I.M.E.* (London). 1908.

HOPKINSON, B.—The effect of size and speed upon the performance of an Internal Combustion Engine. *Institution of Automobile Engineers* (London), 1909.
HUBERT, H.—The construction of Blast Furnace Gas Engines in Belgium. *Iron and Steel Institute* (London), July, 1906.
HUMPHREY, H. A.—Power Gas and Large Gas Engines for Central Stations. *Institute of Mechanical Engineers*, 1904. London.
JUNGE, F. E.—Evolution of Gas Power. New York. January, 1907. *Power.*
LETOMBE, L.—Comparison entre les machines à vapeur et les moteurs à gaz de grande puissance. September, 1906. *Electro.*
MEYER, E.—Uber gross gasmaschinen. February, 1905. *Stahl und Eisen.*
REINHARDT, K.—The Application of Large Gas Engines in the German Iron and Steel Industries. *Iron and Steel Institute* (London), 1906.
RIEDLER, A.—Gross gasmaschinen. May, 1905.
RIGBY, THOMAS.—Power Gas Plants and some of their uses. *Manchester Association of Engineers* (Manchester), June, 1905.
WESTGARTH, TOM.—Notes on Large Gas Engines built in Great Britain, and upon gas-cleaning. *Iron and Steel Institute* (London), 1906.
WITZ, A.—Les Moteurs à gaz à double effet. *L'Eclairage Electrique.* 11th annee, t. xli.

Periodicals specially dealing with the subjects of Gas Power are as follows:—

IN ENGLAND.

Gas and Oil Power, 3, Oxford Court, Cannon Street, London.

Gas and Oil Engine Record, recently incorporated with the *Engineer-in-Charge*, 26 to 29, Poppin's Court, Fleet Street, London.

IN GERMANY.

A similar journal is published under the title *Die Gasmotorentechnik'* (*Fasanenstrasse* 29), Berlin, W.

IN AMERICA.

Gas Power (St. Joseph, Michigan, U.S.A.), and *The Gas Engine* (Cincinnati, Ohio, U.S.A.).

The author has been privileged to discuss various topics in connection with the Internal Combustion Engine before a number of scientific or technical Institutions, amongst which may be mentioned the following:—

To the *Association of Engineers* in connection with the University of Liège, he presented a paper in March, 1904, on the subject of "Modern Gas Engines and their Method of Fuel Supply."

To the (British) *Institution of Mechanical Engineers* at a meeting held at Liège in June, 1905, a paper was read by the author, and afterwards fully discussed, upon "The Growth of Large Gas Engines on the Continent."

At the same period, also at Liège, during one of the Sessions of the *International Congress of Mines, Metallurgy, Machinery and Geology*, another treatise was presented by the author dealing with the "Method of Regulation, Cycles and Construction of Internal Combustion Engines."

APPENDIX II

GAS ENGINE AND GAS PRODUCER MAKERS

America.

Advance Manufacturing Co. (The)	Hamilton (Ohio).
Alamo Manufacturing Co. (The)	Hillsdale (Mich.).
Allis Chalmers Co. (The)	Milwaukee (Wisc.).
American Motor Co.	Brockton (Mass.).
Backus Water Motor Co.	Newark (N. J.).
Bates & Edmonds Motor Co.	Lansing (Mich.).
Bauer Machine Works	Kansas City (Mo.).
Beilfuss Motor Co. (The)	Lansing (Mich.).
Blaisdell Machinery Co.	Bradford (Pa.).
Brown Cochrane Co. (The)	Lorain (Ohio).
Callahan & Co	Dayton (Ohio).
Canada Foundry Co.	Toronto (Ont.).
Capitol Gas Engine Co.	Indianapolis (Ind.).
Clay, E. H. & Co.	Cleveland (Ohio).
Clifton Motor Works (The)	Cincinnati (Ohio).
Colombus Machine Co.	Colombus (Ohio).
Defiance Ironworks (The)	Chatham (Ont.).
De La Vergne Refrigerating Machine Co.	New York (N. Y.).
De Mooy Bros.	Cleveland (Ohio).
Detroit Engine Works	Detroit (Mich.).
Dissinger, C. H. A. & Bros.	Wrightsville (Pa.).
Du Bois Iron Works	Dubois (Pa.).
Eagle Manufacturing Co.	Appleton (Wisc.).
Elyria Gas Engine Co. (The)	Elyria (Ohio).
Fairbanks Co.	New York City.
Fairbanks Co. (Canadian)	Montreal (Que.).
Fairbanks Morse Co.	Chicago (Ill.).
Falls Machine Co.	Sheboygan Falls (Wisc.).
Foos Gas Engine	Springfield (Ohio).
Fort Wayne Foundry and Machine Co.	Fort Wayne (Ind.).
Gardner Gas Engine Co.	Washington (Pa.)
Geiser Manufacturing Co. (The)	Waynesboro (Pa.).
Gibbs Gas Engine Co. (The)	Atlanta (Ga.).
Gibson Manufacturing Co.	Port Washington (Wisc.).
Globe Iron Works	Minneapolis (Minn.).
Goldie Machine Co. (The)	Galt (Ont.).
Goold Shapley and Muir Co.	Brantfort (Ont.).

Gray Motor Co. (The)	Detroit (Mich.).
Hall Rittenhouse Co. (The)	Bucyrus (Ohio).
Hamilton Motor Works, Ltd.	Hamilton (Ont.).
Hartig Standard Gas Engine Co.	Newark (N. J.).
Havana Manufacturing Co.	Havana (Ill.).
Holliday Engineering Co.	Chicago (Ill.).
Ideal Manufacturing Co.	Portsmouth (Ohio).
International Harvester Co.	Chicago (Ill.).
International Oil Engine Co.	Danilson (Conn.).
Jacobson Machine Manufacturing Co.	Atlanta (Ga.).
Kenton Gas Engine Co.	Kenton (Ohio).
Lambert Gas and Gasoline Engine Co.	Anderson (Ind.).
Lammert and Mann	Chicago (Ill.).
Lauson C. P. and J. Co.	Milwaukee (Wisc.).
Lowe Bros. Machine Works	Colombus (Ohio).
Mesta Machine Co.	Pittsburg (Pa.).
Middletown Machine Co.	Middletown (Ohio).
Miller Improved Gas Engine Co. (The)	Springfield (Ohio).
Minneapolis Steel and Machinery Co.	Minneapolis (Minn.)
Model Gas Engine Works	New York (N. Y.).
National Meter Co.	New York (N. Y.).
New-Way Motor Co.	Lansing (Mich.).
North Chicago Machine Co. (The)	North Chicago (Ill.)
Ohio Motor Co. (The)	Sandusky (Ohio).
Olds Gas Power Co.	Lansing (Mich.).
Ontario Wind Engine and Pump Co.	Toronto (Ont.).
Palmer Bros.	Cos-Cob (Conn.).
Peerless Motor Co.	Lansing (Mich.).
Pohl Geo., Manufacturing Co.	Vernon (N. Y.).
Riverside Engine Co.	Oil City (Pa.).
Roberts Motor Co.	Sandusky (Ohio).
Rumely Gas Engine Co.	La Porte (Ind.).
St. Mary's Machine Co. (The)	St. Mary's (Ohio).
Shelbyville Foundry and Machine Works	Shelbyville (Ind.).
Smart Turner Machine Co.	Hamilton (Ont.).
Smith Gas Power Co.	Lexington (Ohio).
Springfield Gas Engine Co.	Springfield (Ohio).
Standard Motor Construction Co.	Jersey City (N. J.).
Stickney Co.	St. Paul (Minn.).
Strang Engine Co. (The)	Chicago (Ill.).
Stratford Mill Building Co.	Stratford (Ont.).
Struthers Wells Co.	Warren (Pa.).
The Rathbun Jones Co.	Toledo (Ohio).
Waterloo Gas Engine Co.	West Waterloo (Iowa).
Weber Gas and Gasoline Engine Co.	Kansas City (Mo.).
Wellmann-Seaver-Morgan Co.	Cleveland (Ohio).
Western Gas Construction Co. (The)	Fort Wayne (Ind.).
Western Gas Engine Co.	Los Angelos (Cal.).
Western Malleable and Grey Iron Manufacturing Co.	Milwaukee (Wisc.).
Westinghouse Machine Co. (The)	East Pittsburg (Pa.).

William Kavanaugh Co. (The)	Norwalk (Ohio).
Williamsport Gas Engine Co.	Williamsport (Pa.).
Witte Ironworks Co.	Kansas City (Mo.).
Wood, R. D. & Co.	Philadelphia (Pa.).
Wooley Foundry and Machine Works	Anderson (Ind.).

Austria-Hungary.

Anthon et Sohne	Flensburg.
Ganz et Co.	Budapest.
Erste Brünner Maschinenfabrik	Bes-Brünner.
Skodawerke A. G.	Pilsen.

Belgium.

Carels frères	Gand.
Forges et fonderies (Soc. A^{me}) de	Haine-Saint-Pierre.
Lambotte	Marchienne.
Mennig (Jules)	Mons.
Onghena (H.) (Soc. A^{me} des ateliers)	Gand.
Société John Cockerill	Seraing.
,, des ateliers du Thiriau	La Croyère.
,, des établissements Fétu-Defize	Liège.
,, des Moteurs à Gaz Bollinckx	Huyssinghem.
,, Westinghouse, 51, rue Royale	Bruxelles.
,, des Ateliers de Boussu	Boussu.
Van Eecke, Gheysens et Co.	Courtrai.

England.

Allen, W. H. & Co.	Bedford.
Anderston Foundry Co.	Glasgow.
Ashworth, Elijah	Colleyhurst, Manchester.
Baker, J. & Son	Willesden.
Bates, W. J. & Co.	Denton, Manchester.
Beardmore, Wm. & Co., Ltd.	Dalmuir, Glasgow.
Bennett & Tapley	Southampton.
Blackstone & Co., Ltd.	Stamford.
Boby, R., Ltd.	Bury St. Edmunds.
Britannia Co.	Colchester.
British Westinghouse Electric and Manufacturing Co., Ltd.	Trafford Park, Manchester.
Brown and May, Ltd.	Devizes.
Campbell Gas Engine Co., Ltd.	Halifax, England.
Capel & Co., Ltd.	Dalston, London.
Crossley Bros., Ltd.	Openshaw, Manchester.
Cundall Gas Engine Co., Ltd.	Keighley, Yorks.
Davey, Paxman & Co., Ltd.	Colchester.
Diesel Engine Co., Ltd.	London.

Dowson Economic Gas and Power Co.	39, Victoria Street, W.
Dudbridge Ironworks, Ltd.	Stroud, Glos.
Duff Bros. & Co.	Liverpool.
Dunstan Engine Works Co.	Dunstan-on-Tyne.
Electric Construction Co., Ltd.	Wolverhampton.
Fairbanks, Brierley & Co.	Shipley, Yorks.
Fielding & Platt, Ltd.	Gloucester.
Fraser and Chalmers, Ltd.	Erith, London.
Galloways, Ltd.	Manchester.
Gardner, L. & Sons, Ltd.	Patricroft, Manchester.
Grices' Gas Engine Co., Ltd.	Carnoustie and Birmingham.
Griffin Engineering Co., Ltd.	Bath.
Hardy and Badmore, Ltd.	Worcester.
Haslam Foundry Engineering Co., Ltd.	Derby.
Hindley & Sons	Bourton, Dorset.
Hopkinson Gas Plant Co.	Huddersfield.
Hornsby, R. & Sons, Ltd.	Grantham.
Horsehay Co., Ltd.	Horsehay, R.S.O., Shropshire.
Howard, J. & F.	Bedford.
Kynoch, Ltd.	Birmingham.
Lilleshall Forge Co., Ltd.	Lilleshall.
Marshall Sons & Co., Ltd.	Gainsborough.
Mason's Gas Power Co.	Levenshulme, Manchester.
Mather & Platt, Ltd.	Salford, Manchester.
McLaren Bros.	Dumbarton, N.B.
Mirrlees, Bickerton & Day, Ltd.	Hazel Grove, Stockport.
Moses Mellor & Co., Ltd.	Nottingham.
Munro, Allan & Co.	Glasgow.
National Gas Engine Co., Ltd.	Ashton-under-Lyne.
Newton Bros.	Derby.
Petter, J. B. & Sons, Ltd.	Yeovil.
Pitt Engineering Co.	London.
Power Gas Corporation, Ltd.	Stockton-on-Tees.
Premier Gas Engine Co.	Sandiacre.
Railway and General Engineering Co.	Nottingham.
Reavell & Co.	Ipswich.
Richardsons, Westgarth & Co.	Middlesboro'.
Robey & Co.	Lincoln.
Robson, J.	Shipley, Yorks.
Rodger & Co.	Glasgow.
Rowland, B. R. & Co.	Reddish, near Manchester.
Ruston, Proctor & Co., Ltd.	Lincoln.
Salmon, Whitfield & Co.	Kettering.
Schofield and Mitchell	Keighley, Yorks.
Sheffield Engineers, Ltd.	Wellington Street, Sheffield.
Simpson, Strickland & Co., Ltd.	Dartmouth, England.
Smart and Brown	Erith, Kent.
Stewart & Co.	Scotstoun, Glasgow.
Tangyes Ltd.	Birmingham.
Teasdale Bros.	Darlington.

Thornycroft, J. J. & Co.	Chiswick, London.
Willans & Robinson, Ltd.	Rugby.
Willock, Reed & Co., Ltd.	Hope Street, Glasgow.

France.

Boutillier et Cie	Orleans.
Brouhot et Cie	Vierzon (Cher.).
Caloin et Marc	Lille (Nord).
Compagnie des moteurs Taylor à gaz pauvre, 16, rue Grange-Batelière	Paris.
Compagnie Duplex	Ferrière-la-Grande.
Delion et Lepeu	Paris.
Fichet et Heurtey	Paris.
Garnier, E., et Faure Beaulieu	Paris.
Kent, William	Nantes.
Labbé, 208, rue Saint-Maur	Paris.
Lefflaive et Cie	Saint Étienne.
Lerouge, Fornas et Cie 110, rue de Crimée	Paris.
Ludt, 78, Rue de Crimée	Paris.
Mora, 27, Rue des Récollets	Paris.
Piat A. et Cie	Paris.
Pierson et Cie	Paris.
Riché H. Gaz	Paris.
Roux, Paul et Cie, 54, Boulevard du Temple ..	Paris.
Sautter Harle et Cie	Paris.
Schneider et Cie	Le Creusot.
Société des brevets et moteurs Letombe	Paris.
,, de Constructions mécaniques et de Moteurs à Gaz	Valenciennes.
,, de Mécanique Industrielle	Anzin.
,, des chantiers et ateliers A. Hormand ..	Paris.
,, des Moteurs Sabathé	Saint-Étienne.
,, Française de Constructions Mécaniques (anciens établissements Cail)	Denain.
,, Française des Moteurs Diesel	Longeville.
,, Française des Moteurs Otto, rue Croix-Nivert	Paris.
,, Générale des Industries Economiques, 40, rue Laffitte	Paris.
,, Westinghouse	Le Havre.
Thévenin frères, L. Seguin et Cie	Paris.

Germany.

Ansbacher Motorenfabrik Karl Bachmann	Ansbach.
Ascherlebener Maschinenbau Aktiengesellschaft (Moteur Oechelhauser)	Ascherleben.
Benz & Co.	Mannheim.
Braunschweigische Maschinenbau Anstalt	Braunschweig.
Crimmitschauer Maschinenfabrik	Crimmitschau.
Deutsch-Luxemburgische Bergwerks und Hutten Ak. G.	Mulheim-a-Ruhr.

Dinglersche Maschinenfabrik A. G.	Zweibrucken.
Dresdner Gasmotorenfabrik A. G. vorm. Moritz Hille	Dresden.
Duisburger Maschinenbau A. G. Vormals Bechem et Keetman	Duisburg.
Ehrhardt & Sehmer	Schleifmühle.
Elsassische Maschinenbau Gesellschaft	Mulhausen.
Englehardt & Co.	Fürth.
Gasmotoren Fabrik Aktien Gesellschaft	Cöln-Ehrenfeld.
Gasmotoren Fabrik Deutz	Cöln-Deutz.
Goebels (Th.) & Co.	Cöln-Ehrenfeld.
Görlitzer Maschinenbau Akt. Gesell..	Görlitz.
Güldner Motoren Gesellschaft	München-Giesing.
Gütehoffnungshutte Aktienverein für Bergbau und Huttenbetrieb	Oberhausen.
Hallesche Maschinenfabrik und Eisengiesserei ..	Halle a/S.
Haniel & Lueg	Düsseldorf.
Gebrüder Körting Aktiengesellschaft	Hanover.
Kook (Ernest)	Cöln-Ehrenfeld.
Krupp (Friedrich)	Essen a. Ruhr.
Leutert, E.	Halle a/S.
Luther G. Aktiengesellschaft	Brunswick.
Markische Maschinenbau Anstalt (Moteur Cockerill)	Wetter a/Ruhr.
Maschinenbau Act. Ges. (Moteur Union)	Essen-sur-Ruhr.
Maschinenbau A. G. Vorm. Ph. Swiderski	Leipzig.
Maschinenbau A. G. Vorm. gebrüder Klein ..	Dahlbruch (Westph)
Maschinenfabrik Kappel A. G.	Chemitz.
Motorfabrik Rastatt	Rastatt.
Pokorny & Wittekind	Frankfurt a/M.
Scharrer & Gross	Nürnberg.
Scheben & Krudewig	Heneff a.d. Sieg.
Schüchtermann & Kremer	Dortmund.
Siegener Maschinenfabrik A. G.	Siegen.
Theisen (Edourd)	Munich.
Thyssen et Cie	Mülheim a/Ruhr.
Vereinigte Maschinenfabrik Augsburg und Maschinenbau Gesellschaft Nürnberg A. G.	Nuremberg.
Wiedenfeld et Cie	Duisburg.

Holland.

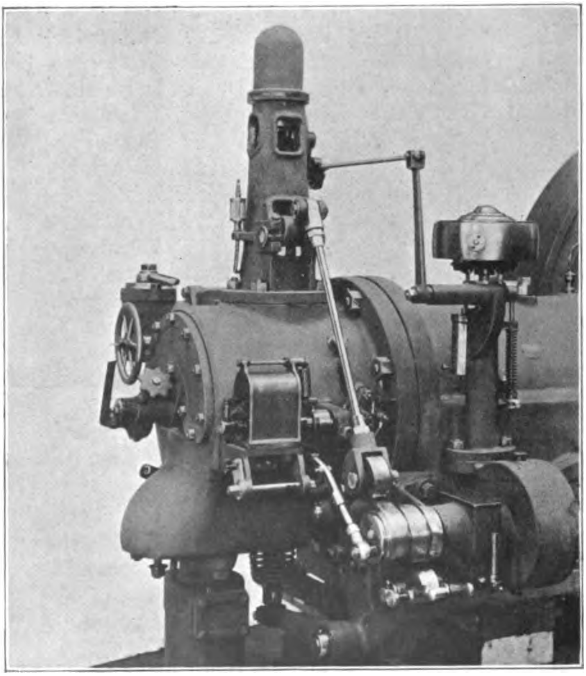
Apeldoornsche Maschinenfabrik en Metaalgieterij	Apeldoorn.
Appingedamer Bronsmotoren fabrik	Appingedam.
Klep Frans	Breda.
Thomassen & Co.	Arnhem.

Switzerland.

Bachtold & Co.	Steckborn.
Gilleron & Amreim	Vevey.
Société Suisse pour la Construction de locomotives et de machines	Winterthur.

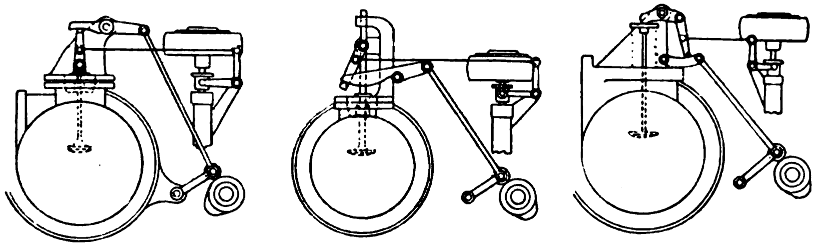
APPENDIX III

RESULTING from his wide experience with numerous engines of varying types, in practical service, as well as under test conditions, the Author has been privileged to assist many gas engine manufacturers, both European and American, by superintending the work of their technical staff when preparing new designs, and by making suggestions for improvements on general lines and also upon points of detail embodying the latest improved principles. In deference to the expressed



wishes of his clients, however, references to such assistance professionally rendered occur only in one or two places in this present volume.

The diagrams here reproduced illustrate a device, patented by the Author in the principal countries, for controlling the speed of internal combustion engines by more or less limiting the degree of opening of the inlet valve under the influence of the governor. The movement of the latter varies the position of a link placed between the operating lever, having a constant stroke, and the valve stem, resulting in the admission of a variable quantity of mixture, the constituents of which are in constant volumetric ratio.



One of the methods in which the invention can be applied is illustrated in the photo reproduction, which represents the Author's patented device as adopted by several well-known British firms.

INDEX

H.

I.

BRADBURY, AGNEW, & CO. LD., PRINTERS, LONDON AND TONBRIDGE.