# INTERNAL-COMBUSTION ENGINEERING

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# INTERNAL - COMBUSTION ENGINEERING

REVISED AND EDITED BY

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# PREFACE

The object of this book is to present a general survey of the construction, applications and operation of all classes of internal-combustion engines, with descriptions of typical engines of each class and their accessories (including gas producers), and it is confidently hoped that in these respects it meets the needs not only of students but also of engineers requiring a reliable work of reference.

Since the previous edition was published developments in internal-combustion engineering have been rapid and extensive in all of its branches, particularly in land, marine and air transport. The chemist, physicist, metallurgist and engineer have all played important parts in these developments. In the present edition, in order to bring the work into line with current practice, the sections on "Aero Engines" and "Oil Engines" have been rewritten and a new section on "Oil-Engines for Road Transport" has been added. The remaining sections have also been brought up to date. In each section of the work emphasis has been laid on fundamental principles governing changes and improvements in design and the trend of future developments has been indicated.

Throughout the work the descriptive matter includes recent developments in design and applications, but it must be realized that such developments are continuous and have accelerated very appreciably quite recently, particularly so far as aero engines are concerned, but since radical changes in design are not involved the matter included in the different sections of the book may be regarded as indicative of recent practice. Much of the complexity in modern engines is apparent only, and a careful study of the aims of the designer combined with a knowledge of the fundamental principles involved will reveal in most cases the skill with which the various components have been designed and constructed to perform their separate functions.

Experience with some engine types which were designed to give certain advantages has shown that there were compensating disadvantages which outweighed these advantages, so that for the present at any rate their manufacture for commercial uses has been abandoned. Reliability in operation, freedom from vibration, accessibility for adjustments and renewal of parts and reasonable simplicity of design are as essential as economy in fuel consumption, since the user is concerned with *total* running costs (including depreciation allowance), and delays due to breakdowns or the necessity for frequent overhauls may overshadow savings in fuel costs during the active life of an engine. These matters have not been overlooked in the discussion of the different types.

The discussion of a number of complex problems, such as those arising from vibrations, fuel distribution in multi-cylinder engines and pressure waves in fuel systems of compression-ignition engines, is beyond the scope of this work, but adequate reference is made to sources of information on these matters.

Acknowledgments are due to the many firms who have generously supplied information, drawings and photographs; also to professional engineering institutions who have given permission for quotations from their Proceedings.

A. T. J. K.

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By G. A. BURLS, M.Inst.C.E.

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## GAS PRODUCERS

By G. A. BURLS, M.Inst.C.E.

Revised by A. T. J. KERSEY, A.R.C.Sc., M.I.Mech.E., M.I.A.E., F.Inst.Fuel



## THE OPERATION OF GAS-ENGINES

By PERCY R. ALLEN, A.M.Inst.C.E., M.I.M.E., A.I.E.E.

Revised by A. T. J. KERSEY, A.R.C.Sc., M.I.Mech.E., M.I.A.E., F.Inst.Fuel



## OIL-ENGINES
## (INCLUDING MARINE OIL-ENGINES)

By JAMES RICHARDSON, B.Sc., M.Inst.C.E., M.I.N.A., and
G. A. BURLS, M.Inst.C.E.

Revised and re-arranged by A. T. J. KERSEY, A.R.C.Sc., M.I.Mech.E.,
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# LIST OF FOLDING PLATES

# GAS-ENGINES

BY

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# Gas-engines

## CHAPTER I

## Introduction

**Historical Summary.**—Space permits of a very brief reference only to the early history of the internal-combustion engine; for a full account the reader may consult Vol. I of Sir D. Clerk's *The Gas, Petrol, and Oil Engine.** Probably the earliest instance of the use in Britain of an explosion to obtain mechanical effect is that of the primitive cannon used by Edward III about A.D. 1327. Three hundred and fifty years later C. Huygens, followed by Papin and d'Hauteville, endeavoured to obtain continuous motion from successive explosions of gunpowder in a cylinder fitted with a piston; and much later still, viz. in 1820, Farish of Cambridge constructed a small engine intended to be driven by gunpowder; this explosive is, however, quite unsuited for use in internal-combustion engines for both thermal and practical reasons.

In 1820 Cecil, also of Cambridge, made what was probably the first actually working gas-engine, using as his fuel an explosive mixture of hydrogen and air; thereafter appeared in succession Brown's " gas-vacuum " engine, Wright's engine, Barnett's engines, and the very singular " free-piston " engines of Barsanti and Matteucci and of Otto and Langen; and in 1860 the once popular, though very uneconomical, Lenoir engine, which may be fairly described as a double-acting steam-engine using a mixture of coal-gas and air in lieu of steam; Hugon (1865) effected improvements in the details of the Lenoir, but its still high fuel consumption caused its abandonment in favour of the much more economical, though mechanically objectionable, Otto and Langen free-piston type. It should be mentioned also here that in 1873 there appeared in America the theoretically efficient " constant pressure " Brayton gas-engine, whose success was prevented by practical difficulties; the modern highly economical Diesel oil-engine illustrates the principle embodied in the Brayton of burning the fuel at (approximately) constant pressure during its admission to the combustion chamber.

The final step in the evolution of the modern four-stroke cycle internal-

* 2 vols., 1913 (Longmans).

combustion engine was made in 1862 when de Rochas first laid down explicitly the procedure to be adopted to obtain maximum economy in the working of gas-engines, comprising suction, compression, explosion and expansion, and exhaust—all performed within the working cylinder.

A four-stroke single-acting engine of the simplest type is illustrated diagrammatically in fig. 1. It includes an open-ended cylinder fitted with a closely fitting piston which drives a crank by means of the usual type of connecting-rod; in the upper end, or " combustion chamber ", of the cylinder a pocket is formed containing the inlet valve A, and exhaust valve B, the latter being forcibly raised at suitable intervals by a cam C and roller-ended tappet-rod D. Suppose the engine to be turning, and the suction stroke about to commence. The piston in its descent creates a partial vacuum above it, the inlet valve A accordingly opens and fresh mixture passes into the cylinder; this is the suction stroke. When the piston has reached the bottom of its stroke and commenced to return the inlet automatically closes and the charge is compressed into the upper part of the cylinder and valve pocket; this is the compression stroke. The mixture is exploded by the ignition plug at, or near, the instant of greatest compression, and the piston is next driven downwards, thus performing the expansion or working stroke. Near the bottom of this stroke the cam C lifts the exhaust valve B, and the burnt gases escape into the atmosphere; B remains open throughout the whole of the succeeding or exhaust stroke; this cycle is repeated indefinitely so long as the engine continues running. The cam-shaft is driven by gearing at half the crank-shaft speed, and is accordingly frequently referred to as the " half-time " shaft. In all modern engines the inlet valve is also cam-operated, an increased volumetric efficiency and increased speed being thus obtained.

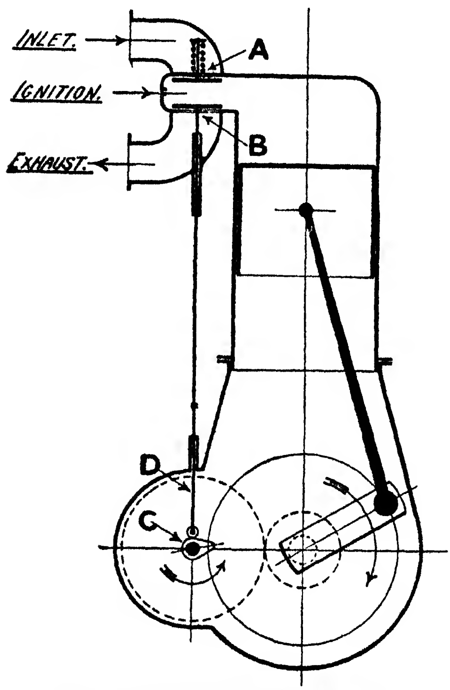

Fig. 1.—Four-stroke Single-acting Engine

**The Otto Silent Gas-engine.**—It was not, however, until 1876 that Dr. Otto produced his world-famous " silent " gas-engine, working exactly upon the four-stroke cycle as laid down by de Rochas fourteen years earlier. Otto realized the de Rochas cycle in a practical and most successful manner and employed flame ignition. The gas consumption of his engines was at once found to be much lower than had ever previously been attained, averaging only from about 24 to 30 c. ft. of coal-gas per brake horse-power hour. The introduction of the "Otto silent gas-engine" marked the beginning of the era of the internal-combustion engine as a prime mover of world-wide importance.

The Otto silent gas-engine was taken up in England by Messrs. Crossley Brothers, Ltd., of Manchester, and progress was rapid. In 1878 a gas-engine of 3 h.p. was regarded as large; in 1881 a 20-h.p. engine was considered as remarkable; in 1898 the largest gas-engine built was of 220 i.h.p. The high cost of coal-gas as a fuel probably restrained the growth of engines, but the possibility of utilizing the hitherto wasted blast-furnace gases for power purposes first demonstrated by Thwaite in 1895, and the evolution of the gas producer by J. E. Dowson and others between 1878 and 1903, provided engineers with suitable gases for power purposes which could be cheaply and easily produced in very great quantities. These discoveries provided the necessary stimulus to the construction of very large engines.

Thus in 1899 the Société Cockerill of Seraing (Belgium) had in operation a single-cylindered single-acting four-stroke engine of 51·2-in. bore and 55·13-in. stroke running on blast-furnace gas at 90 r.p.m., and developing 600 b.h.p.

At the end of 1910 the Nuremberg Company (M.A.N.) had in operation double-acting gas-engines aggregating roundly 450,000 b.h.p., and including two-cylindered tandem units developing 2500 b.h.p. While in 1922 the Premier Gas Engine Co., Ltd., of Sandiacre (Notts.), had designed an eight-cylindered two-crank double-acting four-stroke engine of 6000 h.p. (pp. 50, 51).

**The Two-stroke Cycle.**—In the Otto, or Beau de Rochas, or "four-stroke" cycle the utilization of the same cylinder alternately for pumping and power output, though exceedingly convenient as a simplification in design, introduces the disadvantage that a working impulse is obtained once only in every four strokes of the piston; the crank-shaft speed is consequently far from uniform unless one, or frequently two, very heavy fly-wheels are fitted which by their momentum maintain the rotation only slightly diminished during the exhaust, suction, and compression periods.

Attention was accordingly quickly directed to the problem of increasing the frequency of the impulses, and as early as 1878 Sir D. Clerk produced his first two-stroke cycle engine, quickly improved upon in a second design of which an example was exhibited at the Paris Electrical Exhibition of 1881. This engine is illustrated diagrammatically in fig. 2. A is the power cylinder containing at its outer end exhaust ports E, E′, overrun by the piston C when near the end of its out-stroke. The pump-cylinder B is fitted with a piston D driven by a crank about 90° in advance of the power crank; on its out-stroke D draws into B a charge of mixed gas and air through the sliding valve H and pipe W. When D commences its in-stroke the charge in B becomes slightly compressed; as soon as its pressure exceeds that in the power cylinder A it is delivered into the combustion chamber G through an automatic inlet valve. The power piston next returns, first cutting off the ports E, E′, and next causing the automatic inlet valve to close; the entrapped fresh charge is then compressed into the chamber G, fired, and the working out-stroke follows.

Thus every out-stroke of C is a working stroke, and the engine accordingly gives one impulse every revolution.

The working impulses being thus twice as frequent as in a four-stroke engine, an ideal two-stroke cylinder should develop twice as much power as a four-stroke of the same bore, stroke, and speed. Practically, however, the two-stroke cycle suffers from drawbacks which prevent the realization of this ideal; thus in the four-stroke cycle the inlet valve is opened slightly before the end of the piston stroke, and remains open throughout the whole suction stroke and usually for a short period thereafter; the charging of the cylinder thus continues during about 220° of crank-shaft revolution. On the

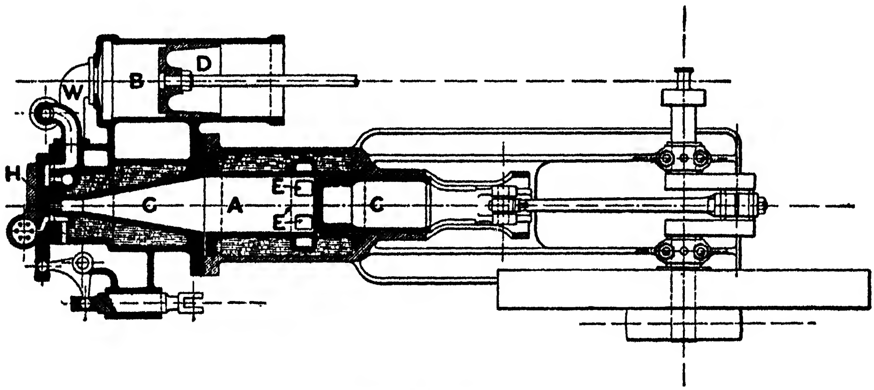


Fig. 2.—Clerk Two-stroke Cycle Engine

other hand, in the two-stroke engine charging must be effected during the short interval elapsing between the uncovering and re-covering of the ports E, E′, which ordinarily occurs in about 80° of crank-shaft revolution; hence the duration of charging in the four-stroke is about three times as great as in the two-stroke engine. Though this drawback is somewhat reduced by providing large inlet area, the two-stroke engine is nevertheless in general incapable of such effective charging as the four-stroke, and more power is also absorbed in the charging operation.*

Again, in the four-stroke engine, the exhaust valve is open during about 240° of crank-shaft revolution, and the burnt gases are moreover positively expelled by the piston during the whole exhaust stroke, the combustion chamber alone remaining filled with burnt gas at, or often slightly below, atmospheric pressure. In the two-stroke the exhaust has to be effected while the crank-shaft turns through about 80°, and the whole cylinder remains filled with burnt gases which, while being assisted in their exit by the incoming fresh charge, heat it, thus reducing its density and diminishing the "volumetric efficiency" of the engine. Further, it frequently happens in small two-stroke

* Hopkinson on "Charging" in *The Engineer*, 31st January, 1914; and Morgan, *The Automobile Engineer*, January, 1923.

engines that there is some loss of fresh charge by direct passage through the exhaust ports; in large engines this is avoided by means described later (pp. 37, 39, 41). Clerk improved the scavenging of his engine and avoided loss of fresh mixture through the exhaust ports by forming his combustion chamber G (fig. 2) in long, conical shape, and admitting the charge under slight pressure at the apex. As early as 1884 his engines were built by Messrs. L. Sterne & Co., Ltd., in sizes ranging from 2 to 12 h.p., running at 200 to 130 r.p.m., and showing for that period considerable economy in fuel consumption, ranging from 29 c. ft. of coal-gas per indicated horse-power hour in the smallest, to about 20 c. ft. in the largest sizes.

The two-stroke engine, with suitable modifications, has long been firmly established, both in the very largest types of gas-engine represented by the

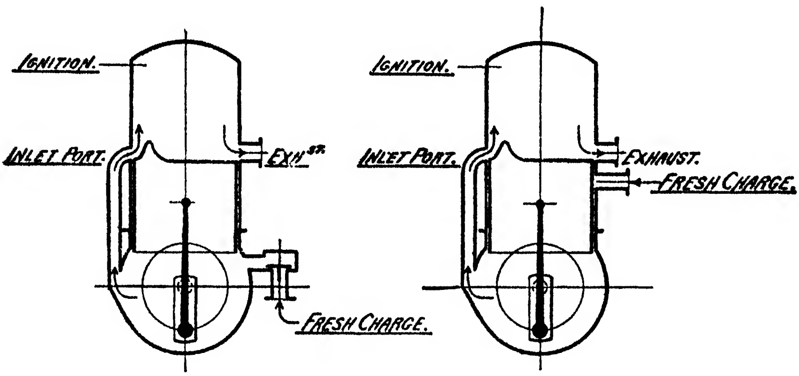


Fig. 3.—The Day Gas-engine

Koerting and Oechelhauser and Diesel designs, and also, in a simplified form next to be described, in immense numbers as the power unit of many motor-boats and of the very popular two-stroke motor-bicycle.

**The Day Gas-engine.**—The very important simplification just referred to was made by Day in 1891, and consisted in using the crank chamber as a charging pump, thus dispensing altogether with the separate pump B of fig. 2.

The Day gas-engine is illustrated diagrammatically in fig. 3 in two forms, that on the left showing the "two-port" type used e.g. in the well-known "Bolinders" marine engine, while the right-hand view illustrates the three-port type as universally employed in motor-cycle applications.

Referring firstly to the two-port type, the crank-chamber is made quite air-tight, and the ascent of the piston accordingly creates in it a partial vacuum causing mixed gas and air to enter through the non-return valve shown; on the descent of the piston this charge is compressed to four or five pounds per square inch pressure. When near the bottom of its stroke the piston first overruns the exhaust port shown, permitting the discharge of the burnt gases, and immediately afterwards uncovers the inlet port, whereupon the compressed fresh charge from the crank-chamber at once enters the cylinder, and assists in the removal of the exhaust gases. The " hump " shown on the piston deflects the entering stream of fresh mixture upwards, and thus minimizes loss by short-circuiting through the exhaust port. The piston next ascends, cuts off the ports, and compresses the entrapped charge into the top part of the cylinder, where it is fired at or about the instant of greatest compression, and the working stroke follows.

The two-port type requires a valve, but the three-port engine is entirely valveless. The ascent of the piston causes a partial vacuum in the crank-chamber as before; when near the top of its stroke, its *lower* edge overruns the port shown, and a charge of gas and air immediately rushes into the crank-chamber. The piston next descending cuts off this charging port, and the mixture is compressed in the crank-chamber, the subsequent action being exactly as in the two-port type. Thus every down-stroke is a working stroke, and in the three-port form the engine is of the simplest possible character, the only moving parts being the piston, connecting-rod, and crank-shaft. It will be observed that these engines will run equally well in either direction; this is a valuable property in marine applications, where ready reversibility is essential, and they are accordingly largely used for the propulsion of motor-launches, particularly in America.

For a fuller account of the history of the evolution of the modern internal-combustion engine, and of the labours of many other distinguished engineers, as Ackroyd, Atkinson, Daimler, Diesel, Robson, &c., reference must be made to the large special treatises, as, e.g., those of Clerk* and of Bryan Donkin†. We proceed therefore to refer briefly to the gaseous mixtures ordinarily employed as fuel by gas-engines.

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# CHAPTER II

## Fuels

**Gaseous Fuels.**—The principal gaseous fuels used in Great Britain are: (1) town's gas, often termed " coal-gas "; (2) producer gas; (3) coke-oven gas; (4) blast-furnace gas. In America (5) natural gas is also used.

1. *Town's gas* obtained by the distillation of coal in closed retorts, and subsequently enriched by the addition of other gases, and particularly of " water-gas " (consisting mainly of carbon monoxide and hydrogen), is still the principal gaseous fuel of the very numerous small-powered stationary gas-engines so widely used for miscellaneous industrial purposes; it is an excellent fuel for gas-engines of any size, but its relatively high cost restricts its use in general to the smaller-powered units. At one time companies

* *The Gas, Petrol and Oil Engine*, 2 vols., D. Clerk (Longmans, 1913).

† *Gas, Oil and Air Engines*, Bryan Donkin (Griffin, 1911).

were compelled to supply gas of a standard illuminating power, but the introduction of the incandescent mantle, and the largely increased use of gas for heating and power purposes, rendered its heating value of the first importance Accordingly in 1920 an Act was passed empowering companies to sell gas at a price proportioned to its heat value, 100,000 British units of heat,* termed one "therm", being taken as the unit of supply. Thus, in 1929, the heating values of the gas supplied by the principal London companies were:

| | | |
|---|---|---|
| Of the South Metropolitan Gas Company .. .. | 560 | B.Th.U. per cubic foot. |
| Of the Gas, Light, & Coke Company .. .. | 500 | |
| Of the Commercial Gas Company .. .. | 500 | |

To ascertain the heat supplied in therms to a consumer one has therefore to multiply the number of cubic feet of gas used by the heating value per cubic foot, and divide by 100,000. Town's gas is a mixture, in somewhat variable proportions, of the combustible gases hydrogen, carbon monoxide, methane ($CH_4$), and other hydrocarbons, with small quantities of the incombustible gases carbon dioxide, oxygen, and nitrogen.

2. *Producer gas* is a mixture of gases formed by the combustion of almost any carboniferous fuel with a limited supply of air, and consists principally of carbon monoxide, hydrogen, and carbon dioxide, with a large quantity of atmospheric nitrogen.

In the most usual form of "suction producer" anthracite or coke contained in a cylindrical fire-brick lined "producer" is maintained incandescent by air drawn through it by the engine suction. Steam is also injected into the fuel, and the issuing mixture of hot and smoky gases after being cooled and cleaned is mixed with a suitable quantity of fresh air and supplied to the engine. For further details of this important subject the reader should refer to the article on "Gas Producers" in this volume. Producer gas is now used all over the world, a great variety of refuse material being utilized as fuel as, e.g., peat, sawdust, straw, coco-nut shells, cotton and sunflower seeds, tea prunings, rice husks, &c. The heat value per cubic foot varies with the fuel from which the gas is made; using anthracite or coke in a suction producer an average value is about 130 B.Th.U., the gas usually requiring for complete combustion rather under 1 c. ft. of air per cubic foot.

Its heat value is thus very much less than that of town's gas; nevertheless, as explained hereunder,† it forms a most satisfactory fuel for gas-engines; using gas coke a suction producer may be expected to give roundly about 180,000 c. ft. of gas per ton of coke burned.

3. *Coke-oven Gas.*—In the production of coke for use in blast-furnaces, foundries, and general metallurgical processes, large quantities of gas (about 10,000 c. ft. per ton of coke made) of high heat value are evolved. The gas varies considerably in composition, and consists of a mixture of the same gases as producer gas, but is characterized by its high hydrogen and

* The B.Th.U. is the heat required to raise 1 lb. of water through 1° F. † See p. 11.

methane content, the hydrogen varying from about 30 per cent to 60 per cent and the methane from about 20 per cent to 25 per cent of the whole volume. Such gas requires for complete combustion from 3 to 4 c. ft. of air per cubic foot, and has a heat value ranging from about 350 to 450 B.Th.U. per cubic foot. Though a valuable fuel for gas-engines, its variable constitution and high hydrogen content introduce some practical difficulties in its use, but these are met by providing special means of rapidly adjusting the admission of gas and air, while the tendency to pre-ignition due to the high proportion of hydrogen may be overcome by Clerk's method of admitting a small quantity of cooled exhaust gases with each fresh charge.*

4. *Blast-furnace Gas.*—In 1895 the late Mr. B. H. Thwaite showed that the hitherto waste gases produced in large quantities by blast-furnaces could be used in gas-engines, and his discovery greatly stimulated the development of the very large gas-engines now so widely used. The gas is of low heat value, averaging only about 100 B.Th.U. per cubic foot, and requiring for complete combustion about 0·75 c. ft. of air per cubic foot. Roundly, some 180,000 c. ft. of gas are ordinarily produced per ton of fuel burnt, the average composition in Great Britain being (by volume) carbon monoxide 25 per cent, hydrogen 2 per cent, nitrogen 66 per cent, carbon dioxide 6 per cent, traces of methane, &c., 1 per cent. The gas must, of course, be cooled and cleaned before use.

5. *Natural Gas.*—Very frequently associated with the oil deposits, vast reservoirs of natural gas exist over a wide area in the United States and elsewhere, as, e.g., the neighbourhood of the Caspian Sea; a small supply also exists in Great Britain at Heathfield, in Sussex, and it has been thought that considerable quantities are probably obtainable in this locality. In the United States the amount obtained annually is immense; e.g. in 1903 over 240,000 *millions* of cubic feet were used for heating, lighting, and power purposes. This amount is, however, decreasing. Natural gas often contains from 90 to 95 per cent methane ($CH_4$), with small quantities of ethane and olefines, and traces of water and oil; it is an ideal gas for use in internal-combustion engines. Considered as methane, its heat value is very high, viz. 1070 B.Th.U. per cubic foot, and it requires per cubic foot for complete combustion 9·5 c. ft. of air; actually its heat value is usually in the neighbourhood of 950 B.Th.U. per cubic foot.

In Table I some average figures relating to the above-mentioned gaseous fuels are collected together for general reference; figures for average petrol are included for comparison. It will be observed that, notwithstanding the large variation in the heat value per cubic foot of *gas*, the heat value per cubic foot of *mixture with air* varies between the much narrower limits of 57 to 95, due to the greatly differing volumes of air necessary to cause complete combustion in the several cases. It is mainly for this reason that the poorer gases prove so useful as engine fuels.

* *The Gas, Petrol, and Oil Engine*, Vol. II, p. 429 (Clerk and Burls).

TABLE I

AVERAGE HEAT VALUES OF THE USUAL GASEOUS MIXTURES

| Fuel. | Average Values. | | |
|---|---|---|---|
| | B.Th.U. per Cubic Foot at 60°F. and 760 mm. | Cubic Feet of Air per Cubic Foot of Gas for just Complete Combustion. | B.Th.U. per Cubic Foot of Mixture with Air. |
| Average petrol (as vapour) | 4650 | 49 | 95 |
| Natural gas .. .. | 950 | 9·5 | 90 |
| Town's gas .. .. | 500 | 5·5 | 77 |
| Coke-oven gas .. .. | 400 | 4·0 | 80 |
| Suction-producer gas from anthracite or coke | 130 | 1·0 | 65 |
| Blast-furnace gas .. .. | 100 | 0·75 | 57 |

Further, excepting in the very special, relatively small petrol-engines of motors and aircraft, experience has shown that in order to avoid trouble from overheating of pistons and cylinders it is necessary to limit the amount of heat supplied to the engine. A convenient method of estimation is to take the heat supplied per cubic foot of *working* stroke swept by the piston, and for gas-engine cylinders British practice in general allows only about 50 B.Th.U. per swept cubic foot with cylinders up to 20 in. in diameter, falling to from 35 to 40 B.Th.U. per swept cubic foot with 30-in. cylinders. This restriction limits the practically attainable (indicated) mean effective pressure in such engines during continuous running, to from about 65 to 55 lb. per square inch. The reduction in heat supply is obtained by increasing the admixture of air; thus, with town's gas, usual proportions are 9 to 10 volumes of air to 1 of gas; and similarly in other cases. See equations 8 and 9 and Table III of p. 31.

# CHAPTER III

## Methods of Ignition

In the evolution of the gas-engine, ignition was for long a source of trouble, and the solution of the problem of igniting the charge with certainty and regularity proved long and difficult.

In the earliest non-compression or low-compression engines, as, e.g., those of Street (1794) and Wright (1833), a crude "touch-hole" flame method was used. The piston near the commencement of its working stroke uncovered a small hole through which an external flame was sucked, thus exploding the fresh charge; the hole was sometimes so small as to cause no perceptible loss of pressure on explosion, or was fitted with a valve closed by the rise of pressure in the cylinder on ignition.

In engines compressing their charge before explosion this simple method was obviously inapplicable; but in 1838 W. Barnett introduced the celebrated "Barnett Igniting Cock", which successfully solved the problem of ignition by flame in compressing engines, and was extensively used from that date down to about 1892. An illustration of this igniting cock is given in fig. 4. It comprises a hollow open-ended single-ported plug A ground into a casing B containing two ports C and D, of which C communicates with the atmosphere and D with the engine cylinder. A gas-jet E burns continuously outside the cock, while F is a gas-pipe terminating in a small burner within the hollow plug A. The plug A is rotated by the engine in working, and in the position illustrated the external flame E ignites the jet within the plug. The rotation next causes the plug port to register with the port D, whereupon the charge within the cylinder is immediately exploded; the explosion extinguishes the jet in the plug, but when next it opens to the atmosphere it is relighted by the flame E in readiness for the next explosion; the action is thus positive and regular. The later improved flame-igniting devices of Hugon, Otto and Langen, and Dr. Otto were based upon the principle of the Barnett cock.

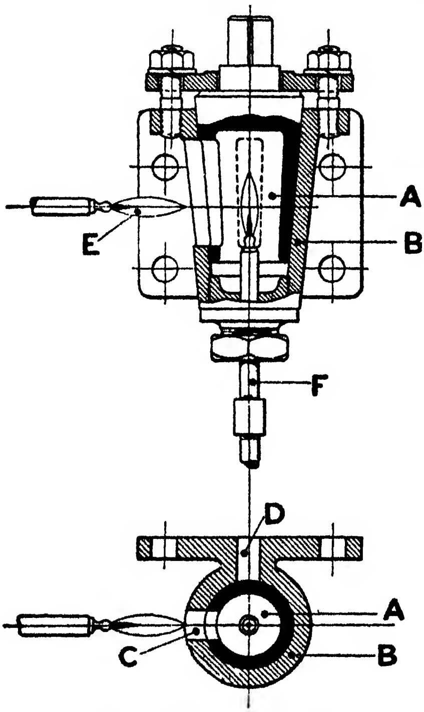

Fig. 4.—Barnett Igniting Cock

In 1799 Lebon proposed to use a machine worked by the engine to produce electric sparks for exploding the charge; in 1850 Stephard suggested the use of a magneto-electric machine for this purpose; in 1860 Lenoir employed a Bunsen battery, Ruhmkorff coil, distributor, and sparking-plug, substantially as was largely used in automobiles between 1898 and 1908; the art of constructing reliable coils and plugs was, however, but little developed in Lenoir's day, and great trouble was always experienced with the ignition of his engines. In Hugon's Lenoir-type engine an important improvement was the replacement of the electrical ignition by a flame device of modified Barnett type. Ignition by magneto-electric machines is now all but universal in gas-engines, but the unfavourable impression created by the early Lenoir arrangement probably retarded its development in Great Britain.

A method very largely used since about 1883, but now quite obsolete, is that known as "hot tube" ignition. Owing to its comparative simplicity its use was continued for some years after electric ignition was available.

As early as 1855 Dr. Drake, of Philadelphia, described an igniter consisting of a hollow cast-iron thimble projecting into a recess in the cylinder side, and maintained constantly at a red heat by an external blow-pipe flame. Early in the working stroke the piston uncovered the recess, and the red-hot thimble then exploded the charge. Hot-tube ignition of the modern type was, however, apparently first employed in the "Stockport gas-engine" in 1883. The principle of this mode of ignition is illustrated in fig. 5. The lower diagram shows the very simple method of ignition by "open tube" wherein the hot tube A is always in free communication with the combustion chamber C; firing occurs when the compressed fresh mixture reaches the red-hot part of the tube, the Bunsen-flame lamp B being adjusted in position by trial until the heated zone of the tube causes firing to occur at the correct instant.

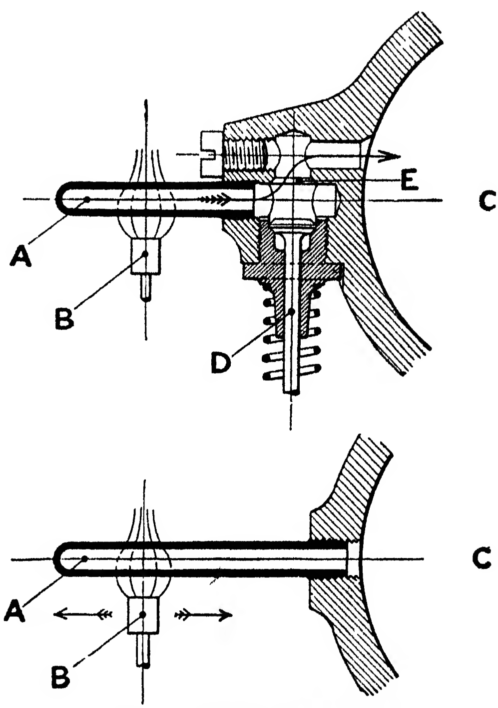


Fig. 5—Typical Hot-tube Igniters

Open-tube ignition proved quite satisfactory in engines of up to about 20 b.h.p., especially when using town's gas; for engines exceeding this power the possibly serious consequences of pre-ignition rendered it necessary to regulate positively and exactly the instant of ignition. This was effected by introducing between the hot tube and combustion chamber a small cam-operated valve termed a "timing" valve, as illustrated in the upper diagram of fig. 5. In this diagram D is the timing valve normally held on its lower seat by a spring, but raised by a cam during the compression stroke so as to seat at E and thus cut off communication between the hot tube and the combustion chamber. Near the end of the compression stroke the cam released the valve, which at once returned to its lower seat by the action of the spring, thus permitting the compressed fresh mixture to enter the hot tube, where it was immediately ignited.

Timing valves, exposed to a constant rush of flame, gave a good deal of trouble and required frequent renewal; with this system also the ignition was necessarily initiated at some little distance from the combustion chamber, which was often unsatisfactory, especially with large cylinders; the now almost universal electrical methods permit ignition to be effected simultaneously at two or more points actually *within* the combustion chamber, thus causing a quicker firing of the charge.

The hot tubes are of wrought iron, porcelain, or nickel alloy. Wrought-iron tubes are cheap, but oxidize rapidly, and have but a short working life. Porcelain tubes are readily heated and resist the chemical action of the flame, but are brittle and at once crack if wetted. Nickel alloy tubes resist the flame action satisfactorily, and have been very largely used. The best material of all is platinum, but its costliness prohibited its use except in the small petrol-engines of the earliest motor-boats, motor-cars, and motor-bicycles.

**Electrical Ignition.**—It has already been stated that electrical ignition was attempted quite early in the evolution of the internal-combustion engine, but its general adoption was for a time impracticable.

After Lenoir the battery-coil method went entirely out of use for many years, but improvements in manufacture caused its revival about 1898, and from that date until about 1908 it was very extensively employed, particularly in the small engines of motor vehicles. The earlier accumulators, however, proved a source of constant trouble, and an important advance was effected by deriving the igniting spark from a small electromagnetic machine driven by the engine itself. Such machines are conveniently classed as of " low tension " and " high tension " type respectively.

Low-tension magneto ignition is widely used, especially in cases where revolution speeds are not very high. The low-tension ignition plug facilitates easy starting, is a very robust and easily adjusted device, and tends to be self-cleansing—an important feature particularly in engines operated by producer gas. Accordingly in stationary gas-engines of all but the smallest sizes, low-tension magneto ignition was generally employed up to 1930, as, e.g., in the standard designs of Crossley, Premier, Ruston, and Tangye. In high-speed gas-engines the tappet action of the low-tension magneto method is too noisy, and the plugs tend to get out of adjustment, and in such cases high-tension ignition is usually employed. Thus, for example, the National Gas Engine Co. adopt high-tension magneto ignition in their smallest types, 3–10 b.h.p. running normally at 475–600 r.p.m., and also in their high-speed vertical engines running normally up to 500 r.p.m.; while in their range of engines from 10 to 80 b.h.p. running normally at 230–240 r.p.m. and in their large tandem vertical designs of 300–1500 b.h.p. running normally at 300–200 r.p.m., low-tension magneto ignition is used, the 300–1500 b.h.p. series being fitted with duplicate ignition, i.e. with two independent low-tension plugs to each cylinder.

A widely-used type of low-tension magneto by Bosch is diagrammatically illustrated in fig. 6; it comprises permanent horse-shoe magnets A between whose pole-pieces a *fixed* shuttle-wound armature B is fitted. Between B and the pole-pieces is a split soft-iron sleeve CC partly surrounding the armature; this sleeve is turned through an angle of about 45° by an engine-driven cam D, and on release is suddenly "flicked" back to its initial position by the strong springs E, E. The terminal K is connected by an insulated lead with the insulated electrode H of the make-and-break ignition plug in the combustion chamber of the cylinder, the other end of the magneto circuit being "earthed" to the engine frame; during the flick back of the sleeve C the light connecting-rod shown causes the "hammer" F to break contact suddenly with the central electrode or "anvil" H within the combustion chamber, as indicated, whereupon a strong spark leaps across the gap thus

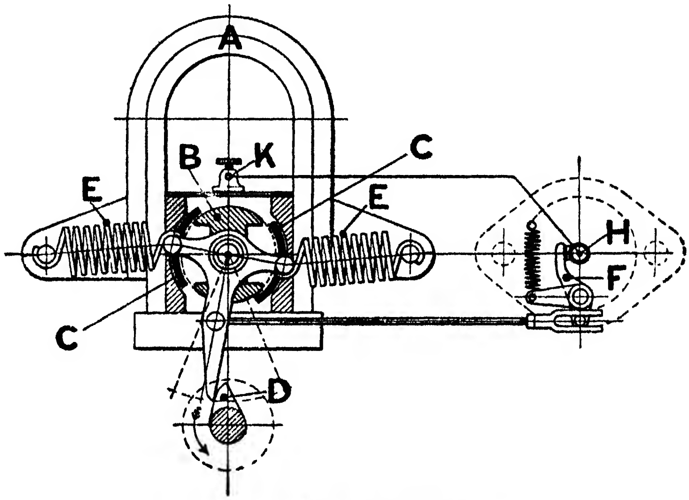


Fig. 6.—Bosch Low-tension Magneto

made, and ignites the charge. In the earliest designs the armature itself was rocked, but its inertia was found soon to cause trouble from disintegration and wear; with improvements suggested by experience, however, modern practice has largely reverted to the use of the rocking armature. The light soft-iron rocking sleeve was adopted in order to minimize inertia effects; it gave a good firing spark and enabled the current to be taken from a stationary armature, thus avoiding the necessity of collecting it by brush contact from a moving surface.

An important advantage of the low-tension magneto method of ignition is that a good "fat" firing spark is obtained no matter how slowly the engine may be running, a point of great value when poor gas is used. Starting is also often facilitated by "flicking" the magneto by hand after first inducing a partly compressed explosive charge in the cylinder by injection of gas or petrol, or by barring round. The make-and-break device is contained in a small casting usually attached to the combustion chamber by a pair of studs or nuts as indicated, and is thus readily removed for adjustment or repair; in starting from "all cold" it is frequently found necessary to remove and warm the make-and-break casting in order to dry off any condensed moisture. The sparking current produced on break of contact is of high voltage, and the effective and durable insulation of the anvil H has occasioned considerable difficulty. Asbestos, steatite (soap-stone), mica, porcelain, enamel, &c., are all used as insulators; the material employed must be not only a good insulator, but must also be capable of withstanding a high temperature and of resisting the disintegrating tendency of the constant impacts of the hammer F on the anvil H. To meet the somewhat rapid burning away at the gap, replaceable contacts of nickel, nickel-steel, and other alloys are provided.

In large cylinders two make-and-break ignition plugs are often fitted, each independently served by its own magneto; this is useful in case of the failure of one system, and also reduces the time of explosion of the charge, which is an important point when using weak fuels, as, e.g., blast-furnace gas, sewer gas, &c.

The high-tension magneto, universally employed for the ignition of the innumerable small high-speed engines of motor vehicles of all kinds, is so well known that a very brief reference to it only is necessary. In the high-tension magneto the shuttle armature rotates continuously between the pole-pieces of the permanent field-magnets and carries two windings, viz. a stout-wire primary and fine-wire secondary; the primary current is automatically broken twice per armature revolution by a contact breaker forming part of the magneto, and the induced high-voltage secondary current is " collected " by a carbon brush and transmitted to a highly insulated " distributor ", by which it is served to the cylinder sparking plugs in proper order. In its modern form the high-tension magneto is extremely reliable, and will often run for years without any more attention than an occasional oiling; it is usually made water- and dust-tight. Modern high-tension sparking plugs also are exceedingly reliable and involve no moving parts as in the low-tension system, while the regulation of the instant of ignition or " timing of the spark " is very simply and easily arranged. In 1922 the British Lighting and Ignition Company introduced a high-tension magneto having a *stationary* armature and rotating magnet, resulting in a considerable saving of weight, and claimed to possess several mechanical advantages over the usual design.* This device has been generally adopted by other makers.

* *The Automobile Engineer*, November, 1922.

# CHAPTER IV

## Governing

Engines in general are required to maintain a considerable degree of uniformity in their speed under normal working conditions, and in this respect the single-cylindered single-acting four-stroke engine is at a serious disadvantage, as a working impulse occurs only, at best, once in every four strokes; the momentum of massive fly-wheels thus has to maintain rotation during at least three-fourths of the running time. This inherent drawback is, however, reduced to a practically negligible amount: (*a*) by using multi-cylindered single-acting engines; (*b*) by using tandem or twin-tandem double-acting engines; and (*c*) by using two-stroke single- or double-acting engines; and in nearly all cases with the addition of a massive fly-wheel.

**Speed Fluctuation.**—The coefficient of fluctuation of speed of an engine is defined as the ratio of the difference between the maximum and minimum angular velocity of the crank-shaft per cycle to its mean angular velocity. The permissible value of this coefficient depends upon the nature of the work performed by the engine; usual values in a number of typical cases are given hereunder:

| Nature of Service. | Approximate Permissible Coefficient of Speed Fluctuation. |
|---|---|
| Driving pumps .. .. .. .. .. | 1/20 |
| Driving machine tools .. .. .. .. | 1/35 |
| Driving textile machinery .. .. .. .. | 1/50 |
| Driving C.C. dynamos .. .. .. .. | 1/200 |
| Driving spinning machinery .. .. .. | 1/100 |
| Driving direct-coupled alternators in parallel .. | 1/250 |

So long as the external resistances overcome by the engine remain unchanged, cyclic speed fluctuation can be reduced to any desired extent by providing sufficient fly-wheel inertia; for a detailed consideration of this question reference must be made to the larger special treatises.* When the external resistance is changeable, it is usually necessary to provide that the engine speed shall not be permitted to vary from the normal by more than a small amount, and some system of governing thus becomes essential.

The governor itself is almost always of the centrifugal type of which the well-known "conical pendulum" of Watt is the parent; such governors have been greatly developed in recent years by British and German engineers, among whom may be mentioned Hartnell, Pröll, Beyer, Hartung, and Rost.† In all these centrifugal action, due to varying engine speed, provides motion which is applied to vary the quantity or quality of the explosive charge

* Cf. R. E. Mathot, *Construction and Working of I. C. Engines* (Constable), or Clerk and Burls, *The Gas, Petrol, and Oil Engine*, Vol. II (Longmans).

† For a full account see Clerk and Burls, Vol. II, Chapter IV.

admitted to the engine cylinder in one or more of the following three principal ways:

1. By completely cutting off the gas supply in one or more cycles—" hit-or-miss " governing.

2. By partially cutting off the gas supply, thus varying the *quality* of the mixture, the mass of the charge remaining unchanged.

3. By varying the opening of the mixture valve, thus varying the *quantity* of charge admitted to the cylinder, keeping the ratio of gas to air unchanged

**" Hit-or-miss " Governing.**—In this method when engine speed increases one or more charges of gas are completely cut out, so that no working stroke occurs until the speed falls again to about its normal value. It was for long almost universally employed in stationary gas-engines, and is still widely used in engines of up to about 100 b.h.p. in cases where extreme uniformity of speed is unnecessary. It possesses the advantage of giving economical fuel consumption both at light and at full loads, and is mechanically simple and reliable, but a very heavy fly-wheel is necessary.

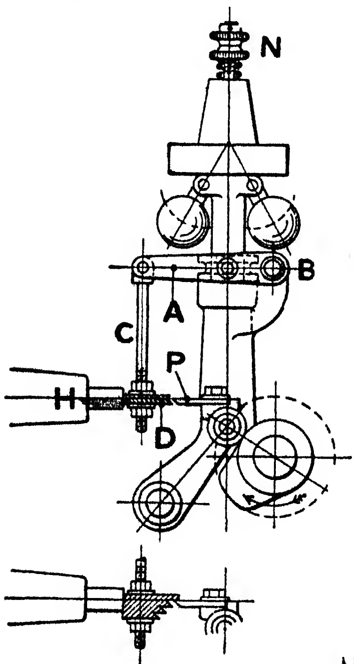

Fig. 7.—Hit-or-miss Arrangement of Governing

A simple and effective hit-or-miss arrangement is illustrated in fig. 7. The " pecker " P normally opens the gas valve H through the " pecker block " D, suspended by a rod C from a lever A which is pivoted at B; obviously a very small rise of the governor balls suffices to lift D out of the way of the pecker P, and when this occurs the gas valve remains closed. The governor is easily and quickly adjusted by varying the spring compression by aid of the milled nuts N on the top of the governor spindle.

**Quality Governing.**—With this method of governing, the mixture becomes progressively weaker as the load on the engine diminishes, though the compression pressure is constant since the mass of the charge remains unchanged. The reduction in the gas admitted is usually affected either: (1) by shortening the duration of opening of the gas valve, the air admission remaining constant; or (2) by throttling the gas supply throughout the suction stroke, with constant air admission. Often the gas valve is not opened until part of the suction stroke has been performed, thus providing a rich and readily ignitible mixture near the firing plugs in the combustion chamber.

At light loads the weak mixtures supplied to the engine were at first difficult to ignite and burned slowly; heat losses were thus often much increased, and the charge was also often not fully burned before release occurred. Combustion was occasionally so slow as to persist during the exhaust stroke and explode the succeeding fresh charge, thus causing what is termed a "back fire". Both these drawbacks are practically overcome by delaying the opening of the gas valve until part of the suction stroke has been performed.

By simply stepping the pecker block as shown in the lower portion of fig. 7, a crude form of quality governing is readily obtained. With a stepped pecker block the pecker, when raised by the action of the governor, moves the gas valve not only later but also through a smaller lift, and finally, when the speed-change is considerable, misses it altogether.

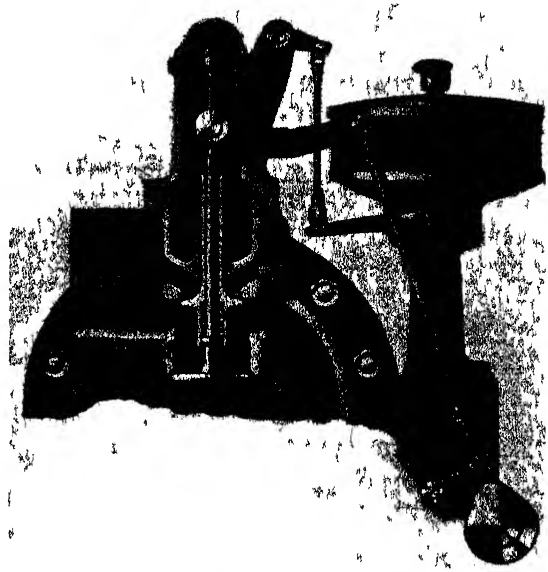

Fig 8—Shifting Fulcrum Device for Quantity Governing

The Premier gas-engines are governed on the "quality" method; a centrifugal spring-loaded enclosed governor driven from the crank-shaft by skew gearing controls a throttle valve in the gas supply. A difficulty gas-engine designers have to overcome is that of the considerable variation in the quantity and composition of the gas supplied to the engine, particularly when only one suction producer is employed, and provision for manual adjustment of the mixture is therefore necessary. In the Premier engines a hand-adjusted master-throttle determines the total air supply to the engine, while separate throttles to the cylinders enable the mixture strength of each to be independently regulated.

**Quantity Governing.**—In this method, the mass or "quantity" of the working charge is reduced as the engine load diminishes, but the ratio of air to gas is kept unaltered; the compression pressure is accordingly also then reduced. Sir Dugald Clerk observes:* "The reduction in the compression pressure causes some diminution of thermal efficiency at such times; but as the compression and expansion curves rise and fall together the variation of crank-pin effort is not unfavourably affected. With poor gas of small hydrogen content high compression pressures (175 lb. per square inch and above) may safely be used, thus giving increased economy and more rapid and complete combustion of the charge, and in such cases quantity governing

* Clerk and Burls, Vol. II, p. 364.

is most satisfactory, since the compression at light loads remains still high enough to ensure ready ignition of the mixture . . . the frictional resistances of the engine are also reduced by the diminished compression. On the whole the balance of practical advantage in general favours governing by this method rather than by that of quality." Quantity governing is largely employed, as, e.g., in the Crossley, Hornsby-Stockport, and Tangye gas-engines. The exceedingly simple and effective "shifting fulcrum" device used in the quantity governing of the Crossley gas-engines in shown in fig. 8, wherein the rocking lever fulcrum is in the position for light load; at full load the governor moves the fulcrum towards the right, causing the two arms

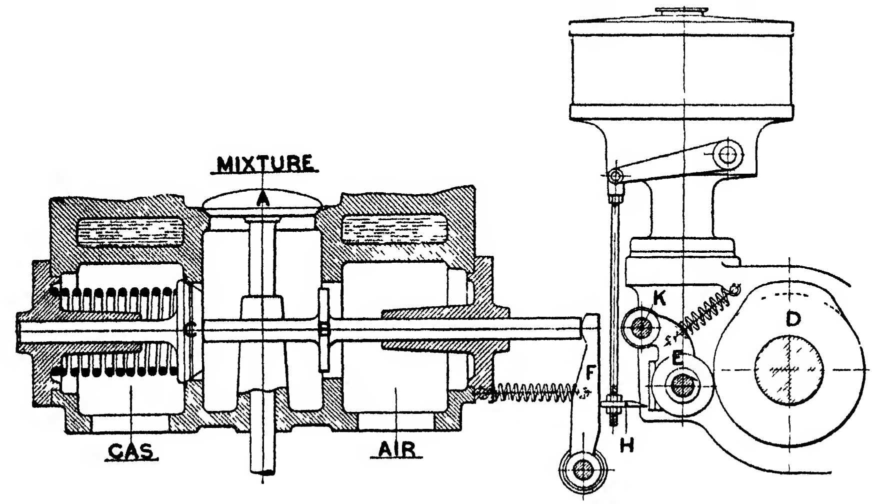


Fig. 9.—"National" Gas-engine Governing Arrangement

of the rocker to become nearly equal, thereby increasing the lift of the mixture inlet valve. The gear being in full view of the attendant, inspection of the position of the fulcrum indicates at once the amount of load on the engine. With this governor the permanent speed variation between no load and full load is only about ± 2 per cent from the mean. Messrs. Tangye also use a shifting fulcrum device, but of somewhat different type.

The governing arrangement used in the "National" gas-engines ingeniously combines all three methods, viz. of hit-or-miss, quality, and quantity; it is illustrated diagrammatically in fig. 9. The gaseous mixture is admitted to the combustion chamber of the cylinder through the charge inlet valve A; this valve is operated by its cam in an unvarying manner at all loads, opening about 15° before inner dead centre, and closing about 55° after outer dead centre of the crank-pin. The air valve B is a circular disc of somewhat smaller diameter than the hole in the casing in which it works, so that even when the gas-valve C remains seated it is still possible for air alone to enter the cylinder through the charging valve A.

The cam D on the side shaft actuates the air-valve spindle—and through this the gas valve also as indicated—through the agency of the two levers E and F and the governor-controlled plate H; the lever E is pivoted at K, and when the cam D is out of action the serrated ridge of the lever E is clear of the plate H, and during such times the governor is free to raise or lower this plate agreeably with its speed of rotation. When the speed increases, H is raised and the opening of the gas and air valves is then reduced, and vice versa; in this way the lift of these valves may be varied from $\frac{1}{8}$ in. at light loads to $\frac{3}{4}$ in. at full load.

The cam D is so shaped that the cut-off point of the gas remains nearly constant; the gas is cut off early enough to allow the combustible mixture in the space below the charging valve to be sucked into the combustion chamber before this valve closes. This is necessary in order that this space may be filled with air which then alone enters the cylinder at the commencement of the next suction stroke, thus cooling the residual exhaust gases before fresh inflammable mixture enters the combustion chamber.

At full load the plate H occupies its lowest position, and a full charge of gas and air is admitted. For somewhat smaller loads the gas supply is reduced while the air supply suffers but little diminution; the mass of the charge and accordingly the compression pressure are therefore but little affected, and the governing is of the quality type. At lighter loads the air valve B is nearer to its housing, and the ratio of gas to air admitted then remains more nearly constant while the mass admitted is reduced; the governing is then of the quantity type. Finally, at very light loads the plate H may be raised so high that it is occasionally missed altogether by the serrated ridge of the lever E, and the governing is then of the hit-or-miss type.

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# CHAPTER V

## Starting

For gas-engines of less than about 30 b.h.p. no special starting apparatus is usually needed. When preparing to start, attention should be given: (1) To the oiling system. All lubricators and oil reservoirs should be replenished, oil wicks adjusted, and sight-feed lubricators started. (2) To the cooling water system. The cock on the water-supply pipe to engine should be opened, drain cocks closed, &c. (3) The inlet, exhaust, and throttle valves should be moved by hand to ascertain that they are not "gummed up". (4) The driving belt should be on the loose pulley. (5) If a compression relief be fitted, the exhaust valve roller should be placed opposite the narrow or "relief" cam. (6) The ignition should be fully retarded. (7) The gas should be turned on only when ready to start, ascertaining first that the gas-supply pipe is free from air and charged with pure gas up to the engine by turning on the vent pipe and lighting the test burner until it burns well and steadily. (8) The fly-wheel should next be turned by hand as quickly as possible until the engine starts, pulling downwards on the rim and away from the engine; the feet should never be placed on the fly-wheel spokes. (9) After a few explosions have occurred the ignition may be slightly advanced and the relief cam put out of action. (10) When full speed is attained and the engine is warmed up the ignition should be advanced to the normal working position, and the air and gas regulators adjusted. Finally the load may be imposed on the engine.

With engines of over about 30 b.h.p. some form of special starting apparatus is, in general, necessary. Fly-wheels are usually furnished with a series of holes round the rim for the insertion of a crowbar, or an internally-toothed ring operated by a small pinion and hand-wheel or small motor, by which the engine may be " barred " round so as to place the crank-pin in a favourable position for starting, i.e. 15° to 20° beyond the inner dead centre on the firing stroke.

Many medium large gas-engines, from 30 or 40 b.h.p. up to about 200 b.h.p., are started by means of a small pump fitted to the engine, by which an initial charge of gas and air is pumped by hand into the combustion chamber and ignited by " flicking over " the magneto by hand. The explosion thus obtained imparts sufficient motion to the engine to enable it to take up its normal working cycle. A small quantity of petrol is often pumped in with the air, as an alternative to gas, and this gives a more powerful starting impulse. Such starters are very effective, and are used, e.g., in the National, Anderson-Grice, and Brotherhood gas-engines.

With large gas-engines the single impulse furnished by the hand-pump method is insufficient to effect a start, and accordingly compressed air, stored in cylindrical steel reservoirs, is employed for starting purposes. With new engines the reservoir is sent out charged, but thereafter its pressure may be maintained: (1) from the engine cylinder and piston through a special delivery valve; or (2) from a small air-compressor belt-driven by the engine; or (3) by a small auxiliary engine and air-compressing pump. Starting by compressed air is very simple; a cam opening a small compressed-air supply valve in the combustion chamber, during the earlier part of the firing stroke, is put in action. The compression relief cam of the engine is also put in action and the ignition retarded; the engine is then barred round until the crank-pin is in the starting position (i.e. just well over the inner dead centre on the firing stroke); and the gas is turned on. On opening the air reservoir cock the engine at once commences to turn, and usually takes up its normal working cycle after three or four air impulses. The air-valve cam is then moved out of action and the air cock shut; the reservoir is made of capacity sufficient to provide a large number of engine impulses in case of difficulty in starting occurring.

Having started, the relief cam is put out of action, the ignition advanced, and the mixture adjusted as usual, the load being finally put on the engine when everything is well warmed up.

The air is stored in the reservoirs at a pressure usually of from 100 to

250 lb. per square inch; and with multi-cylindered engines air starting gear is often fitted only to some of the cylinders, as in the four-cylindered Premier gas-engines, where two only of the cylinders are so fitted. In the case of a 400-b.h.p. four-cylindered engine, sufficient air for starting purposes, compressed to 250 lb. per square inch, is stored in three steel cylinders or " air bottles " each of 16 in. diameter and 8 ft. long.

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# CHAPTER VI

## Cooling

The adequate cooling of gas-engines has proved a problem of very great difficulty, and it is still necessary to limit strictly the supply of heat to large engines in order to avoid trouble from overheating (p. 11).

Of the whole heat supplied to a gas-engine, roundly from 25 to 35 per cent usually appears in the cooling water. Taking the heat supply in normal everyday working as 10,000 B.Th.U. per brake horse-power hour, it is clear that the cooling water must carry away from 2500 to 3500 B.Th.U. per brake horse-power hour. Assuming the rise of temperature of the water to be 60° F. in passing through the engine, it will accordingly be necessary to pass through the jackets, &c., from 4 to 6 gall. per brake horse-power hour in normal full load working; when the cooling water is run to waste this is not difficult to arrange, the outlet temperature being kept at about 120° F., as, e.g., in small engines cooled from town mains. With larger engines the more usual practice is to provide an overhead storage tank from which water is supplied to the cylinder jackets under a gravity head of not less than about 20 ft. (approximately 10 lb. per square inch pressure); for engines of up to, say, 100 b.h.p. and in temperate climates the " thermo-siphon " system of cooling is often employed, but in such cases the outlet temperature of the water is higher and may be 130° F. to 140° F., though it should not exceed the latter value, while the incoming water may be fully 100° F. in temperature. The rise of temperature being less than as previously assumed, more water must be circulated through the jackets, and hence it is common to find provision made for passing fully 10 gall. of water through per brake horse-power hour, with a water storage capacity of 25 to 35 gall. per (maximum) brake horse-power of engine. With large engines the water is usually pump-circulated, a cooling tower being often installed from the top of which the heated water from the engine falls through the air in fine streams. Thus a large 600-b.h.p. " Simplex " engine working on blast-furnace gas at the Ormesby Iron Works, Middlesborough, is cooled by water from a tank built over the engine-house, and delivered under a head of about 60 ft. On leaving the engine it passes into a small tank containing a float so arranged that should the circulation fail the float sinks and stops the engine through the governing gear; from this small tank the water passes to a reservoir, whence it is pumped up to the top of a "Klein" open-type water cooler fixed above the elevated tank. The makers recommendation was that about 12 gall. of water should be passed through the jackets per brake horse-power hour, i.e. about 7000 gall. per hour total.

Soft water is preferable, but the available water is commonly of some degree of hardness which in time causes the formation of deposits in the jacket and cylinder liner; this can be largely prevented by using a specially large cooling tank, and by treating the water, when chalk-hard, with common soda, 1 lb. being added per 250 gall. of water in the tank, once a month. When tanks and coolers are used, only the loss by evaporation has to be made up, and this may be estimated at from 0·2 to 0·4 gall. per brake horse-power hour. Some further references to cooling details occur in the description of actual engines given later.

## CHAPTER VII

# Lubrication

With every engine a part of the work done by the gases in the cylinder is expended in overcoming the internal resistances of the engine itself; of these resistances piston friction constitutes by far the greatest item, and accounts, under normally good running conditions, for fully 50 per cent of the difference between the indicated horse-power and the brake horse-power. Piston friction is thus large, and it is also variable, being very dependent upon the condition of fit of the piston and rings in the cylinder, on the nature and extent of the lubrication, temperature of the jacket water, and age of the engine. In all gas-engines careful provision is accordingly always made for the adequate lubrication of the pistons. In small horizontal engines an adjustable glass sight-feed drip

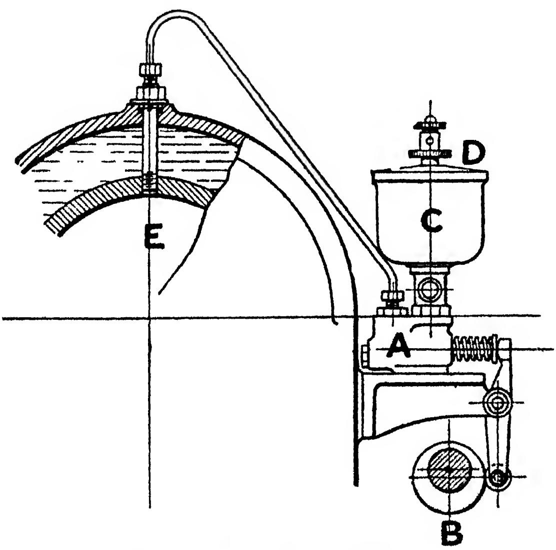

Fig. 10.—Tangye Cylinder Lubricator

lubricator is commonly fixed on the top of the cylinder, frequently near the open end, and delivers oil on to the piston, which is furnished with grooves down which it runs, thus reaching all parts of the surface. With large gas-engines the oil is force-fed to the cylinder lubricator by a small oil-pump usually driven from the half-speed shaft. The arrangement employed in the Tangye engines, for example, is shown in fig. 10: A is a small oil-pump cam-operated from the half-speed shaft or "side-shaft" B; the pump takes oil from the reservoir C through a suction duct hand-regulated by the milled nut D, and delivers it as shown into the top of the engine cylinder E.

Gudgeon bearings of horizontal engines are commonly lubricated by an adjustable visible drip-feed lubricator delivering oil by aid of a "wiper" into a short open trough fitted on the top of the small end of the connecting-rod; often also the gudgeon bearing is oiled by the piston lubricator through a hole in the upper part of the piston, as shown in fig. 12.

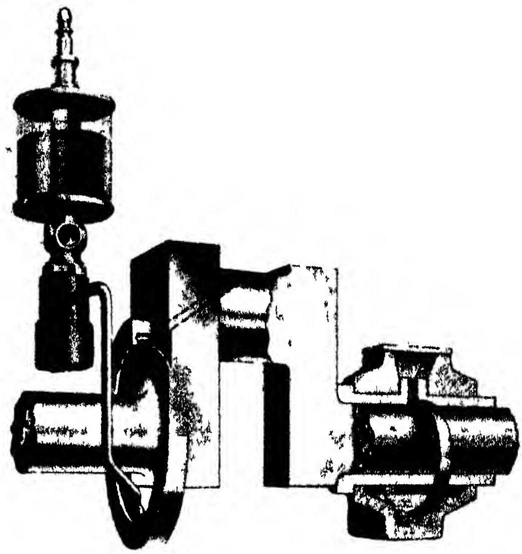

Fig 11 —Crank-pin Lubricator

Crank-pins are usually and effectively oiled by a sight-feed lubricator in conjunction with a centrifugal oiling ring attached to the crank web as clearly indicated in fig. 11, illustrating the method used in, e.g., the Crossley, Tangye, Campbell, and Anderson-Grice engines. The oil delivered into the ring is discharged by the centrifugal action, through the duct shown, to the outer surface of the crank-pin.

Main crank-shaft bearings are most commonly oiled by "ring" lubrication, also illustrated in fig. 11. The ring runs loosely on the shaft and is driven round by it, thus continuously raising oil from the reservoir in the base of the bearing and delivering it on the upper surface of the shaft. A drain plug is fitted in the oil reservoir enabling it to be emptied and cleaned out when necessary. Side-shaft bearings are also often ring-lubricated, though ordinary cotton wick lubricators are frequently fitted, while, in small engines, simple oil holes suffice; the gear wheels driving the side-shaft are usually run in an oil-bath. All rollers, pins, and valve-spindles should be lightly oiled; if the gas used produces a tarry deposit, exhaust valve spindles should be lubricated with a mixture of oil and ordinary paraffin to prevent

" sticking up ". Engine bedplates and crank-pits are so arranged as to form troughs in which waste oil may collect, thus preventing the engine foundations from becoming oil-soaked.

Many of the lubricating details referred to above may be studied *in situ* in later illustrations herein.

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# CHAPTER VIII

## Modern Gas-engines

For long almost the only type of gas-engine built was the single-cylindered single-acting four-stroke horizontal design, and this is the type still most largely used from the smallest powers up to as much as 250 b.h.p. in factories, mills, and for general industrial purposes where a cyclic speed fluctuation as high as 4 to 5 per cent is permissible. In 1929, for example, Messrs. Anderson-Grice, Crossley, National Company, Premier Company, Ruston-Hornsby, Tangye, &c., in Great Britain were all building ranges of single-cylindered horizontal engines of from 1 to 250 b.h.p., and of double-cylindered, and coupled, horizontals ranging from 50 to 500 b.h.p., together with a few four-cylindered horizontal designs ranging from about 250 to 600 b.h.p. per unit; all these were of the open crank-chamber single-acting four-stroke type with uncooled pistons, and running at speeds ranging from 450 to 600 r.p.m. in the smallest sizes to 150 to 160 r.p.m. in the largest. British gas engineers have not, so far, favoured the extra complication involved in the double-acting cylinder with water-cooled piston, but have consistently adhered to the single-acting uncooled piston type; the largest uncooled British pistons are found in certain Crossley engines of 26-in. cylinder bore, and in some of the large vertical engines of the National Company which have 24-in. cylinders.

Large horizontal engines of the Continental type have been built in Great Britain to a limited extent by Messrs. Beardmore, Mather & Platt, Galloways, Richardsons Westgarth, Vickers, and the Lilleshall Company, this latter firm having produced some fine examples of the Nuremberg or "M.A.N." type (p. 31).

Vertical gas-engines were developed rapidly from about 1900 onwards, prompted largely by the necessity of saving floor space and weight in multi-cylindered single-acting engines; one of the first attempts to produce engines of this type was made by Messrs. Burt, of Glasgow, in 1894. High-powered quick-speed multi-cylindered enclosed verticals had become common in 1929, prominent British builders being:—

1. Messrs. Browett-Lindley, who manufactured a range having two, three four, or six cylinders, with an output from 60 to 750 b.h.p. per unit, and running at from 450 to 200 r.p.m. At the Waterloo Colliery, near Leeds, two of their four-crank, four-cylinder enclosed vertical engines are installed

each driving a 250-kw. generator. Each of these engines comprises four single-acting four-stroke cylinders of 19-in. bore and 20-in. stroke, capable of a maximum output of 400 b.h.p. at 250 r.p.m.

2. The late Campbell Gas Engine Company in 1922 built successful two-cylindered and four-cylindered engines.

3. The National Gas Engine Company build a large range of gas-engines from small single-cylindered horizontal designs of 3–4 b.h.p. running normally up to 600 r.p.m., to very large 12-cylindered 6-crank tandem verticals of 1500 b.h.p. at 200 r.p.m. All their engines are of the single-acting four-stroke type. There is also a series of high-speed verticals with from 1 to 6 cylinders, all running normally at 450 r.p.m. with outputs ranging from 25 to 150 b.h.p. when using producer gas, or 27 to 162 b.h.p. on town's gas; this series is equipped with duplicate high-tension magneto ignition, i.e. two independent sparking plugs to each cylinder.

The National Company also build heavy-oil "compression ignition" or "semi-Diesel" engines for all purposes, and a series of special interest is that of their convertible engines, adapted to run on either heavy oils or gas These are of the multi-cylindered inverted vertical type, with from 1 to 6 cylinders, started by compressed air, and run normally at 450 to 275 r.p.m., developing, on producer gas, from 17 to about 500 b.h.p.

When using heavy oil, airless injection of the fuel is adopted; when gas is to be used, the fuel atomiser is replaced by a high-tension sparking plug, and the compression plates fitted below the feet of the connecting-rods are changed. The engines are fitted with inlet valves for gas and air which do not require removal when oil is used as fuel. The gas and air are kept apart, and mingle only in the valve cage just prior to entering the cylinder.

TABLE II

"NATIONAL" VERTICAL GAS-ENGINES, 1930

(See also p. 46)

| B.H.P. | | Number of Cranks. | Stroke in Inches. | Speed, R.P.M. | | Approximate Weights, Tons. | |
|---|---|---|---|---|---|---|---|
| Normal Full. | Overload. | | | Normal Full. | Recommended, Ordinary. | Fly-wheel. | Total Net. |
| 300 | 330 | 2 | 18 | 300 | 275 | 5 | 31·5 |
| 450 | 495 | 3 | 18 | 300 | 275 | 5 | 43 |
| 600 | 660 | 4 | 18 | 300 | 275 | 5 | 53 |
| 750 | 825 | 3 | 24 | 200 | 185 | 13·5 | 78 |
| 1000 | 1100 | 4 | 24 | 200 | 185 | 13 5 | 98 |
| 1500 | 1650 | 6 | 24 | 200 | 185 | 13·5 | 149 |

Two cylinders, in tandem, act upon each crank. The normal full brake horse-power is that which the engines are capable of maintaining continuously; the "overload" is 10 per cent greater and should only be imposed during short emergency periods. The powers as stated are, moreover, for engines working at sea-level and in a temperate climate; when the plant is installed above sea-level a deduction of about 3½ per cent should be made from the power rating for every 1000 ft. of height. In Table II the initial temperature of the gaseous mixture before entering the engine is also assumed as 60° F.; in hot countries some power loss is incurred by reason of the diminished density of the mixture when supplied to the engine at a higher temperature; the loss from this cause may be estimated as 1 per cent of the sea-level rating for every increase of 5° F. above 60° F.

The normal full revolution speeds given in Table II are also maxima values for continuous running. The fly-wheels referred to in the table are as used with engines driving direct-current electrical generators, air-compressors, &c.; when alternating-current generators are being driven the necessary steadying rotational inertia is usually furnished by the generator rotor itself, and in such cases only a coupling at the crank-shaft end, or a light wheel for barring round the engine, is necessary.

# CHAPTER IX

## Typical Engines Described

**Horizontal Engines.**—A longitudinal section through a 100-b.h.p. horizontal National engine is given in fig. 12. The replaceable cylinder liner

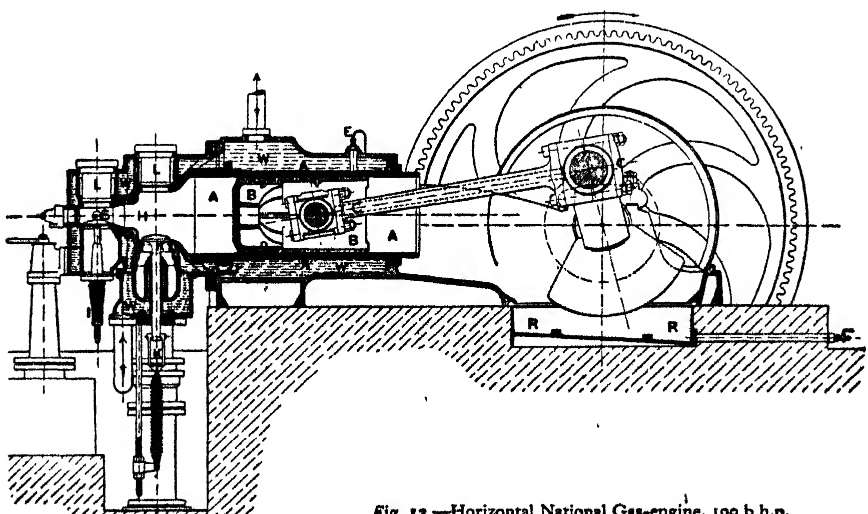



*Fig.* 12.—Horizontal National Gas-engine, 100 b.h.p.

or "working barrel" AA is of specially hard close-grained cast iron, and is connected to the breech casting by a bolted joint, and fitted at its outer end with a rubber water-tight expansion joint enabling it to expand relatively to the jacket casting when heated during working, thus avoiding setting up internal stresses within the cylinder due to differences of temperature. The liner is made separate partly because it may then be of the most suitable cast iron to resist abrasion, and partly because after long service it becomes "ovalled" by wear, and may then be withdrawn and replaced without sacrifice of the whole cylinder. Within the liner slides a long piston BB of hard close-grained cast iron, driving the crank-shaft through a connecting-rod CC. The piston fits the cylinder closely, but gas tightness is maintained, even after some wear has taken place, by the six cast-iron spring rings DD; its flat-topped end or "crown" is strongly supported by ribs beneath as shown; their function is, in addition, to facilitate the conduction of heat away from the crown, which is normally one of the hottest parts of a gas-engine, and one of the most difficult to cool.

The large gudgeon-pin, of steel, is lubricated from the pipe E through the hole in the piston as indicated.

Within the combustion chamber H are situated the exhaust valve K and the mixture inlet valve I; on removing the plugs L, L these valves may be easily withdrawn for examination, adjustment, or renewal. The valves are of steel of the usual "poppet" or "mushroom" type, normally held to their conical seats by adjustable strong helical springs as shown, and raised as necessary during working by rocking levers operated by cams upon the side-shaft.

The cylinder liner, combustion chamber, and valve casings are cooled by the capacious water jacket WWW, the water entering and leaving as indicated by the arrows. It will be noted that the combustion chamber is a casting separate from the jacket, and that the inlet valve casing is again a separate casting; by disconnecting these all jacket water spaces are readily accessible for cleansing and removal of deposit. This arrangement necessitates double joints; of these the inner are made with asbestos, and are screwed up hard so as to remain tight under the explosion pressure; the outer joints having to maintain only water-tightness are made with rubber. The exhaust valve seat and stem are carefully water-jacketed, and it will be seen that the valve-stem guides are separate castings readily replaceable when worn. The cast-iron crank-pit trough RR prevents waste oil from soaking the engine foundations. The ignition plug is indicated at S.

The compression ratio in this engine was 5·5, corresponding to a compression pressure of about 140 lb. per square inch (absolute).

With the small engines of motor vehicles and aircraft the high revolution speed renders it impracticable to take indicator diagrams, but with the lower speeds of gas-engines this difficulty does not exist; in "indicating" a gas-engine, however, experience has shown that consecutive diagrams vary much more than those of a steam-engine under steady load, and accordingly the mean should be taken of a considerable number of diagrams traced upon the same card.

The brake horse-power of gas-engines is also, in general, easily obtainable, particularly when driving electric generators.

Using the notation as in the article on "Oil-Engines" in this volume (p. 173), the indicated horse-power of a single-cylindered single-acting four-stroke engine is expressed by

$$\text{I.H.P.} = 992d^2snp \times 10^{-9} \ldots\ldots\ldots\ldots\ldots\ldots (1)$$

The brake is less than the indicated power by the horse-power necessary to overcome the internal frictional resistances of the engine itself; thus:

$$\text{engine friction horse-power} = \text{I.H.P.} - \text{B.H.P.} \ldots\ldots\ldots (2)$$

The mechanical efficiency, usually denoted by $\eta$, is the ratio of the brake horse-power to the indicated horse-power; thus:

$$\text{mechanical efficiency} = \eta = \frac{\text{B.H.P}}{\text{I.H.P.}} \ldots\ldots\ldots\ldots (3)$$

So that equation (1) may be written:

$$\text{B.H.P.} = 992d^2sn\eta p \times 10^{-9}, \ldots\ldots\ldots\ldots\ldots\ldots (4)$$

and the product $\eta p$ is termed the "brake mean effective pressure".

The piston speed in feet per minute ($\sigma$) is expressed by:

$$\sigma = \frac{ns}{6}, \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (5)$$

and is an important quantity in design. Lastly, the mean effective pressure can easily be shown to be given by

$$p = 5{\cdot}4h\epsilon, \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (6)$$

where $h$ is the number of British thermal units of heat supplied per cubic foot of swept working stroke, and $\epsilon$ is the absolute indicated thermal efficiency of the engine. If H.B.Th.U. of heat be given to the engine per indicated horse-power hour then

$$\epsilon = \frac{2545}{\text{H}} \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (7)$$

Combining equations (6) and (7) gives the useful relation:

$$h = \frac{1}{13700}\text{H}p \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (8)$$

If H′ be the number of British thermal units supplied to the engine per brake horse-power hour, then $\text{H}' = \frac{\text{H}}{\eta}$, and equation (8) is written in this case:

$$h = \frac{1}{13700}\text{H}'\eta p \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (9)$$

The above relations are in constant use in power considerations relating to gas-engines.

**Trial Results.**—The performance of the 100-h.p. National engine has been found, from trials in actual practice, to be as shown in Table III below.

TABLE III

SOME TEST RESULTS FROM THE 100-H.P. NATIONAL ENGINE

Horizontal, Single-cylindered, 16-in. Bore × 22-in. Stroke, 210 r.p.m

| Fuel Used. | Max. Horse-power. | | | Mean Eff. Press., lb./sq. in. | | Efficiencies. | | Heat Supply in B.Th.U. | | | Piston Speed in Feet per Minute, $\sigma$. |
|---|---|---|---|---|---|---|---|---|---|---|---|
| | Indic. | Brake. | Engine Friction. | Indic., $p$. | Brake, $\eta p$. | Mechanical, $\eta$. | Abs. Indic. Th. Eff., $\epsilon$. | H. | $h$.* | H′. | |
| Coal gas .. | 111·2 | 94·2 | 17 | 95 | 80·5 | 0·847 | 0·34 | 7480 | 51·8 | 8850 | 770 |
| Benzol .. | 99·5 | 82·5 | 17 | 85 | 70·5 | 0·830 | 0·34 | 7480 | 46·3 | 9000 | 770 |
| Producer gas from anthracite | 93·7 | 76·7 | 17 | 80 | 65·5 | 0·819 | 0·33 | 7720 | 44·9 | 9425 | 770 |
| Producer gas from coke .. | 88·0 | 71·0 | 17 | 75 | 60·5 | 0·807 | 0·33 | 7720 | 42·2 | 9575 | 770 |

* See p. 30.

**Large Horizontal Engines.**—The largest type of four-stroke horizontal engine is the double-acting tandem Continental pattern which has been developed by the Nuremberg (M.A.N.), Ehrhardt & Schmer, Deutz, and Haniel & Lueg, &c., companies, and which are or have been built to a limited extent in Great Britain by, among others, the Lilleshall Company, Vickers, Galloways, and Richardsons Westgarth.

The double-acting four-stroke tandem engine confers the very important advantage in respect of steadiness of running that every stroke is a working stroke.

An external view of one of these large engines, giving 1200 b.h.p. at 125 r.p.m. and direct-coupled to an alternator, is shown in fig. 13. The two enclosed cylinders are in line, with their water-cooled pistons mounted on a common piston-rod, to one end of which is attached a massive connecting-rod of the usual type, by which the single-throw crank-shaft is turned. The large fly-wheel is supported by a substantial outer bearing and forms the alternator rotor, the pole-pieces being arranged at equal intervals around its periphery. A section through one cylinder is shown in fig. 14. The cylinder itself is a symmetrical casting AA, having the liner integral with the jackets and valve boxes; it is of as simple and uniform a section as possible in order to avoid distortion due to heating when at work. Each end is closed by a cast-iron water-cooled cover of very deep section through which the hollow piston-rod CC passes, gas-tightness being preserved by stuffing-boxes fitted with rings alternately of cast iron and white metal, each in three parts, pressed against

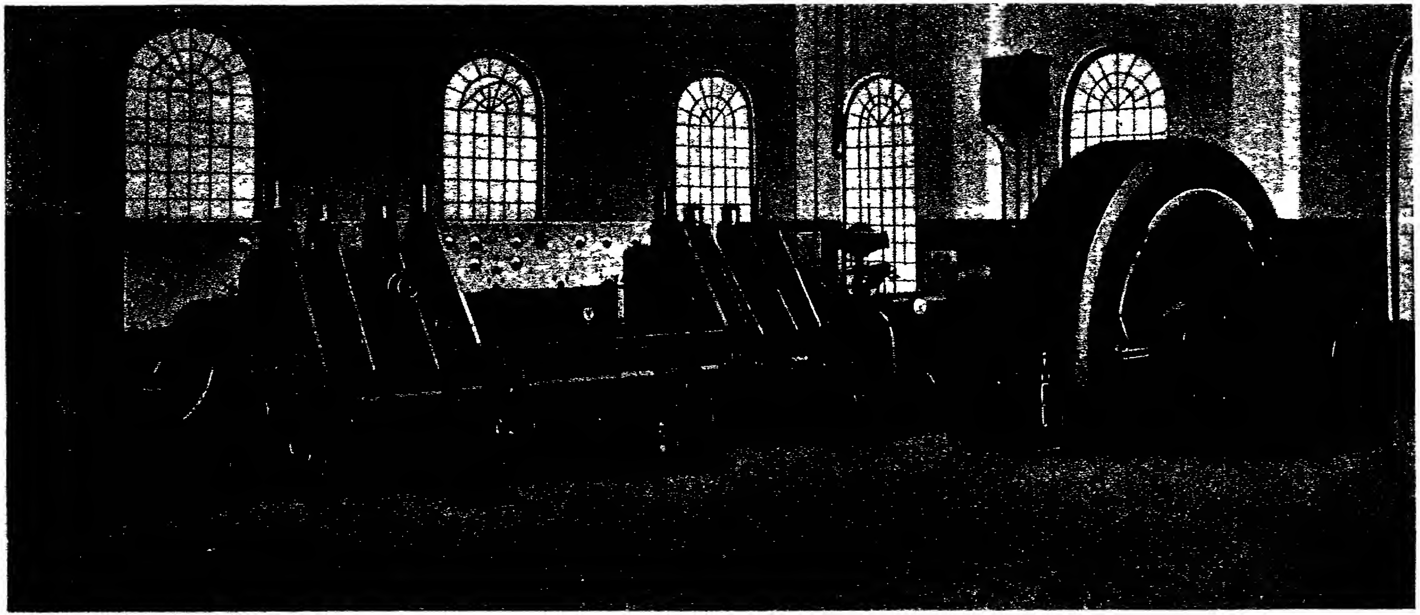


Fig. 13.—Typical Four-stroke Horizontal Double-acting Tandem Continental-pattern Gas-engine as developed by Nuremberg (M A N ), Richardsons Westgarth, and others

the rod by small spiral springs; the wear is confined to the white-metal rings, which are easily renewed when necessary. Lubricating oil is delivered under pressure to the centre of each stuffing-box.

The two water-cooled pistons, one of which is shown at BB, are simple hollow castings, each having six cast-iron spring rings; the cooling water is supplied and removed through the hollow piston-rod. It has already been remarked that piston friction, in general, constitutes a large proportion of the internal engine resistance; in the Nuremberg engines each piston is easily adjustable to the exact centre of its own cylinder. The piston-rod is carried by three white-metal lined well-lubricated external crossheads, two

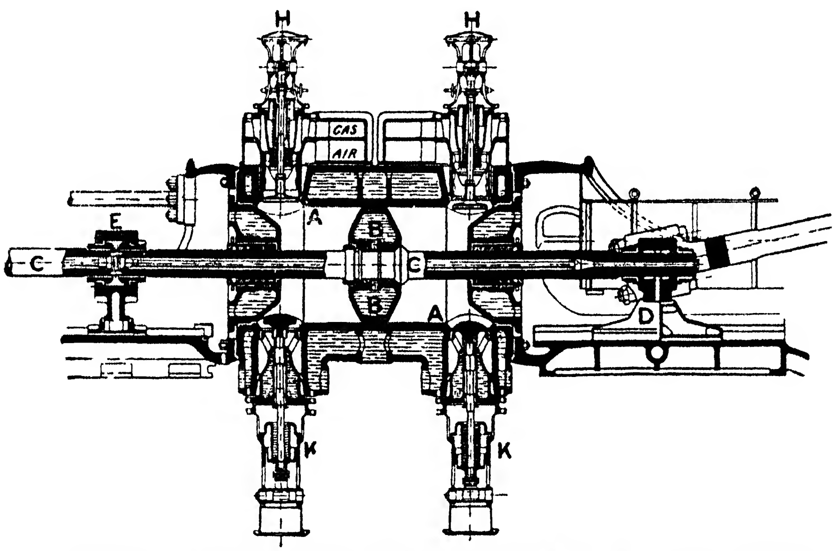



Fig. 14.—Section through one Cylinder of Four-stroke Double-acting Tandem Engine (Continental Pattern)

of which, D and E, are shown in fig. 14; the rod is given a slight upward camber when free, so that when loaded with the two heavy water-filled pistons it becomes quite straight; in this way the pistons are caused to " float " in their cylinders, and stuffing-box friction is minimized, thus very materially reducing the internal resistance of the engine and consequent creation of heat by piston-friction.

The gas-and-air-mixture inlet valves are combined as indicated at H, H, the air supply being by a sleeve rigidly connected to the gas and inlet valves. The governor varies the lift of the combined valves, and the speed regulation is thus effected by the " quantity " method, the " quality " of the mixture remaining practically constant at all loads; when the gas varies in composition the mixture quality is capable of ready adjustment by hand while the engine is running.

The exhaust valves K, K are provided with heavy heads to prevent overheating and " burning ", and it will be noted that their seats and stems are well water-cooled.

Ignition is effected by two, and sometimes three, low-tension firing-plugs in each end of each cylinder; the time of ignition can be adjusted, singly or collectively, while the engine is at work. Lubrication is forced, separate small oil-pumps being provided for the cylinders, stuffing-boxes, and exhaust-valve stems. Each cylinder is furnished with three points of supply, one being at the top and two at the sides. A large oil-tank is installed above the engine, and from this the oil is taken by the pumps, &c. Surplus oil drains into the basement of the engine-house, where it is filtered and then returned to the tank by a pump driven from the crank-shaft. Rollers and cams are grease-lubricated, which is a quite efficient method for large slow-speed engines.

**Cooling.**—As indicated in fig. 14, very ample jacket spaces are provided to which access is obtainable for cleaning through several hand-holes in the outer walls. For the cylinder barrels and covers the cooling water is supplied at a pressure of about 15 lb. per square inch, but for the pistons and piston-rod a water pressure of 50 to 60 lb. per square inch is necessary to ensure satisfactory circulation, on account of their reciprocating motion. When water of this pressure is available all delivery points are supplied from a common main; otherwise a separate pump, driven from the crank-shaft, is provided for the piston cooling water. This water is conveyed to and from the pistons through the hollow piston-rod, to the ends of which water connections of the swinging link type are fitted; the circulation should be such that the outlet water temperature is about 120° F.

Each cylinder is furnished with an open water tank into which the several water outlets deliver in full view of the attendant; each outlet is provided with a thermometer and regulating valve so that the temperature of the water can be adjusted as desired for every part of the engine. A master cock in the main supply pipe enables the whole of the water to be shut off when the engine is stopped.

In the very large Lilleshall Nuremberg engines installed at Kamata (Japan) the cooling water supply is at the rate of 8 gall. per brake horse-power hour, and the rise of temperature is from 60° F. to 105° F.; thus 3600 B.Th.U. are removed per brake horse-power hour by the jacket water, &c. Cooling towers are used by which the actual loss of water by evaporation is reduced to about 0·2 gall. per brake horse-power hour.

The Kamata engines are started by compressed air stored at 300 lb. per square inch, in steel receivers each 4 ft. in diameter and 12 ft. long, one to each engine. The four receivers are supplied by two motor-driven air compressors cooled by a water-supply from the cylinder main. Each of these huge engines has a bore of 47¼ in. and stroke of 51⅛ in.; running at 94 r.p.m. an output of 2130 b.h.p. is stated to be obtained.

**Vickers-Nuremberg Type Engines.**—In 1919 Messrs. Vickers (Barrow) completed eight large horizontal double-acting four-stroke tandem engines for the Appleby Iron Co., Ltd. Each of these has a bore of 43·3 in and stroke of 47·3 in., the piston-rod diameter being 10·75 in. The engines are used for driving electric generators and for blowing purposes. Driving generators, the output of each engine is about 1800 b.h.p. at 100 r.p.m., while when used for blowing the speed is reduced to 80 r.p.m., giving then an output of 27,000 c. ft. of free air per minute delivered against a pressure of 12·5 lb. per square inch.

The engines are worked on blast-furnace gas, and a trial of the first showed that it ran quite satisfactorily at an output ranging from 480 to 1310 kw., corresponding, roundly, to from 725 to 2000 b.h.p. The following average results were obtained:*

British thermal units of heat supplied to engine per brake horse-power hour:

| | |
|---|---|
| At full load .. .. .. .. | 9350 |
| At three-quarter load .. .. .. | 11,600 |
| At half load .. .. .. .. | 12,500 |
| Compression pressure .. .. .. | 137·5 lb. per square inch |
| Maximum pressure .. .. .. | 264·0 „ „ |
| Mean indicated pressure ($p$) .. .. | 65·5 „ „ |

Per 8-hour shift each engine used about 2 gall. of lubricating oil.

The exhaust gas temperature, taken at the first bend after the exhaust casing, was 813° F. at full load and 790° F. at three-quarter load. If $\delta$ denote the piston-rod diameter in inches, then as there are two impulses per revolution it is obvious that equation (4) of p. 30 becomes for these tandem double-acting four-stroke engines:

$$\text{B.H.P.} = 3{\cdot}968(d + \delta)(d - \delta)sn\eta p \times 10^{-6} \ldots\ldots\ldots (10)$$

The normal maximum output of 1800 b.h.p. at 100 r.p.m. gives, from this equation, a value of 54·5 lb. per square inch for the brake mean effective pressure $\eta p$. Comparing this with the average mean indicated pressure of 65·5, given above, shows an average mechanical efficiency, $\eta$, of about 83 per cent. Also, by equation (5), the piston speed at 100 r.p.m. is 790 ft. per minute; and by equation (7) the absolute brake thermal efficiency at full load is $\frac{2545}{9350} = 0{\cdot}272$, the corresponding indicated absolute thermal efficiency being $\frac{0{\cdot}272}{0{\cdot}83} = 0{\cdot}328$. And, lastly, by equation (9) the heat supply per cubic foot of swept working stroke at full load was roundly 41 B.Th.U. (p. 11), the value of $\eta p$ for 2000 b.h.p. being 60·5 lb. per square inch.

Between 1908 and the outbreak of the 1914–18 war the Maschinenfabrik Augsburg-Nürnberg A. G. (usually referred to as the Nuremberg Company, or M.A.N.), and their licensees, had built roundly 300 engines of this large double-acting type aggregating about 500,000 b.h.p. In addition to these, the other Continental makers and British builders had also produced many examples: while in America the Allis-Chalmers Company, the Snow Company, the American Westinghouse Company, and the William Tod Company had also built largely. In 1922 it was computed that the total output of

* *Engineering*, 30th July, 1920.

† Crook and Lyon-Ewan, *High-power Gas Engines* (Hobson, Salop).

existing engines of Nuremberg type amounted to between 2 and 3 millions of brake horse-power. One of the largest Continental installations is at Brückhausen, where Messrs. Thyssen have engined a station of 65,000 h.p. capacity. And in America Messrs. Allis-Chalmers have installed at Gary (Ind.), in one engine-house, a plant of roundly 60,000 h.p. capacity.

**Large Horizontal Two-stroke Engines.**—A general account of the two-stroke cycle, including the early Clerk and Day engines, has already been given in this article (pp. 5–8). Two-stroke cycle internal-combustion engines have also made great progress in recent years and are now very largely employed, not only in the smallest sizes for propelling motor-launches and bicycles, but also in the largest types of Diesel and other oil-engines.

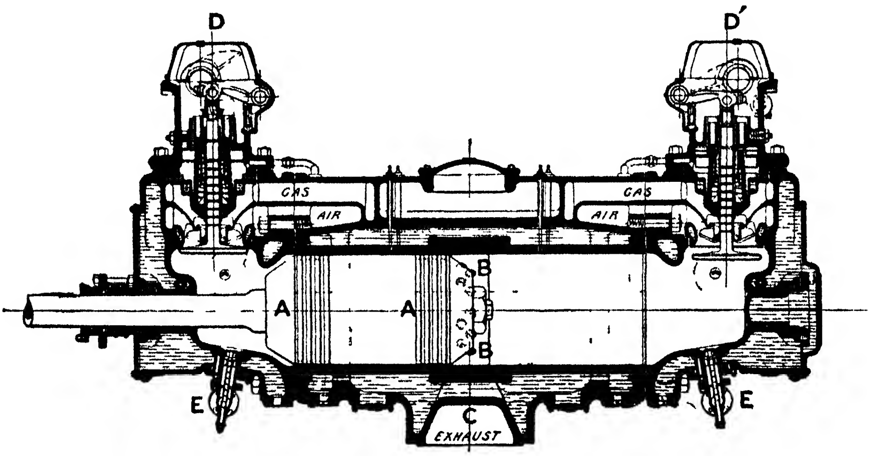



Fig. 15—Section of 600-b.h.p. Körting Engine Cylinder

We are here concerned more particularly with gas-engines, and the two outstanding examples of large horizontal two-stroke cycle gas-engines call therefore for attention. These are both developments of the Clerk cycle, and some account is first given of the well-known Körting engine which has been specially developed, since about 1900, by Messrs. Körting Brothers, of Hanover. The Körting is a single-cylindered double-acting two-stroke horizontal engine, and the piston thus receives two impulses per crank-shaft revolution, i.e. every stroke of the piston is a working stroke; a sectional view of a 600-b.h.p. Körting engine cylinder by Messrs. Mather & Platt is shown in fig. 15. The engine is specially characterized by the very long water-cooled piston A, fully half the length of the cylinder, which, in its extreme positions to right and left, uncovers a central belt of exhaust ports B, thus permitting the escape of the burnt gases into the exhaust chamber C. At each end of the long working barrel of the cylinder is a mixture inlet valve, D, D′, both operated, in the Mather & Platt design, by a single eccentric on the crank-shaft actuating a simple arrangement of levers above each valve As shown in fig. 15 the contents of the right-hand end of the cylinder are discharging themselves through the central ports B; simultaneously the inlet valve D′ is opened, admitting firstly a quantity of air alone, followed immediately afterwards by a charge of mixed gas and air; the air and gas are delivered to the valve boxes by separate double-acting pumps placed by the side of the engine, and driven from the crank-shaft. The piston moving towards the right first cuts off the exhaust ports B, and next compresses the mixture into the combustion chamber, where it is fired as usual, the resulting pressure driving the piston again towards the left. Simultaneously the same cycle of operations is performed in the left-hand end of the cylinder, and thus the piston receives a working impulse at every stroke.

Ignition is by low-tension magneto with two make-and-break plugs in each end of the cylinder placed about a diameter apart; one of these at each end is indicated in fig. 15. The double-acting charging pumps are driven by a crank about 110° ahead of the main crank, and their pistons have accordingly performed a portion of their delivery strokes when the main piston A is at the end of its stroke. The inlet valve is opened when the pressure in the cylinder has fallen to about that of the atmosphere, i.e. rather before the power piston has completed its working stroke, and the air-pump then delivers a scavenging charge of pure air into the cylinder, effectively clearing out the products of combustion and cooling any residual gas. In the gas-pump the pressure has not risen as the gas is by-passed during the first part of the gas-piston stroke, but very shortly after the scavenging air charge the gas valve opens, admitting gas to the entering air and thus delivering to the cylinder a mixed charge of gas and air through the inlet valve. Thus a considerable volume of air only at first enters the cylinder, cooling and displacing the exhaust gases, and the subsequent explosive charge of gas and air on entering meets only cooled gas, preventing risk of pre-ignition; loss of fresh gas through the exhaust ports B is also thus minimized. It will be noted that the gas and air are only mixed just above the inlet valve; in all large gas-engines it is very important to avoid the formation of explosive mixture in chambers or pipes, as a " back-fire " in a large gas-engine might in this case easily have serious consequences.

The air-pump always delivers a full charge of air whether the engine be running loaded or light, but the gas delivered is varied in amount by the governor gear by-passing the gas in the pump during a lesser or greater portion of its stroke. As the engine load decreases the gas delivery occurs later and later in the pump stroke, but *during* delivery the mixture of gas and air that enters the cylinder is of practically constant richness, as this depends only on the ratio of the gas and air pistons; thus the governing is of the quantity type, but with constant charge mass and full scavenging under all circumstances. Moreover the explosive charge entering last ensures that after compression there is always a readily ignitible mixture present in the neighbourhood of the firing plugs.

**Test Results.**—A test of a 600-b.h.p. engine having a bore of 29·7 in. and stroke of 55·1 in., the piston-rod being 8·1 in. diameter, running at

80 r.p.m. on producer gas from anthracite furnished the following results:*

| | |
|---|---|
| Indicated horse-power .. .. .. .. | 845·0 |
| Brake horse-power .. .. .. .. | 673·0 |
| Power absorbed by gas- and air-pumps.. .. | 88·0 h.p. |

The air-pump piston was 31·4 in. in diameter, and the gas-pump piston 27·6 in. in diameter; stroke of each pump was 42·5 in.; and both pumps were double-acting.

**Mechanical Efficiency.**—The indicated horse-power of 845, shown by diagrams taken from the power cylinder, is expended: (1) in overcoming the internal frictional resistances of the engine; (2) in working the gas- and air-pumps; and (3) in performing useful external work; this is the brake horse-power.

If the mechanical efficiency be computed as the value of the ratio b.h.p. to the i.h.p. we have $\eta = \frac{673}{845} = 0{\cdot}797$. On the other hand, if the power expended in working the pumps be deducted from the total indicated horse-power we have $\eta = \frac{673}{845 - 88} = 0{\cdot}889$; this latter value gives the efficiency of the mechanism apart from the power expended in charging and discharging the working cylinder, i.e. apart from the so-called " fluid resistances "; so long as it is made clear how $\eta$ is obtained it appears immaterial which method of computation be used.

**Power Formula.**—In the single-cylindered double-acting two-stroke Körting engine the piston-rod occurs on one side of the piston only; hence equation (10) of p. 35 is modified for this case and written:

$$\text{B.H.P.} = 3{\cdot}968\left(d + \frac{\delta}{\sqrt{2}}\right)\left(d - \frac{\delta}{\sqrt{2}}\right)sn\eta p \times 10^{-6}, \ldots\ldots (11)$$

and from this by aid of the results given above the brake mean effective pressure, $\eta p$, is found to be 45·5 lb. per square inch.

The consumption of anthracite, taken over a 23-hr. run, amounted to only 0·8 lb. per brake horse-power hour.

**Power and Weight.**—The largest Körting engines have been built on the Continent by the Siegener Company, giving outputs of 2000 h.p. from a single cylinder; the leading dimensions of these very large engines are:

Bore, 42 in.; stroke, 55 in.; normal full speed, 80 to 90 r.p.m.; the inlet valves are no less than 19·68 in. in diameter, with a lift of 3·14 in.

At Buffalo the American de la Vergne Company have installed Körting type engines aggregating 40,000 b.h.p. In all, it is safe to say that fully 300,000 b.h.p. of engines of this type are now in existence.

Starting is effected by means of compressed air maintained at a pressure of 150 to 300 lb. per square inch in steel receivers, e.g. by a separately driven two-stage compressor. The air starting valves are shown at E, E in fig. 15.

Körting engines built in Great Britain by Mather & Platt, and employed

* Junge, *Gas Power* (Hill Publishing Company, New York).

in driving electric generators, blast-furnace blowers, pumps, and mills, covered the following range:

KÖRTING ENGINES OF MATHER & PLATT

| Normal Full B.H.P. | Normal Maximum Revolutions per Minute. | Approximate Weights in Tons. | |
|---|---|---|---|
| | | Fly-wheel only. | Engine with Fly-wheel. |
| 400 | 100 | 10 | 45 |
| 500 | 100 | 13 | 50 |
| 600 | 100 | 16 | 75 |
| 700 | 95 | 20 | 85 |
| 1000 | 80 | 32 | 135 |

**The Duplex Engine.**—Several examples of this type of large gas-engine, due to Mr. A. E. L. Chorlton, were built by Messrs. Mather & Platt. The power unit includes two valveless two-stroke double-acting vertical cylinders in constant free communication with one another, somewhat

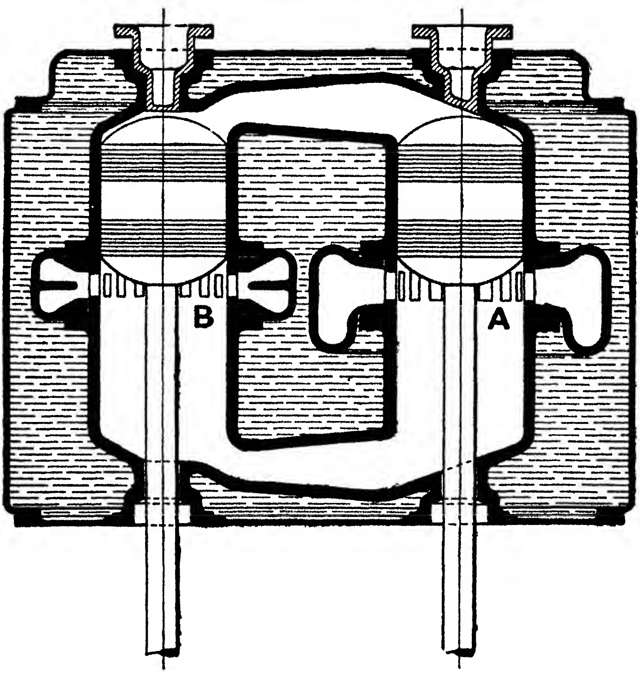



Fig. 16.—The Duplex Engine

as in the early Lucas "valveless" engine,* each cylinder being furnished with a long Körting-type piston; a diagrammatic view of the cylinder unit is given in fig. 16.

* Clerk and Burls, Vol. II, p. 762 (Longmans).

The two long pistons move almost, but not quite, together, there being an angle of 15° between the cranks. The leading crank is on the side of the exhaust ports A, to ensure that these ports are closed before the *gas*-pump delivers its charge thiough the inlet ports B. The charge is fired at or near the instant of maximum compression, and at about 0·8 of the stroke the right-hand piston overruns the ring of exhaust ports A, allowing the exhaust gases to escape into the atmosphere; at about 0·9 of the stroke the left-hand piston overruns the belt of inlet ports B, whereupon a charge of *air only* enters the cylinder and scavenges the residual exhaust gases. On the return stroke the right-hand piston first closes the exhaust ports A; the left-hand crank being behind, the inlet ports are still open, and during this period the *gas*-pump delivers its charge into the cylinder. On the closing of the inlet

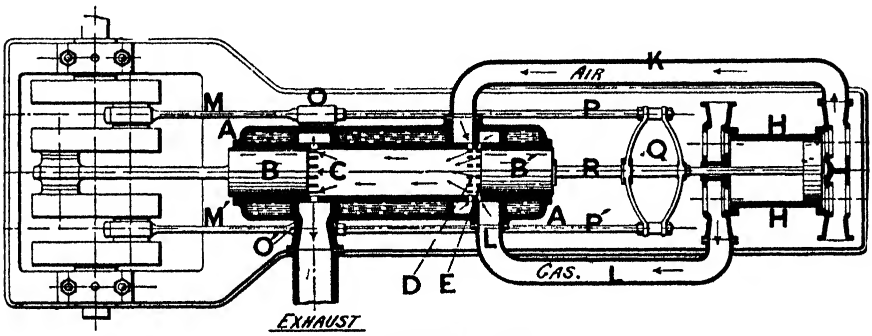



Fig. 17.—The Oechelhauser Engine

ports the charge is trapped and the ascent of the two pistons next compresses it into the common combustion chamber; the cylinders being double-acting, each power unit furnishes two impulses per revolution.

The water-cooled pistons are mild-steel forgings turned up in two sections, thus giving walls of small and uniform thickness. Ignition is by high-tension magneto with two plugs in each combustion chamber, one directly above, and one directly below each piston.

Around the power cylinders and enclosing also the gas- and air-pumps (not shown in the figure) is a simple light rectangular jacket flexibly attached to the cylinder castings to prevent expansion stresses from arising during working. Duplex engines have been built of 500 and 700 h.p; and a twin engine of 1000 h.p., formed by associating together two standard 500 h.p. units on a common bedplate, and driving one crank-shaft, was also constructed.

**The Oechelhauser Engine.**—The second outstanding type of large gas-engine working upon a modification of the Clerk two-stroke cycle is the well-known design first introduced by Dr. Oechelhauser, in Germany, at the end of the last century; a 600-h.p. Oechelhauser engine was at work in the Hoerde Ironworks in January, 1899. The type comprises a single-acting cylinder open at both ends and fitted with two pistons which alternately approach and recede from each other, the working mixture being contained between them; a diagrammatic section is shown in fig. 17.

The open-ended water-cooled working cylinder AA is fitted with two pistons B and B′ driving a three-throw crank-shaft; B drives its crank directly by a connecting-rod in the usual manner, while B′ drives the two side cranks through a crosshead Q, side rods P, P′ terminating in slides O, O′, and a pair of return connecting-rods M, M′.

The gas-and-air pump HH is directly driven from the crosshead Q, the front end of this pump dealing with gas only, and the rear end with air only as indicated in the diagram. The air and gas are separately stored during working in the receiver pipes K and L respectively, and only mix when actually within the engine cylinder.

The action is as follows: the engine being supposed running, the pistons B, B′ separate, and when nearing the end of the out-stroke, B first overruns the belt of exhaust ports C, permitting the burnt gases to escape through the exhaust, the pressure almost instantly falling to that of the atmosphere; the piston B′ next overruns the belt of air ports D, whereupon a scavenging and cooling charge of pure air rushes into the cylinder from the reservoir pipe K; B′ next overruns a second belt of ports E, permitting the charge of gas to enter the now cooled and air-charged cylinder. To minimize loss of fresh gas by "short-circuiting" through the exhaust ports C, it is provided that the maximum charge of gas and air admitted is only about 70 per cent of the full cylinder capacity.

The pistons next perform their in-strokes, cutting off in order the gas, air, and exhaust ports, and then compressing the fresh charge between them; at or about the instant of maximum compression the charge is fired, whereupon the pistons are driven apart and describe their working stroke; the cycle is thus of the two-stroke type.

It will be observed that the driving action on the crank-shaft is approximately that of a simple torque or "couple", and that reciprocating inertia is also nearly balanced; thus there is no cylinder, frame, nor large main bearing reaction to be resisted as is the case in all other engines; this enables engine weight to be kept low, and accordingly the Oechelhauser engines, although single-acting, weigh rather less per brake horse-power than the double-acting Körting engines of equal power when fly-wheel weight is excluded; when the fly-wheel is included the Körting is, however, frequently the lighter engine. The *relative* piston speed of the Oechelhauser engine is twice that of an ordinary engine of the same revolution speed, and the working gases are thus rapidly expanded, resulting in reduced heat losses to jackets.

A great difficulty in the early use of blast-furnace gas was that of freeing it sufficiently from dust;* modern cleaning apparatus overcomes this difficulty completely, but the Oechelhauser engine with its plain, smooth, valveless cylinder proved itself particularly well adapted to use the early dust-laden gas without risk of any working parts becoming choked with deposit from the fuel.

Messrs. Beardmore, of Glasgow, have built many Oechelhauser engines, and in their later designs considerably diminished size and weight, firstly by

* *Gas, Oil, and Air Engines*, B. Donkin, pp. 278–82 (Griffin).

placing the charging pump alongside the working cylinder and driving it directly from the crank-shaft, and secondly by greatly reducing the length of the working cylinder itself. The reduction in cylinder length was effected by fitting the spring rings into each end of the *cylinder* instead of in the pistons as usual; the pistons are thus plain cylindrical plungers which issue from the cylinder ends during a considerable portion of their stroke. Each piston was attached by a short, stout rod to an external crosshead, thus preventing "canting" in the cylinder, and also relieving it of all side pressure due to connecting-rod obliquity.

The capacities of the air and gas reservoirs were also increased, and the reservoirs themselves incorporated in the engine frame below the cylinder; the "stratification" of the mixture charge following the scavenging charge was considered to be thus better effected, and in consequence a larger charge of explosive mixture could be admitted without fear of loss through the exhaust; whence resulted a higher mean effective pressure and consequent increased power output from the cylinder.

In the earliest Oechelhauser engines the power absorbed by the pump amounted sometimes to as much as 25 per cent of the whole output, but in their improved design Messrs. Beardmore reduced this loss to about 7 per cent only.

The table following gives leading dimensions of the range of Oechelhauser engines built by Messrs. Beardmore:—

LEADING PARTICULARS OF BEARDMORE-OECHELHAUSER ENGINES

| Cylinder Bore in Inches. | Piston Stroke in Inches. | Revolutions per Minute. | Speed of Each Piston in Ft./Min. | B.H.P. | Approximate Weight in Tons. | |
|---|---|---|---|---|---|---|
| | | | | | Engine without Fly-wheel. | Fly-wheel. |
| 24 | 30 | 130 | 650 | 400 | 32 | 20 |
| 26 | 30 | 125 | 625 | 500 | 42 | 25 |
| 26 | 37·5* | 125 | 781 | — | — | — |
| 30 | 37·5 | 125 | 781 | 750 | 62 | 38 |
| 34 | 37·5 | 125 | 781 | 1000 | 80 | 50 |
| 42 | 51 | 94 | 797 | 1500 | 180 | 100 |
| 48 | 60 | 80 | 800 | 2500 | — | — |

* In this engine the back and front pistons had different stroke-lengths.

A test of an early 500-h.p. Oechelhauser engine by Professor Meyer in 1903 furnished the following results running on coke-oven gas having a (lower) calorific value of 382 B.Th.U. per cubic foot:

Mean effective pressure in cylinder ($p$) .. 69·3 lb. per square inch.
Revolutions per minute ($n$) .. .. .. 106·1

The cylinder bore was 26·6 in. and the average stroke of the two pistons 37·4 in. It is evident that the engine may be regarded as two single-cylindered single-acting two-stroke engines without piston rods, and hence from equation (11) of p. 38 we have for this case:

$$\text{I.H.P.} = 3{\cdot}968 d^2 snp \times 10^{-6}, \ldots\ldots\ldots\ldots\ldots(12)$$

and from this the above data give 772 as the indicated horse-power of the engine.

In this case the air-pump had a cylinder 44·9 in. in diameter and a stroke of 19·7 in., while the gas-pump had a 23·2-in. cylinder and 19·7-in. stroke; indicator diagrams showed that these two pumps jointly absorbed 79 h.p., or just over 10 per cent of the total indicated horse-power.

The engine drove a blower of 65-in. bore and 37·3-in. stroke, and the brake horse-power was taken to be the power shown from the blower diagrams; this amounted to 575 h.p. Hence the mechanical efficiency of the engine, $\eta$, is $\frac{575}{772} = 0{\cdot}745$, or $\frac{575}{772 - 79} = 0{\cdot}83$, according as the pump power is not, or is, deducted from the indicated horse-power in making the calculation.

The indicated horse-power was accordingly expended thus:

| Item. | Horse-power Absorbed. | | Per Cent. |
|---|---|---|---|
| In actual blowing .. .. .. | 575 | .... | 74·5 |
| By gas- and air-pumps .. .. .. | 79 | .... | 10·2 |
| In total friction of engine and pumps .. | 118 | .... | 15·3 |
| Indicated horse-power = | 772 | .... | 100 |

The gas consumption amounted to 17·04 c. ft. per indicated horse-power hour, corresponding to a heat supply, H, of 6509 B.Th.U. per indicated horse-power hour. Hence, by equation (7) of p. 30 the absolute indicated thermal efficiency, $\epsilon$, was $\frac{2545}{6509} = 0{\cdot}391$. While from equations (6) or (8) the heat supplied per cubic foot of swept stroke was $h = 32{\cdot}8$ B.Th.U. The absolute brake thermal efficiency, $\eta\epsilon$, was $0{\cdot}391 \times 0{\cdot}745 = 0{\cdot}291$, and heat supply per brake horse-power: $H' = \frac{6509}{0{\cdot}745} = 8737$ B.Th.U. The cooling water used per brake horse-power hour amounted to 5·9 gall.; the temperature on entering was 71·6° F. and on leaving 107·6° F.; thus $59(107{\cdot}6 - 71{\cdot}6) = 2124$ B.Th.U. of heat were carried off by the cooling water per brake horse-power hour, or 1582 B.Th.U. per indicated horse-power hour. Hence the 6509 B.Th.U. of heat supplied to the engine per indicated horse-power hour was expended as follows:

| Item. | B.Th.U. | | Per Cent. |
|---|---|---|---|
| In performing indicated work .. .. | 2545 | .... | 39·1 |
| In heating cooling water .. .. | 1582 | .... | 24·3 |
| In loss in exhaust, radiation, &c. .. | 2382 | .... | 36·6 |
| Total .. .. | 6509 | ... | 100 |

The lubricating oil used in the main cylinder amounted to 1·19 lb. per hour.

In a modified form the Oechelhauser type of engine was used in automobile service as a vertical engine working on the four-stroke cycle in the Gobron-Brillié motor-cars,* but the engine is equally workable as a gas-engine; a diagrammatic view of the power unit is given in fig. 18, and comprises two cylinders and four pistons acting on a three-throw crankshaft, the whole being enclosed in a light cast-iron casing. Two of these

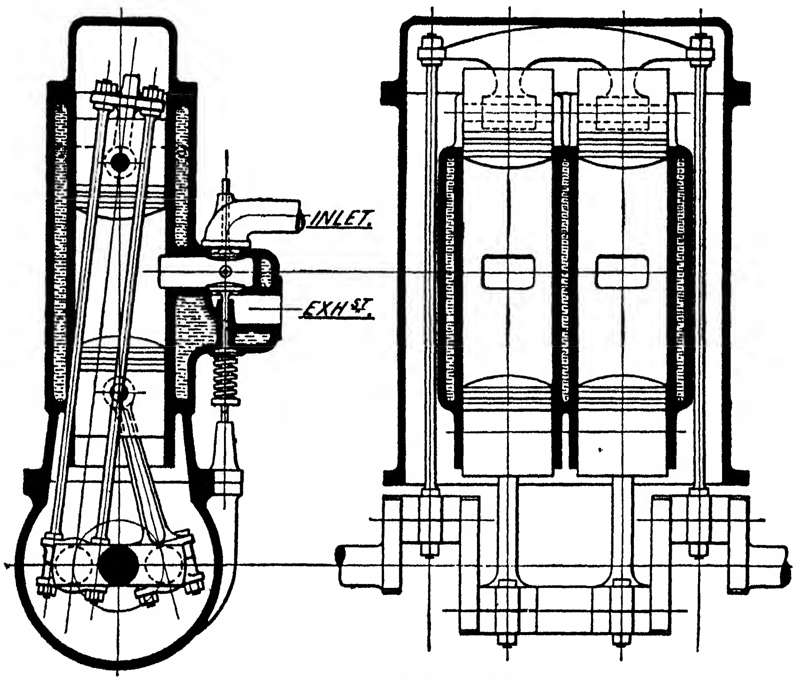



Fig. 18.—Gobron-Brillié Motor-car Oechelhauser-type Engine

units actuating one three-bearing six-throw crank-shaft are combined to form the standard 40-h.p. Gobron engine. The cranks of the second power unit are at 180° to those of the first; the remaining essential features are clearly indicated in the illustration.

An important development of the Oechelhauser engine is due to Mr. Fullagar, who has succeeded in retaining the characteristic advantages of balanced action, straight-through scavenging, and torque drive of this type with the addition of marked economy in space, weight, and cost, by dispensing with the rather cumbrous side rods, return connecting-rods, and two side cranks operated by the back piston of the normal Oechelhauser type; a diagram showing the distinctive features of the Fullagar engine (arranged to operate on oil on the Diesel cycle) is shown in fig. 19.†

The two-stroke power unit includes two vertical open-ended cylinders

* First introduced about 1899. † *The Engineer*, 17th July, 1914.

placed side by side; each cylinder contains two pistons, A and B, and C and D respectively, as in the Oechelhauser, but in the Fullagar these pistons are connected across by the diagonal tension rods GH and KL, thus enabling a normal flat two-throw crank-shaft EF to be employed. The obliquity of the diagonal rods is small, and the side thrusts arising therefrom are resisted by crossheads and guides as illustrated. As the engine is of the two-stroke type, each power unit furnishes two crank-shaft impulses per revolution. The charge of gas and air may be supplied by separate pumps driven from the engine, or by pumps formed by boxing in the top crossheads as shown, which gives these pumps a rectangular section; the charge is delivered to the engine at a pressure of about 3 lb. per square inch. In common with many two-stroke engines, the Fullagar may be run in either direction by giving it the necessary initial motion and suitably timing the electric ignition.

An early installation* of a 550-h.p. engine was that at the Gateshead station of the Newcastle Electricity Company. This comprised two power units with cylinders of 12-in. bore, each of the eight pistons having a stroke of 18 in. The normal speed was 250 r.p.m., corresponding to a piston speed of $\frac{250 \times 18}{6} = 750$ ft. per minute, and a *relative* piston

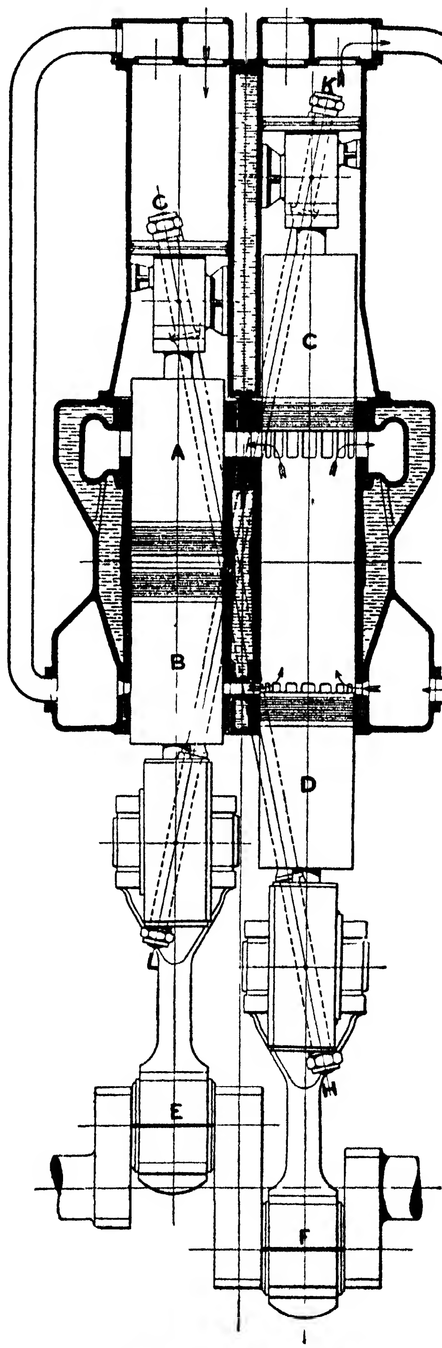


Fig 19.—The Fullagar Engine

* About 1914.

speed therefore of 1500 ft. per minute. A 30-hr. test by Professor Hopkinson showed an absolute indicated thermal efficiency of $37\frac{1}{2}$ per cent, and brake thermal efficiency of about 30 per cent. The mean effective pressure, $p$, was 67 lb. per square inch at full load, whence by aid of equation (12) of p. 43—the constant being now $4 \times 3{\cdot}968$—we have 690 as the value of the indicated horse-power. The gas- and air-pumps of this early engine absorbed more power than necessary.

A Fullagar engine, using heavy oil in the Diesel manner, was installed in 1922 in the power station of the English Electric Company at Rugby, and proved so reliable and satisfactory that the building of the type was taken up by this company, who have installed engines in a number of power stations; they are particularly suitable for driving alternators and direct-current generators.

Notable cases are of 750 b.h.p. units at Clacton-on-Sea and Worthing, while in Bermuda and the Sudan units direct-coupled to alternators proved so satisfactory that in each case a second engine of 1125 b.h.p was installed; these are three-unit, six-crank designs, running at 250 r.p.m. Messrs. Cammell Laird have also built marine Fullagar engines, and two 500-b.h.p. sets were installed in the 6000-ton twin-screw motor ship *Malia* in 1921; on trial, using Diesel oil as fuel, these two engines jointly developed 1100 b.h.p. at 125 r.p.m. on a consumption of 0·42 lb. per brake horse-power hour. Other trials, made at Birkenhead, on a 2-unit (4-cylinder) engine of 18·5 in. bore, each of the eight pistons having a stroke of 25 in. (i.e. a *relative* stroke of 50 in.), running at 120 r.p.m. and rated at 1000 b.h.p., gave an output of roundly 1200 b.h.p. on a consumption of only 0·4 lb. of 0·94 specific gravity " boiler oil " per brake horse-power hour.

**Large Vertical Engines.**—It has already been pointed out that the urgent necessity of economy in bulk and weight has stimulated the development of the multi-cylindered vertical engine, and to these considerations may be added also that of obtaining a small cyclic speed fluctuation without recourse to excessive fly-wheel weight. The large slow-running horizontal type with enormous crank-shafts, in some cases exceeding even 30 in. in diameter; cylinders each weighing 25 tons; fly-wheels of 100 tons weight, and upwards of 20 ft. in diameter; and other details on the same Cyclopean scale, require very special plant for their manufacture, and are, in general, costly and difficult to produce, to handle, and to transport.

For particulars of the vertical tandem four-stroke single-acting gas-engines of the National Company, see page 27. A transverse section through one pair of associated cylinders is now given in fig. 20.

The unit comprises two slightly offset cylinders A and B placed one above the other, the upper having a bore of 23 in. and the lower of 22 in.; in each works an uncooled single-acting cast-iron piston of 24-in. stroke; the two pistons are rigidly connected by a hollow cast-iron sleeve through which passes a long nickel-steel bolt, holding the three pieces tightly together; gas-tightness is preserved by five cast-iron spring rings in each piston as shown, while near the bottom of the skirt of the lower piston a sixth or

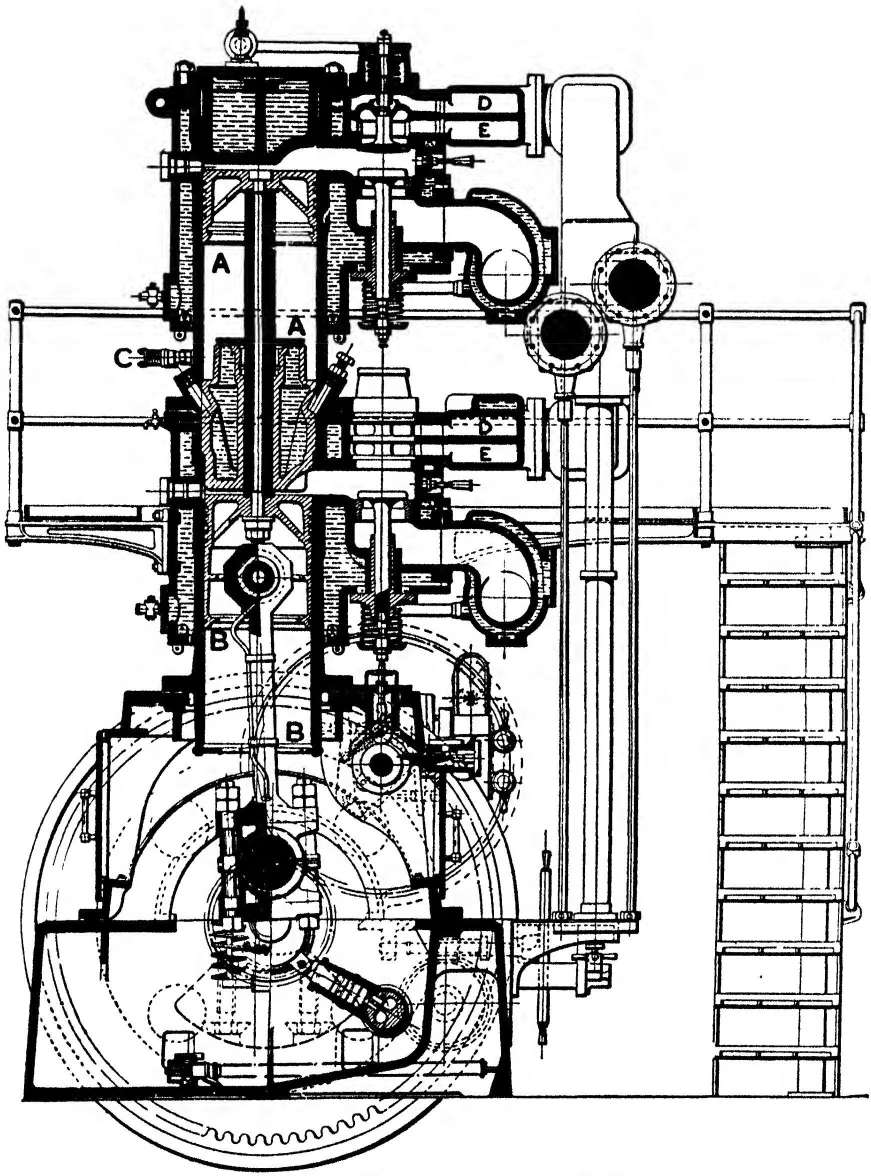



Fig. 20.—Four-stroke Vertical Single-acting Tandem National Gas-engine

" scraper " ring is fitted to prevent oil from the crank chamber working upwards.

The lower portion of the upper cylinder contains air only; this is alternately compressed and expanded during the running of the engine, and by its cushioning action softens the motion by assisting in the reversal of the piston motion when passing the bottom of the stroke; the cranks are also balanced in the usual manner Experience has shown that in some conditions of running combustible mixture may be formed and fired in this air chamber, and accordingly small relief valves, one of which is shown at C, are fitted to release any undue pressure thus created. Crank chambers of enclosed internal-combustion engines are also liable to become filled with explosive vapours when in service, and should therefore never be examined with a naked light.

Each cylinder working barrel, with its jacket, is a separate casting; the lower portions of the barrels are not jacketed.

The inlet and exhaust valves open into a common port and are placed one above the other, the inlets being uppermost; the valve seats are separate rings easily removed and renewed when necessary. The gas and air passages are formed by the partitioned casings D, E; the governing system and arrangement of gas and air valves are described in p. 20.

Forced lubrication is employed for all reciprocating and rotating parts, the oil being supplied by engine-driven valveless pumps at a pressure of about 20 lb. per square inch; the used oil drains into a crank-case sump through a strainer, after which it is again delivered to the bearings, &c., by the pump. The lower cylinder is lubricated by the oil splashed from the gudgeon bearing, while the upper cylinder and the metallic packing of the piston-rod gland are supplied from a sight-feed lubricator fed by a small engine-driven oil-pump. Starting is effected by means of compressed air. Ignition is by high-tension magneto, two firing plugs being fitted in each cylinder.

**Performance.**—From tests made on a three-unit engine, i.e. of 750 b.h.p., the following results are derived:

TEST RESULTS FROM A 3-CRANK 750-B.H.P. VERTICAL TANDEM NATIONAL ENGINE

Cylinders, 23 and 22 in. diameter; piston rod, 6 in. diameter; stroke, 24 in. Fuel: Producer gas from anthracite

| Item. | At Overload. | Full Load. | Three-quarter Load. | Half Load | Quarter Load. |
|---|---|---|---|---|---|
| Mean effective pressure from diagrams (lb./sq. in.) | 73·7 | 64·3 | — | 37·3 | — |
| Revolutions per minute .. .. .. .. .. | 203 | 204 | 204 | 206 | 206 |
| I.H.P. from equation 13 .. .. .. .. | 1040 | 915 | — | 535 | — |
| B.H.P. from electrical measurements .. .. | 903 | 748 | 571 | 380 | 208 |
| Mechanical efficiency .. .. .. .. | 0·87 | 0·82 | — | 0·71 | — |
| Heat value of gas (lower), B.Th.U./c. ft. .. .. | 151 | 148·6 | 139·2 | 149·8 | 149·8 |
| Cubic feet of gas per B.H.P. hour .. .. .. | 70·2 | 66·3 | 67·7 | 84·9 | 124·2 |
| B.Th.U. per B.H.P. hour .. .. .. .. | 10,400 | 9860 | 9420 | 12,720 | 18,580 |
| B.Th.U. per I.H.P. hour .. .. .. .. | 9050 | 8085 | — | 9030 | — |
| B.Th.U. per cubic foot of swept stroke .. .. | 48·5 | 38·0 | — | 24·5 | — |
| Absolute indicated thermal efficiency .. .. | 0·282 | 0·315 | — | 0·282 | — |
| Absolute brake thermal efficiency .. .. .. | 0·245 | 0·258 | 0·270 | 0·200 | 0·137 |
| Cooling water (gallons per B.H.P. hour) .. .. | 6·96 | 6·55 | 8·6 | 11·3 | 19·25 |
| Temperature of gas supplied to engine .. .. | 63° F. | 71° F. | 73° F. | 72° F. | 72° F. |
| Vacuum in inlet pipe (in. mercury) .. .. .. | 2·0 | 5·5 | 8·5 | 13·5 | 15·5 |
| Engine friction horse-power .. .. .. .. | 137 | 167 | — | 155 | — |
| Piston speed (feet per minute) .. .. .. .. | 812 | 816 | 816 | 824 | 824 |

At full load the cooling water temperature at inlet was 58° F., and at outlet was 100° F.; hence 65·5(100 − 58) = 2751 B.Th.U. of heat were lost to the cooling water per brake horse-power hour, or 2751 × 0·82 = 2250 B.Th.U. per indicated horse-power hour. So that at full load the 8085

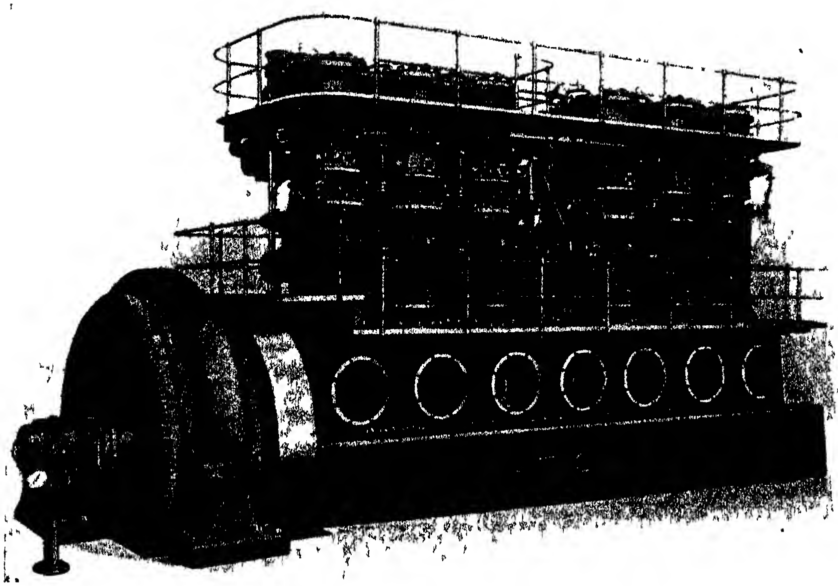


Fig 21.—The National Gas-Engine

B.Th.U. of heat supplied per indicated horse-power hour was expended as follows:

| Item. | Heat Expended. | Per Cent. |
|---|---|---|
| In producing indicated work .. | .. 2545 .... | 31·5 |
| In heating jacket water .. | .. 2250 .... | 27·8 |
| In exhaust gases, radiation, &c. | .. 3290 .... | 40·7 |
| Total .. | .. 8085 .... | 100 |

The average engine friction horse-power appears as 153. The engine was run light at 206 r.p.m., and was then found to consume 16,200 c. ft. of gas per hour, corresponding on this estimate to 106 c. ft. of gas per engine friction horse-power hour running light.

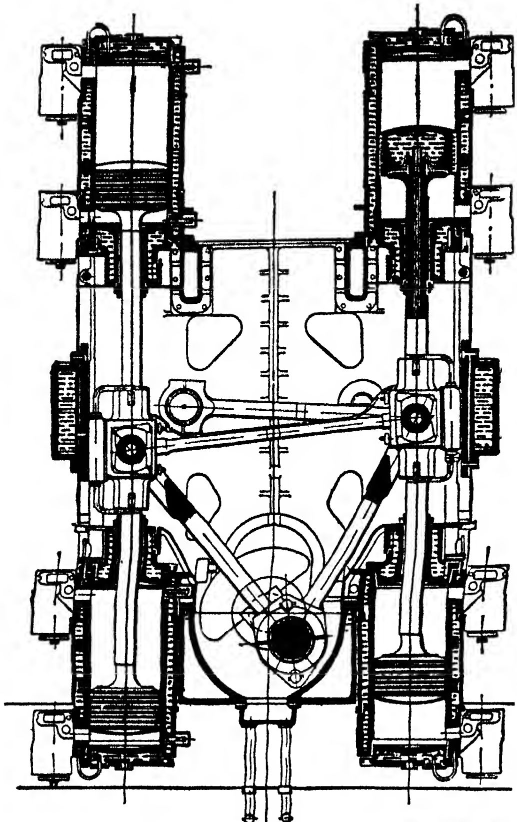
**Power Formula.**—The unit of these engines may clearly be regarded as a single-cylindered single-acting four-stroke cycle engine of piston area $\frac{\pi}{4}\left(d_1^2 + d_2^2 - \delta^2\right)$ square inches, where $d_1$, $d_2$, and $\delta$ are the diameters of pistons and piston-rod respectively; hence equation (1) of p. 30 becomes for this case:

$$\text{I.H.P.} = \text{N} \times 992\,(d_1^2 + d_2^2 - \delta^2)\,snp \times 10^{-9}, \ldots\ldots(13)$$

where N is the number of associated units forming the engine, viz. three in the case just considered.

Experience shows that when using producer gas from coke the power output is about 95 per cent of the above, while with blast-furnace gas the power is reduced to about 85 per cent.

Any tendency to pre-ignition during prolonged running at full load is quenched by Sir Dugald Clerk's method of introducing into the inlet pipe, at atmospheric pressure, from 10 to 20 per cent by volume of cooled exhaust gases. This addition of inert gases reduces inflammability without reducing the total mass of the charge; it is easily effected, and by its aid the maximum practicable power of the engine is materially increased.



From *The Engineer*.

Fig. 22.—The Vertical Premier Engine

**The Vertical Premier Engine.** — In 1920 the Premier Gas-engine Company introduced a large gas-engine embodying some unusual and interesting features. The power unit comprises four double-acting four-stroke engines arranged vertically in the manner shown in fig. 22, all acting upon a single crank through a triangular connecting-rod as shown; the engine runs normally at 125 r.p.m.

The two upper angular points of the connecting-rod triangle are furnished with gudgeon-pins working in short transverse slides, one in each of the piston-rod crossheads; the right-hand gudgeon-pin is connected to the free ends of a pair of radius rods which take the side thrust, and define the position of the triangular connecting-rod. By this arrangement the pressure due to connecting-rod obliquity—which ordinarily produces considerable friction on the pistons or crossheads—is taken by the radius rods, the only side thrust upon the piston-rod crossheads being that due to the small friction of the gudgeon-pins in their transverse slides; the whole motion here, due to the circular movement of the ends of the radius rods, amounts only to about 1 in. The Premier Company claims that by this arrangement the frictional loss in converting the piston motion into that of the crank-shaft is reduced to a minimum.

**Valves.**—A separate valve-box is fitted to each end of each cylinder, and each box contains three valves, viz. the air and gas valves above and the exhaust valve below; between them is the port opening into the cylinder. The valves are actuated by horizontal rocking levers and grooved cams carried on two vertical bevel-driven shafts placed at the sides of the engine, and running in plain footstep bearings. The valves are opened *and closed* positively by the rocking levers, proper closing being ensured by making the stroke of the levers about $\frac{1}{8}$ in. too long, and taking up this excess by the compression of short springs housed in the connection between the valve spindle and lever.

Governing is on the quality method; the supplies of gas and air are delivered to their respective valves through two rectangular trunks, so that mixture occurs only in the port; a butterfly valve in the gas trunk, controlled by the governor, adjusts the gas supply to the engine load.

Low-tension ignition is fitted, duplicate make-and-break igniters being provided in each valve box; the igniter rockers are operated by tappets and link-work from the valve cams. Starting is effected by compressed air admitted to one pair of cylinders through valves operated from the half-speed shaft; a pressure of 180 lb. per square inch is employed.

**Performance.**—The 1000-b.h.p. engine comprises a single unit of four cylinders each 24·75 in. in diameter, the stroke being 30 in., and normal full speed 125 r.p.m. Equation (1) of p. 30 obviously becomes for this case:

$$\text{I.H.P.} = 4 \times 2 \times 992\, d^2snp \times 10^{-9}, \qquad (14)$$

and the makers state that when using producer gas a mean effective pressure, $p$, of 66 lb. per square inch is obtained, and a brake horse-power of roundly 1000. The corresponding indicated horse-power from equation (14) is 1200, giving a mechanical efficiency of 83·3 per cent. With blast-furnace gas it is stated that 900 b.h.p. is obtained, the mean effective pressure being then about 59½ lb. per square inch.

The Premier Company has also designed larger engines, having 39-in. cylinders, for an output of 3000 h.p. per unit of four cylinders; and a two-unit eight-cylinder two-crank design for an output of up to 6000 h.p. The single crank and connecting-rod, and the considerable amount of duplication of parts in this design tend to economy in production; the over-all dimensions of the 1000-h.p. unit, including a coupled generator, are 18 ft. × 25 ft. × 20 ft. in height from floor-level; this compares very favourably with the space occupied by an equal engine of the tandem horizontal type. The essential characteristic of the design is the triangular connecting-rod and the considerable obliquity of its action; but a 1000-h.p. engine installed at the works of Messrs. A. Hickman, of Wolverhampton, in 1920 was reported upon after the first six months of running in regular service as having given no trouble whatever.

**Humphrey Gas Pump.**—In this pump, which is the invention of H. A. Humphrey, the water acts as the piston, and the explosion of a mixture of gas and air in the combustion chamber forces a column of water in the cylinder (which is vertical) downward and up to a much greater height at the far end of a pipe communicating with the cylinder. Owing to the inertia of the moving body of water, expansion in the cylinder takes place to a pressure below atmospheric, and water from the lower level enters the cylinder through a belt of water valves in the lower portion. At the same time scavenging air enters the cylinder through valves at the top.

On the return swing (inward) of the water in the pipe and cylinder exhaust takes place. A third swing (outward) of shorter stroke gives the suction stroke, during which the mixture of gas and air passes into the cylinder, and the return swing (inward) gives the compression stroke, at the end of which ignition takes place.

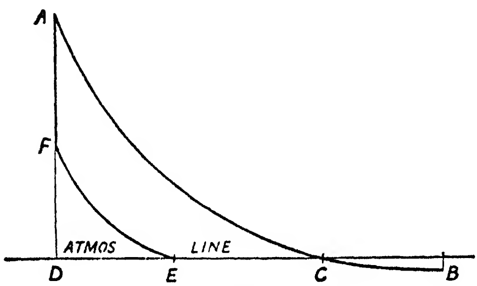


Fig. 23

Arrangements are made for locking the respective valves during the parts of the cycle where they are not required to open.

Fig. 23 shows the nature of the pressure variations in the chamber during the cycle.

AB is the expansion stroke (exhaust valves closed; gas valves locked; air-scavenge valves closed from A to C, open from C to B).

BCD is the exhaust stroke (exhaust valves open; gas valves locked; scavenge valves closed).

DE is the induction stroke (exhaust valves locked; gas valves open; scavenge valves locked).

EF is the compression stroke (all valves closed).

Four large pumps and a smaller one were installed at Chingford in 1912 for the Metropolitan Water Board and have been in regular operation since. The lift is about 30 ft. and the delivery 33,000 gallons per min. in the case of the larger pump, and 18,000 gallons per min. for the smaller pump. The gas used is obtained from Dowson producers using anthracite, and tests showed a fuel consumption of under 1 lb. of anthracite per pump horse-power hour.

These pumps can only work at a slow speed, owing to the large mass of water operating as the equivalent of the piston. In the above case the pumps work at about 9 cycles per minute, so that the weight of water delivered per cycle by the larger pumps is about 15 tons, or about 530 c. ft. Their combustion chambers are 7 ft. in diameter.*

* An investigation of the theory of the Humphrey pump is given by Dr. W. J. Walker in *Engineering*, Feb. 11, 1921.

**Conclusion.**—The evolution of the gas-engine has been here briefly sketched from the earliest crude military applications of six hundred years ago to the many varied and highly efficient mechanisms of the present day. Utilizing, in addition to petrol, oil, and other liquid and gaseous fuels, a great number of formerly wasted industrial refuses, the internal-combustion engine has proved a most potent and far-reaching factor in the amelioration of the material conditions of modern life.

In recent years, however, the steady progress in reliability of the compression-ignition oil engine, using heavy oil as fuel, has stimulated its use in many cases where formerly gas engines were installed, so that subsequent to 1930 the demand for gas engines for relatively small powers has declined very considerably.

A gas-engine of 30 b.h.p., using 18 c. ft. of coal gas per b.h.p. hour at 4*s*. 2*d*. per 1000 c. ft., would cost 2*s*. 3*d*. per hour for fuel at full load. An oil engine, using 0·4 lb. of fuel oil per b.h.p. hour at 11*d*. per gallon, would cost 1*s* 5*d*. per hour for fuel at full load, the saving being 10*d*. per hour, so that in 500 hours' running at full load the total saving in fuel cost would be about £21. The actual saving would be less, as in most cases the engine would not be running at full load all of the time, but in any case it would be substantial enough to make the oil engine a formidable competitor. If a suction gas producer were installed the difference in fuel costs would be reduced considerably, but the cost of attendance and time required for starting up would still favour the oil engine in many cases.

# GAS PRODUCERS

BY

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# Gas Producers

**A Gas Producer** is an apparatus for producing inflammable gaseous mixtures from solid or liquid fuels. Thus widely defined, carburettors and vaporizers for petrol and paraffin, oil-gas making devices, town's (or coal) gas plants, and, in a lesser degree, coke-oven installations and blast furnaces are all included. The whole subject is very extensive, and is accordingly dealt with in many separate special treatises; but whenever reference occurs simply to "gas producers" it is usual to conclude that some apparatus is referred to specially made for the production of large quantities of inflammable gaseous mixture from some solid carboniferous fuel, and that the gas produced is intended primarily for power development or heating purposes.

As, however, town's (or coal) gas, in addition to lighting and heating, is widely used for power production in the smaller types of gas-engine, some reference to it is necessary in this article. Similarly coke-oven gas and blast-furnace gas though secondary, and formerly almost wasted products, are now of such importance as sources of power and heat that a brief account is given also of these gases. In the cases of town's gas and of coke-oven gas, coal is distilled in closed retorts by the application of external heat, whereas in all "gas producers" as ordinarily understood (and in blast furnaces), the carboniferous fuel is partly burned within the producer itself.

**Air.**—For the purposes of this subject it is sufficient to regard dry air as a mechanical mixture of nitrogen and oxygen in the following proportions:

By weight: 1 lb. of air contains 0·77 lb. of nitrogen and 0·23 lb. of oxygen.
By volume: 1 c. ft. of air contains 0·79 c. ft. of nitrogen and 0·21 c. ft. of oxygen.

Thus 1 lb. of oxygen is contained in 4·348 lb. of air; and 1 c. ft. of oxygen is contained in 4·762 c. ft. of air.

Also at 32° F. and 1 atmosphere pressure, 1 lb. of air occupies a volume of 12·387 c. ft.; or, reciprocally, 1 c. ft. of air then weighs 0·08073 lb.

If $v$ denotes the volume of 1 lb. of air in cubic feet when at pressure $p$ lb. per square inch and at an absolute temperature T° F.,* then always

$$pv = 0{\cdot}37T \ldots\ldots (1)$$

Air always contains some moisture, also traces of carbon dioxide and other gases, but these may here be neglected. It is, however, of interest to remark that air containing moisture is somewhat *lighter* than dry air (p. 60).

**Water.**—In liquid form 1 lb. of water at 32° F. has a volume of 0·01602 c. ft., so that 1 c. ft. weighs 62·4 lb. As 1 gall. of water weighs 10 lb., there are roundly 6¼ gall. of water in 1 c. ft. In gaseous form, as a saturated vapour, the relations between pressure and temperature are as given in Table I hereunder, which shows also the volume occupied by 1 lb. when just all evaporated.

TABLE I

DATA RELATING TO GASEOUS WATER FROM 32° F. TO 212° F.

| Temperature in Degrees Fahrenheit. | | Absolute Pressure in pounds per square inch, $p'$. | Cubic feet occupied by 1 lb., $u$. |
|---|---|---|---|
| Ordinary, $t$. | Absolute, $T = 460 + t$. | | |
| 32 | 492 | 0·089 | 3283 |
| 40 | 500 | 0·122 | 2455 |
| 60 | 520 | 0·253 | 1210 |
| 80 | 540 | 0·502 | 640 |
| 100 | 560 | 0·939 | 353 |
| 150 | 610 | 3·699 | 97 |
| 200 | 660 | 11·496 | 33·6 |
| 212 | 672 | 14·70 | 26·75 |
| 225 | 685 | 18·93 | 20·78 |
| 250 | 710 | 29·88 | 13·54 |

**Latent Heat.**—The reader is reminded that the British unit of heat, usually written 1 B.Th.U., is the quantity of heat necessary to raise 1 lb. of liquid water at about 39° F. through a temperature of 1° F., and that this quantity of heat corresponds to 778 ft.-lb. of mechanical work.

If 1 lb. of water in the form of saturated vapour (i.e. 1 lb. of just "dry" steam) be condensed to liquid at unchanged temperature, a quantity of heat termed the "latent heat of evaporation" is evolved by the vapour during condensation. Denoting the latent heat by L, its value in B.Th.U. per pound at any temperature $t$° F. (ordinary scale) may be found approximately from the empirical formula:

$$L = 1113 - 0{\cdot}7t. \ldots\ldots (2)$$

Thus for the range of temperature as in Table I:

* Absolute temperature = ordinary temperature + 460.

## TABLE II

### LATENT HEAT OF WATER IN B.TH.U. PER POUND

| Temperature, Degrees Fahrenheit, $t$. | Latent Heat, L |
|---|---|
| 32 | 1091 |
| 40 | 1085 |
| 60 | 1071 |
| 80 | 1057 |
| 100 | 1043 |
| 150 | 1008 |
| 200 | 973 |
| 212 | 965 |
| 225 | 956 |
| 250 | 938 |

**Air saturated with Water Vapour.**—The quantity of moisture that can be suspended in air depends only upon the temperature, and is estimated by aid of Dalton's law, according to which 1 c. ft. of saturated air at pressure $p$ lb. per square inch and temperature T° F. (absolute) consists of:

(1) 1 c. ft. of water vapour at pressure $p'$ (see Table I) corresponding to T° F. (absolute); this weighs $\frac{1}{u}$ lb.

(2) 1 c. ft. of air at pressure $(p - p')$ lb. per square inch and T° F. (absolute); this weighs $\frac{p - p'}{0{\cdot}37T}$ by equation (1).

Accordingly if $w$ denotes the weight in pounds of 1 c. ft. of saturated air at $p$ lb. per square inch pressure and T° F. (absolute) temperature, then

$$w = \frac{1}{u} + \frac{p - p'}{0{\cdot}37T} \quad \ldots \quad (3)$$

By aid of this formula the results given in Table III have been calculated. These are sometimes useful for reference in connection with the supply of steam-laden air to gas producers.

Figures for 100 c. ft. of saturated air are tabulated to avoid small decimal quantities. When at a sufficiently high temperature to behave sensibly as a perfect gas, water is 9 times as heavy as an equal volume of hydrogen; if this condition held, for example, at 32° F. and 1 atmosphere, 1 c. ft. of gaseous water would then weigh $9 \times 0{\cdot}005592 = 0{\cdot}050328$ lb.

Actually at 32° F. water is a saturated vapour whose pressure is but 0·089 lb. per square inch (see Table I). If it be imagined as behaving as a perfect gas at this temperature, 1 c. ft. at 32° F. and *1 atmosphere* pressure would weigh $\frac{14{\cdot}7}{0{\cdot}089 \times 3283} = 0{\cdot}05031$ lb.

## TABLE III

WEIGHT AND MOISTURE CONTENT OF 100 C. FT. OF SATURATED AIR AT 1 ATMOSPHERE PRESSURE

$p = 14{\cdot}7$ lb. per square inch throughout (see equation 3)

| Temperature in Degrees Fahrenheit. | | Weight of 100 C. Ft. of Saturated Air, in Pounds. | | | Volumes in Cubic Feet per Pound. | | Ratio of Density of Saturated Air to Dry Air. |
|---|---|---|---|---|---|---|---|
| Ordinary, $t$. | Absolute, T. | Weight of Moisture, $\frac{100}{u}$ lb. | Weight of Dry Air, $\frac{100(p-p')}{0{\cdot}37T}$ lb. | Total Weight in Pounds ($100w$). | Of Saturated Air. | Of Dry Air. | |
| 32 | 492 | 0·03046 | 8·0241 | 8·0546 | 12·415 | 12·387 | 0·998 |
| 40 | 500 | 0·04073 | 7·8779 | 7·9186 | 12·628 | 12·56 | 0·995 |
| 60 | 520 | 0·08264 | 7·5068 | 7·5894 | 13·176 | 13·09 | 0·993 |
| 80 | 540 | 0·15625 | 7·1042 | 7·2605 | 13·773 | 13·60 | 0·987 |
| 100 | 560 | 0·2833 | 6·6396 | 6·9229 | 14·445 | 14·10 | 0·976 |
| 150 | 610 | 1·0309 | 4·8729 | 5·9038 | 16·938 | 15·36 | 0·907 |
| 200 | 660 | 2·9762 | 1·3117 | 4·2879 | 23·322 | 16·62 | 0·713 |
| 212 | 672 | 3·7383 | — | 3·7383 | 26·75 | 16·92 | 0·632 |

**Elementary Fuels.**—There are but two combustible elements with which the engineer is concerned in this subject, viz. hydrogen and carbon. In all the solid fuels employed, and in all the inflammable gaseous mixtures produced, the quantities of these two elements contained in the one and the amount and manner of their occurrence in the other in gaseous and combustible form, are the two points of primary importance. It is accordingly necessary briefly to consider the behaviour, when burnt in air, of hydrogen and carbon, with some combinations of these together, and with oxygen.

**Hydrogen** is a very light, colourless, odourless, permanent, non-poisonous, inflammable gas, producing *water* when burnt in air. At a temperature of 32° F., and under a pressure of 1 atmosphere (14·7 lb. per square inch), 1 lb. of hydrogen occupies a volume of 178·83 c. ft.; or, reciprocally, 1 c. ft. weighs 0·005592 lb. 2 lb. of hydrogen unite with 16 lb. of oxygen, forming 18 lb. of water; thus 1 lb. of hydrogen forms 9 lb. of water. By *volume*, 2 c. ft. of hydrogen unite with 1 c. ft. of oxygen to form 2 c. ft. of water (in gaseous form); the volume shrinks from 3 c. ft. before combustion to 2 c. ft. after—the temperature and pressure being restored to their initial values. Further, in the combustion of 1 lb. of hydrogen to water it is found that roundly 62,000 B.Th.U. of heat are evolved, the 1 lb. of hydrogen being initially at 32° F. and 1 atmosphere, and the 9 lb. of water produced being reduced also to *liquid* at 32° F. and 1 atmosphere. This is termed the "higher calorific value" of the gas.

If the resulting water be reduced to 32° F. and 1 atmosphere, but *not* liquefied, its latent heat, amounting at that temperature to roundly 1090 B.Th.U. per pound (see equation 2), is not liberated; accordingly in this case the heat evolved by the combustion of the 1 lb. of hydrogen, termed its "lower calorific value", has the reduced value of $62{,}000 - 9 \times 1090 = 52{,}190$ B.Th.U. The heat evolved in the combustion of 1 c. ft. of hydrogen (at 32° F. and 1 atmosphere) is $0{\cdot}005592 \times 62{,}000 = 347$ B.Th.U., "higher" value, or $0{\cdot}005592 \times 52{,}190 = 292$ B.Th.U., "lower" value.

In Great Britain, America, and Germany "lower" calorific values of gaseous fuels are ordinarily used in the calculation of efficiencies; in France, however, "higher" values are usually employed. It will be seen on reference to Table IV that the difference between the two values is sufficiently great to be of importance; hence in all cases it is necessary to indicate clearly whether lower or higher calorific values have been used.

As 1 c. ft. of hydrogen requires 0·5 c. ft. of oxygen for just complete combustion, it follows immediately that 1 c. ft. of hydrogen needs $0{\cdot}5 \times 4{\cdot}762 = 2{\cdot}381$ c. ft. of air for just complete combustion. The volume of the hydrogen-air mixture is accordingly $1 + 2{\cdot}381 = 3{\cdot}381$ c. ft.; and the "higher" and "lower" calorific values per cubic foot of mixture with air (at 32° F. and 1 atmosphere) are thus $\frac{347}{3{\cdot}381} = 102{\cdot}5$ B.Th.U. and $\frac{292}{3{\cdot}381} = 86{\cdot}4$ B.Th.U. respectively.

The 3·381 c. ft. of mixture contains $2{\cdot}381 \times 0{\cdot}79 = 1{\cdot}881$ c. ft. of nitrogen—which is unchanged by the combustion; the 1·5 c. ft. of mixed hydrogen and oxygen shrink after combustion to $\frac{2}{3} \times 1{\cdot}5 = 1{\cdot}0$ c. ft. of gaseous water (at 32° F. and 1 atmosphere). The products of combustion consist therefore of

| | | |
|---|---|---|
| Nitrogen .. | 1·881 c. ft. .... | 65·3 per cent |
| Gaseous water | 1·000 " .... | 34·7 " |
| Total | 2·881 c. ft. | 100·0 |

and the ratio of the final volume to the initial volume is $\frac{2{\cdot}881}{3{\cdot}381} = 0{\cdot}853$.

**Carbon.**—Carbon forms two oxides, namely carbon monoxide, CO, and carbon dioxide, $CO_2$. With hydrogen, carbon forms innumerable compounds, five only of which it is necessary to mention here, viz. methane, ethylene, ethane, acetylene, and benzene.

**Carbon Monoxide, CO.**—This is an invisible, odourless, highly poisonous, inflammable gas fourteen times as heavy as hydrogen. 12 lb. of carbon combined with 16 lb. of oxygen give 28 lb. of carbon monoxide. Thus 1 lb. of carbon produces $2\frac{1}{3}$ lb. of carbon monoxide, and for this $1\frac{1}{3}$ lb. of oxygen or $1\frac{1}{3} \times 4{\cdot}348 = 5{\cdot}797$ lb. of air are needed, corresponding to $5{\cdot}797 \times 12{\cdot}387 = 71{\cdot}8$ c. ft. of air at 32° F. and 1 atmosphere pressure.

The 6·797 lb. of mixture consists, therefore, after combustion, of

$5{\cdot}797 \times 0{\cdot}77 = 4{\cdot}463$ lb. of nitrogen and $2{\cdot}334$ lb. of CO. This occupies a volume of $\frac{4{\cdot}463 + 2{\cdot}334}{0{\cdot}0783} = 86{\cdot}8$ c. ft., as nitrogen has the same density as CO (see Table IV). In burning 1 lb. of carbon to carbon monoxide it is found that, roundly, 4326 B.Th.U. of heat are evolved.

**Combustion of Carbon Monoxide.**—Burnt in air, 28 lb. of CO combine with 16 lb. of oxygen to form 44 lb. of carbon dioxide; thus 1 lb. of CO requires $\frac{16}{28} \times 4{\cdot}348 = 2{\cdot}4846$ lb. of air, and produces $\frac{44}{28} = 1{\cdot}5714$ lb. of $CO_2$, the remaining $1{\cdot}9132$ lb. being nitrogen.

By volume, 2 c. ft. of CO unite with 1 c. ft. of oxygen to form 2 c. ft. of $CO_2$; thus a mixture of CO and O shrinks after combustion to two-thirds of its original volume; and 1 c. ft. of CO produces 1 c. ft. of $CO_2$.

In burning 1 lb. of carbon monoxide to $CO_2$ it is found that, roundly, 4360 B.Th.U. of heat are evolved.

Also as 1 c. ft. of CO at 32° F. and 1 atmosphere weighs $14 \times 0{\cdot}005592 = 0{\cdot}0783$ lb., it follows that in burning 1 c. ft. of CO to $CO_2$, $4360 \times 0{\cdot}0783 = 341$ B.Th.U. of heat are evolved. Or, again, as 1 c. ft. of CO requires for just complete combustion $\frac{1}{2} \times 4{\cdot}762 = 2{\cdot}381$ c. ft. of air, the heat evolved in burning 1 c. ft. of CO-air mixture is $\frac{341}{3{\cdot}381} = 100{\cdot}9$ B.Th.U. There is here no question of "higher" or "lower" calorific value, as no water is produced by the combustion.

**Carbon Dioxide, $CO_2$,** also termed carbonic-acid gas (the "choke-damp" of miners), is an invisible, odourless, inert, non-inflammable, slightly poisonous gas 22 times as heavy as hydrogen; thus 1 c. ft. at 32° F. and 1 atmosphere weighs $22 \times 0{\cdot}005592 = 0{\cdot}123$ lb. 1 lb. of carbon burnt to $CO_2$ requires $\frac{32}{12} = 2\frac{2}{3}$ lb. of oxygen, or $11{\cdot}595$ lb. of air; the $12{\cdot}595$ lb. of resulting products consist of $3{\cdot}667$ lb. of $CO_2$ and $8{\cdot}928$ lb. of nitrogen. In burning 1 lb. of carbon to $CO_2$ it is found that, roundly, 14,500 B.Th.U. of heat are evolved. If the combustion be regarded as taking place in two stages, viz. (1) burning the 1 lb. of carbon to $2\frac{1}{3}$ lb. of carbon monoxide, and (2) burning the $2\frac{1}{3}$ lb. of carbon monoxide to $2\frac{1}{3} \times \frac{44}{28} = 3{\cdot}667$ lb. of $CO_2$, then,

| Stage. | B.Th.U. Evolved. |
|---|---|
| 1. 1 lb. C burnt to $2\frac{1}{3}$ lb. CO .. .. .. .. | 4,326 |
| 2. $2\frac{1}{3}$ lb. CO burnt to $3\frac{2}{3}$ lb. $CO_2 = 2\frac{1}{3} \times 4360$ .. | 10,173 |
| Total .. .. .. | 14,499 |

substantially as stated above.

At 32° F. and 1 atmosphere the $3\frac{2}{3}$ lb. of $CO_2$ resulting from the combustion of 1 lb. of carbon occupy $\frac{3{\cdot}667}{0{\cdot}123} = 29{\cdot}8$ c. ft. Hence in the formation of 1 c. ft. of $CO_2$ from free carbon, $\frac{14500}{29{\cdot}8} = 486{\cdot}6$ B.Th.U. of heat were evolved.

## TABLE IV

### Heats of Combustion, &c., of the Usual Fuel Gases

All Volumes measured at 32° F. and 1 Atmosphere Pressure

| Gas. | | Weight and Volume. | | | Burnt to | Heat of Combustion of 1 lb. of Gas. | | Heat of Combustion of 1 C. Ft. of Gas. | | C. Ft. of Air to burn 1 C. Ft. of Gas. | Volume of Gas-air Mixture in C. Ft. | Heat of Combustion of 1 C. Ft. of Gas-air Mixture. | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Name. | Symbol. | Weight with H = 1. | Weight of 1 C. Ft. in Pounds. | Volume of 1 lb. in C. Ft. | | Higher Value. | Lower Value. | Higher Value. | Lower Value. | | | Higher Value. | Lower Value. |
| Hydrogen .. | H | 1 | 0·005592 | 178·83 | $H_2O$ | 62,000 | 52,190 | 347 | 292 | 2·381 | 3·381 | 102·5 | 86·4 |
| Carbon monoxide | CO | 14 | 0·07830 | 12·77 | $CO_2$ | 4,360 | — | 341 | — | 2·381 | 3·381 | 100·9 | — |
| Methane .. | $CH_4$ | 8 | 0·04474 | 22·35 | $H_2O$ & $CO_2$ | 23,540 | 21,085 | 1053 | 943 | 9·524 | 10·524 | 100·0 | 89·6 |
| Ethylene .. | $C_2H_4$ | 14 | 0·07830 | 12·77 | ,, | 21,420 | 20,017 | 1677 | 1567 | 14·286 | 15·286 | 109·7 | 102·5 |
| Ethane .. | $C_2H_6$ | 15 | 0·08388 | 11·92 | ,, | 22,330 | 20,366 | 1873 | 1708 | 16·67 | 17·67 | 106·0 | 90·0 |
| Acetylene .. | $C_2H_2$ | 13 | 0·07270 | 13·76 | ,, | 21,850 | 21,095 | 1588 | 1534 | 11·90 | 12·90 | 123·1 | 119·0 |
| Benzene .. | $C_6H_6$ | 39 | 0·21810 | 4·59 | ,, | 18,000 (average) | 17,245 | 3926 | 3761 | 35·70 | 36·70 | 107·0 | 102·4 |

**Methane,** also termed "marsh gas" and "light carburetted-hydrogen" (the "fire-damp" of miners), is a colourless, odourless, inflammable, stable gas having the formula $CH_4$, and is accordingly $\frac{12 + 4}{2} = 8$ times as heavy as hydrogen. It is a very valuable gaseous fuel of high calorific value, and is the main ingredient of the "natural gas" so largely used in the United States and elsewhere for heating and power purposes.

Burnt in air it produces carbon dioxide and water, agreeably with the formula:

$$CH_4 + 4O = CO_2 + 2H_2O \ldots\ldots\ldots\ldots\ldots (4)$$
$$16 \text{ lb.} + 64 \text{ lb.} = 44 \text{ lb.} + 36 \text{ lb.}$$

Regarded as free carbon and free hydrogen, its higher calorific value per pound would be $\frac{12 \times 14{,}500 + 4 \times 62{,}000}{16} = 26{,}375$ B.Th.U., but energy is required to dissociate the carbon and hydrogen, and actual calorimetric measurements show the higher heat value to be only, roundly, 23,540 B.Th.U. per pound. In general, the heat value of gases by calorimetric determination is somewhat less than as calculated from their carbon and hydrogen content; but there are important exceptions, as, e.g., acetylene, which, from the formula $C_2H_2$, gives a calculated higher value of $\frac{1}{26}\{24 \times 14{,}500 + 2 \times 62{,}000\} = 18{,}154$ B.Th.U. per pound, whereas calorimetric experiments show its higher heat value to be about 20 per cent greater, viz., roundly, 21,850 B.Th.U. per pound.

Useful data relating to the combustion of the above gases, and including also the heavier hydrocarbons, ethylene, ethane, and benzene, are collected together for reference purposes in Table IV.

In Table V data are given for the same seven gases relating to the volumes of oxygen and of air necessary to just completely burn 1 c. ft. It will be noted that in general the volume of the products of combustion differs from that of the gas-air mixture before combustion, being sometimes less and sometimes greater, while with methane and ethylene there is no change.

**Town's Gas, or "Coal Gas".**—Obtained by the destructive distillation of coal in closed retorts, coal gas was for many years manufactured mainly as an illuminant, and for this purpose cannel coal was largely employed in order to obtain the necessary heavy hydrocarbon gases. The introduction of the incandescent mantle, and the great increase in the use of gas for domestic heating and cooking and for power purposes within the past forty years, combined to render its illuminating power of secondary importance. Accordingly in 1920 the Gas Regulation Act was passed empowering companies to supply gas to consumers on the basis of its heating power alone, the unit adopted, termed the "therm", being 100,000 B.Th.U. of heat.

Thus if H denote the heat value of the gas supplied in B.Th.U. per

## TABLE V

VOLUME OF AIR REQUIRED TO JUST BURN 1 C. FT. OF GAS

All Measurements made at 32° F. and 1 Atmosphere Pressure

| Gas. | C. Ft of Air per C. Ft. of Gas. | Volume of Mixture with Air before Combustion, C. Ft. | | | | Volume of Mixture after Combustion, C Ft | | | | Ratio of Final to Initial Volume. |
|---|---|---|---|---|---|---|---|---|---|---|
| | | Of Gas. | Of Atmospheric Oxygen. | Of Atmospheric Nitrogen. | Total Volume in C. Ft | Of Water as a Gas. | Of Carbon Dioxide | Of Nitrogen. | Total Final Volume in C. Ft. | |
| Hydrogen .. | 2·381 | 1·0 | 0·500 | 1·881 | 3·381 | 1·0 | 0·0 | 1·881 | 2·881 | 0·853 |
| Carbon monoxide | 2·381 | 1·0 | 0·500 | 1·881 | 3·381 | 0·0 | 1·0 | 1·881 | 2·881 | 0·853 |
| Methane .. | 9·524 | 1·0 | 2·00 | 7·524 | 10·524 | 2·0 | 1·0 | 7·524 | 10·524 | 1·000 |
| Ethylene .. | 14·286 | 1·0 | 3·00 | 11·286 | 15·286 | 2·0 | 2·0 | 11·286 | 15·286 | 1·000 |
| Ethane .. .. | 16·67 | 1·0 | 3·50 | 13·17 | 17·67 | 3·0 | 2·0 | 13·17 | 18·17 | 1·028 |
| Acetylene .. | 11·90 | 1·0 | 2·50 | 9·40 | 12·90 | 1·0 | 2·0 | 9·40 | 12·40 | 0·960 |
| Benzene .. .. | 35·7 | 1·0 | 7·50 | 28·20 | 36·70 | 3·0 | 6·0 | 28·20 | 37·20 | 1·014 |

cubic foot, if V be the volume supplied in cubic feet, and if T denote the corresponding number of therms, then

$$T = \frac{HV}{100{,}000} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (5)$$

Gas companies are now permitted a considerable latitude in respect of the quality of the gas they choose to manufacture, but they must declare the calorific value of the gas supplied to consumers, and the declared value is permitted but a small amount of variation. Moreover, the quality being now at the choice of the company, a stimulus has been given to the process termed the *complete gasification* of coal, by which is meant such treatment as leaves finally no other solid residue than incombustible ash.

Complete gasification is effected either in *single-stage* or *two-stage* plants, in each of which there is produced (1) normal or "straight" coal gas, and (2) "blue" water-gas (p. 69).

It is also a frequent practice with companies to reduce production costs by adding to their "straight" coal gas a proportion of water-gas, either untreated (blue) or more commonly enriched or "carburetted" by admixture of heavy hydrocarbon vapours obtained from naphtha, schist oil, tar, &c. A difficulty in the use of water-gas is, however, its large content of the poisonous carbon monoxide; for public supply the proportion of CO should, for safety, never exceed about 16 per cent of the whole volume of the gas. Some undertakings, as, e.g., the large South Metropolitan Gas Company of London, do not use water-gas at all, but supply merely a "straight" coal gas; the practice of this important company may be briefly described.*

Durham gas coal, of an average heat value of 13,500 B.Th.U. per pound, after being crushed so as to pass through a 2-in.-square mesh, is mechanically charged into horizontal retorts made of refractory fireclay. Each retort is 20 ft. in length, and of oval cross-section 15 in. × 21 in., and to each end is attached a tightly fitting iron door and an iron pipe through which the gas is removed. Ten such retorts are grouped in one setting, and heated by producer gas generated from part of the coke produced. The furnace surrounding the retorts is maintained at a temperature of about 2400° F., the charge within the retort being thereby heated to about 1920° F. A charge of 10½ cwt. of coal is placed in each retort, and this is then heated or "carbonized" for a period of ten hours. The volatile products are withdrawn from the retorts by gas exhausters, by which they are delivered to the purification plant comprising (1) water-cooled condensers for the extraction of tar; (2) washers and "scrubbers" for the recovery of ammoniacal liquor; (3) boxes containing prepared iron oxide for removing hydrogen sulphide; (4) apparatus containing nickel catalyst for converting the bisulphide of carbon to hydrogen sulphide; (5) secondary iron oxide boxes for removing the last traces of hydrogen sulphide; and (6) washers con-

* By courtesy of Dr. C. Carpenter, M.Inst.C.E.

taining anthracene oil for the extraction of naphthalene. The gas, thus purified, is passed finally through meters into the gas-holders, and is ready for distribution.

Analysis shows this "straight" coal gas to have the following composition:

| Item. | Percentage by Volume. | |
|---|---|---|
| Hydrogen | 51·1 | Combustible. |
| Carbon monoxide | 8·0 | Combustible. |
| Methane | 30·0 | Combustible. |
| Unsaturated hydrocarbons | 3·0 | Combustible. |
| Oxygen | 0·2 | |
| Nitrogen | 5·9 | |
| Carbon dioxide | 1·8 | |
| | 100·0 | |

together with sulphur compounds amounting to about 6 gr. in 100 c. ft.

About 12,680 c. ft. of this gas are obtained per ton of coal, together with the following valuable by-products: (1) about 14 cwt. of coke, $3\frac{1}{2}$ cwt. of which are utilized in making the producer gas for heating the retorts; (2) 10·5 gall. of tar; (3) 40 gall. of ammoniacal liquor containing about 2 per cent of ammonia; and (4) about 11 lb. of sulphur, which is subsequently converted to sulphuric acid.

**Calorific Value.**—Estimated by aid of Table IV (p. 63), 1 c. ft. of this gas at 32° F. and 1 atmosphere pressure should have a heat value of

$$0{\cdot}511 \times 347 + 0{\cdot}08 \times 341 + 0{\cdot}3 \times 1053 + 0{\cdot}03 \times 2000^* = 580$$ B.Th.U., higher, or

$$0{\cdot}511 \times 292 + 0{\cdot}08 \times 341 + 0{\cdot}3 \times 943 + 0{\cdot}03 \times 1850^* = 515$$ B.Th.U., lower;

while from Table V (p. 65),

$$0{\cdot}511 \times 0{\cdot}5 + 0{\cdot}08 \times 0{\cdot}5 + 0{\cdot}3 \times 2{\cdot}0 + 0{\cdot}03 \times 6^* = 1{\cdot}076 \text{ c. ft.}$$

of oxygen are necessary for the just complete combustion of 1 c. ft. of the gas. As the gas itself contains 0·002 c. ft. of free oxygen per cubic foot, $1{\cdot}076 - 0{\cdot}002 = 1{\cdot}074$ c. ft. of atmospheric oxygen, or $1{\cdot}074 \times 4{\cdot}762 = 5{\cdot}1$ c. ft. of air are needed.

Finally, by aid of Table IV, the weight of 1 c. ft. of the gas at 32° F. and 1 atmosphere pressure is

$$0{\cdot}511 \times 0{\cdot}005592 + 0{\cdot}08 \times 0{\cdot}0783 + 0{\cdot}3 \times 0{\cdot}04474 + 0{\cdot}03 \times 0{\cdot}15^* + 0{\cdot}002 \times 0{\cdot}08947 + 0{\cdot}059 \times 0{\cdot}0783 + 0{\cdot}018 \times 0{\cdot}123 = 0{\cdot}034 \text{ lb.};$$ thus its density, relative to air, is $$\frac{0{\cdot}034}{0{\cdot}08073} = 0{\cdot}42.$$

* These are usual assumed figures.

The declared normal calorific values of the gas supplied by three of the largest London companies during 1929 were:

| | |
|---|---|
| The South Metropolitan Gas Company .. .. .. | 560 |
| The Gas, Light, & Coke Company .. .. .. | 500 |
| The Commercial Gas Company .. .. .. .. | 500 |

These figures are higher values as determined by calorimeter, and relate to 1 c. ft. of gas saturated with water vapour, at 60° F. and 1 atmosphere (14·7 lb. per square inch) pressure.

The corresponding heat value per cubic foot of dry gas at 32° F. and 1 atmosphere may be obtained as follows: from Table I, 1 c. ft. of moist gas at 60° F. and 1 atmosphere consists of 1 c. ft. of dry gas at 60° F. and $(14{\cdot}7 - 0{\cdot}253) = 14{\cdot}447$ lb. per square inch, together with $\frac{1}{1210}$ lb. of water vapour, and, taking, for example, the first case above, it is found that this evolves 560 B.Th.U. of available heat on burning.

Now let H denote the calorific value of 1 c. ft. of dry gas at 32° F. and 1 atmosphere. Then at 60° F. and 1 atmosphere, 1 c. ft. of dry gas would evolve $\frac{492}{520}$H thermal units. If the pressure of 1 c. ft. at 60° F. be but 14·447 lb. per square inch, the heat evolved will be reduced to $\frac{14{\cdot}447}{14{\cdot}70} \times \frac{492}{520} \times$ H. Finally, if the 1 c. ft. be of *moist* gas, by Table II $\frac{1071}{1210}$ B.Th.U. of heat are evolved in condensing the contained water vapour from and at 60° F. Accordingly:

$$\frac{14{\cdot}447}{14{\cdot}70} \times \frac{492}{520} \times \mathrm{H} + \frac{1071}{1210} = 560,$$

whence H = 600 B.Th.U. at 32° F. and 1 atmosphere pressure for 1 c. ft. of dry gas. The difference between this result and that calculated above, viz. 580 B.Th.U., arises from the unknown constitution of the 3 per cent of "unsaturated hydrocarbons". A value of 2000 B.Th.U. per cubic foot is very commonly assumed for these, but, as reference to Table IV will show, their calorific value may easily substantially exceed this figure. In the present instance the calculated and observed values agree by taking 2700 B.Th.U. as the calorific value of the unsaturated hydrocarbons in the gas.

An idea of the magnitude of the coal-gas industry may be gained from reflecting that in 1922 the South Metropolitan Gas Company alone carbonized 1,225,042 tons of coal, while the whole industry in Great Britain carbonized about 20,000,000 tons in the year. The average heat value of the coal was, roundly, 13,000 B.Th.U. per pound, and the gas produced was consumed in about the following proportions: heating, 55 per cent; lighting, 35 per cent; power, 10 per cent. Approximately 25 per cent of the heat of the coal retorted is recovered in burning the gas made, but there are, in addition to the coke, very valuable by-products formed, some of which are inflammable. In 1938 the Gas Light and Coke Company, Ltd., in one day distributed 264,691,000 c. ft. of gas, involving the carbonization of 13,860 tons of coal and 144,019 gall. of oil for enriching water gas.

**Water-gas.**—When steam is passed through incandescent carbon (coke) maintained at a temperature of about 2100° F., decomposition occurs as indicated by the formula:

$$C + H_2O = CO + H_2. \quad \ldots\ldots\ldots\ldots\ldots\ldots(6)$$

Accordingly 12 lb. of carbon and 18 lb. of steam yield 28 lb. of CO and 2 lb. of H; or 1 c. ft. of carbon (gaseous) and 2 c. ft. of steam yield 2 c. ft. of CO and 2 c. ft. of H; thus the product consists of equal volumes of CO and H.

If the carbon, or coke, be maintained at the lower temperature of about 1200° F., the chemical reaction is expressed by:

$$C + 2H_2O = CO_2 + 2H_2, \quad \ldots\ldots\ldots\ldots\ldots(7)$$

$$12 \text{ lb.} + 36 \text{ lb.} = 44 \text{ lb.} + 4 \text{ lb.},$$

and the only inflammable constituent of the product is hydrogen.

In the first case, from 1 lb. of carbon having a heat value of 14,500 B.Th.U. $\frac{28}{12}$ lb. of CO and $\frac{2}{12}$ lb. of H are obtained. These have, jointly, a (lower) heat value of $\frac{28 \times 4360}{12} + \frac{2 \times 52{,}190}{12} = 18{,}870$ B.Th.U.

In the second case, 1 lb. of carbon yields $\frac{4}{12}$ lb. of H with a (lower) heat value of $\frac{4 \times 52{,}190}{12} = 17{,}397$ B.Th.U.

In both cases, therefore, the gaseous products contain more heat than the carbon burnt; both reactions are thus " endothermic ", and the heat absorbed is derived from the incandescent coke, which accordingly suffers a rapid fall of temperature.

The following table (from Bunte) shows how the nature of the gaseous products varies with the temperature of the incandescent coke.

TABLE VI

WATER-GAS: PRODUCTS AT DIFFERING TEMPERATURES

| Mean Temperature of Coke, Degrees Fahr. | Composition of Gas Produced: Percentage by Volume. | | |
|---|---|---|---|
| | Of Hydrogen. | Of Carbon Monoxide. | Of Carbon Dioxide. |
| 1265 | 65·2 | 4·9 | 29·8 |
| 1582 | 59·9 | 18·1 | 21·9 |
| 2057 | 50·9 | 48·5 | 0·6 |

As 1 c. ft. of true water-gas contains 0·5 c. ft. each of H and CO, its (lower) calorific value per cubic foot at 32° F. and 1 atmosphere should be, by Table IV, $\frac{292 + 341}{2} = 316{\cdot}5$ B.Th.U. As actually made with coke, anthracite, or charcoal as fuel its heat value is somewhat lower, and ordinarily ranges from about 270 to 300 B.Th.U.

In the manufacture of "straight" coal gas it has been seen that all the volatile matter in the coal is driven off in the retorts. The residual coke being used to produce water-gas, and this gas—either pure (blue) or more usually enriched or *carburetted* by the addition of some heavy hydrocarbon "oil" vapour—being then added to the straight coal gas, it is evident that complete gasification of the coal is attained. In making water-gas, hot air is first blown through the ignited fuel to raise it to incandescence; a temperature of about 2000° F. being attained, the air blast is cut off and steam is next passed through; this intermittent action is termed the *blow and run* process. As the run proceeds the temperature of the fuel falls, and the water-gas produced undergoes progressive deterioration. Usually the duration of a blow is three to four minutes followed by a run of five to six minutes, but in certain modern plants producing "blue gas" the blowing period is reduced to from about one to one and a half minutes only. During 1913 the gas companies in Great Britain manufactured upwards of 15,000,000,000 c. ft. of water-gas, the enriching *oil* used amounting to about 37,000,000 gall.

In Table VII data are given relating to the average composition by volume of London "town's" gas from 1884 to 1922, together with figures for "blue" and "carburetted" water-gas for comparison.

TABLE VII

TOWN GAS (LONDON) AND WATER-GAS

Averages (Per Cent)

| Constituent. | 1884. | 1909. | | 1922. | | | Water-gas. | |
|---|---|---|---|---|---|---|---|---|
| | Straight Coal Gas. | Straight Coal Gas. | Mixed Gas. | Straight Coal Gas.† | Mixed Gas.‡ | Complete Gasification. | "Blue." | "Carburetted." |
| H .. .. | 48·0 | 49·2 | 46·75 | 51·1 | 45·05 | 51·0 | 48·6 | 29·35 |
| CO .. .. | 3·75 | 8·2 | 14·50 | 8·0 | 13·70 | 30·0 | 44·0 | 33·19 |
| $CH_4$ .. .. | 36·0 | 31·2 | 24·6 | 30·0 | 24·05 | 8·0 | 0·4 | 20·48 |
| Heavy hydrocarbons .. | 4·5 | 3·4 | 4·9 | 3·0 | 2·85 | 1·4 | — | 11·32 |
| O .. .. | 0·25 | 0·25 | 0·75 | 0·2 | 1·02 | — | — | — |
| N .. .. | 6·0 | 6·25 | 7·1 | 5·9 | 10·43 | 5·0 | 3·7 | 5·66 |
| $CO_2$ .. .. | 1·5 | 1·5 | 1·4 | 1·8 | 2·90 | 4·6 | 3·3 | — |
| Calorific value * | 649 | 595 | 569 | 580 | 513 | 392 | 323 | 657 |

* Higher; in B.Th.U. per cubic foot at 32° F. and 1 atmosphere.
† The South Metropolitan Gas Company, 1922. ‡ The Gas, Light, and Coke Company, 1922.

**Coke-oven Gas.**—As above stated, an immense quantity of coal is carbonized annually to produce town's gas. Nearly as great an amount is also specially carbonized to obtain the coke required in various metallurgical operations.

In coke ovens, coal is very slowly heated up to a temperature of about 1850° F., nearly all the volatile matter being driven off, leaving a hard coke capable of withstanding the heavy crushing load sustained when used in blast-furnaces, &c.

Most metallurgical coke is now made in by-product ovens. Although the principal end is coke-production, the valuable by-products ammonia, tar, and benzole are also recovered, while enormous quantities of "coke-oven gas" are evolved. The principal heat-waste with coke ovens usually occurs in the sensible heat of the coke itself when discharged, and about 0·4 lb. of steam may be raised from each pound of the hot coke, though the additional plant involved is costly, and so but little adopted.

During 1927, 11,837,000 tons of coke were made, each ton of coal yielding 12 to 16 cwt. of coke, 20 to 26 lb. (and sometimes even 36 lb.) of ammonia sulphate, 2 to 3½ gallons of 65 per cent crude benzole, 85 to 110 lb. of tar, and 10,000 to 13,000 c. ft. of coke-oven gas. During the year about 5340 millions of cubic feet of this gas were sold to gas-works for domestic supply, and it is also sometimes used, mixed with producer or blast-furnace gas, in steel melting and reheating furnaces; but its largest application is in steam-raising for generation of electricity. Being really a waste product, it is subject to rapid and considerable fluctuations in its composition, though it is usually characterized by a large hydrogen content, while nitrogen is often present in considerable quantity.

In Table VIII the results of analysis of four coke-oven gases are given; these well illustrate the considerable variations in its composition.

TABLE VIII

ANALYSES AND HEAT VALUES OF SOME COKE-OVEN GASES

| Constituent. | From British Plants: Percentage by Volume. | | | A German Plant. |
|---|---|---|---|---|
| Hydrogen | 63·42 | 27·46 | 47·07 | 55·0 |
| Carbon monoxide | 5·21 | 5·42 | 5·66 | 7·0 |
| Methane | 23·14 | 22·85 | 26·06 | 26·0 |
| Heavy hydrocarbons | 0·80 | 1·35 | 1·60 | 1·5 |
| Oxygen | 0·42 | 3·31 | 0·91 | 0·0 |
| Nitrogen | 5·00 | 35·51 | 15·82 | 10·5 |
| Carbon dioxide | 2·01 | 4·10 | 2·88 | |
| *Calorific Value*: * | | | | |
| Higher | 497 | 420 | 489 | 490 |
| Lower | 436 | 374 | 432 | 457 |
| Air required in c. ft. per c. ft. of gas | 4·04 | 3·19 | 4·15 | 4·38 |

* At 32° F. and 1 atmosphere.

These figures may usefully be compared with those of Table VII; the calorific values and air volumes are calculated from Tables IV and V respectively.

Calorific values are frequently expressed at 60° F. and 1 atmosphere pressure. Such values are readily derived from those at 32° F. by multiplying by $\frac{460 + 32}{460 + 60} = 0{\cdot}946$.

**The "Gas Producer".**—As early as 1839 Bischof obtained an inflammable gaseous mixture for heating purpose by burning peat in a brick chamber, with a limited supply of air at atmospheric pressure admitted

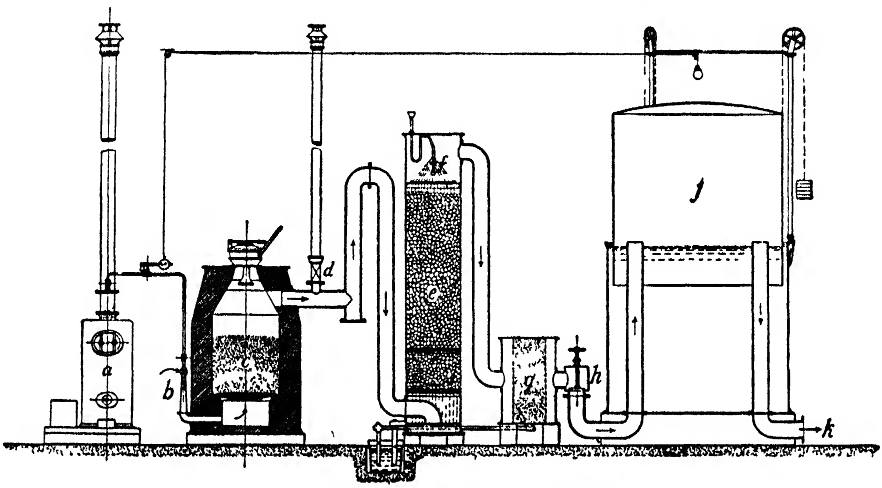


Fig. 1.—Dowson Gas Plant. Steam-jet Pressure Type

through holes in the ash-pit cover. About 1841 Messrs. Thomas and Laurent obtained inflammable gas by blowing air into an enclosed furnace, the reactions being assisted by a jet of superheated steam, thus anticipating the modern "pressure producer". Messrs. Kirkham in 1852 first adopted the "intermittent" procedure in gas-making, substantially as described in p. 70. Among other pioneers must be mentioned Sir W. Siemens, who, from 1861 onwards, devoted much attention to the production of gaseous mixtures capable of use instead of solid fuel in furnaces, and his researches first brought the subject of gaseous fuel into prominence. Modern Siemens generators continue extensively in use for the supply of fuel gas to annealing ovens, crucible furnaces, and normalizing, case-hardening, and tempering apparatus, &c.

To Mr. J. E. Dowson, however, especial credit is due as the first to construct a complete "pressure" gas producer capable of supplying a fuel gas suitable for general heating purposes, and particularly for power production. The first Dowson pressure producer was exhibited at York in 1881, supplying the gaseous fuel operating a 3-h.p. Otto gas-engine. As illustrated diagrammatically in fig. 1, it included a small separately fired steam boiler *a*, supplying superheated steam, mixed with air, through an injector *b* into the ash-pit of a closed brick-lined producer *c*, partly filled with incandescent anthracite. The mixed steam and air passed through the fire-bars and upwards through the ignited fuel, the products of combustion being conveyed through a pipe as indicated into the bottom of a tall cylindrical iron vessel loosely filled with coke, *e*, on which a jet of water was sprayed. This is termed the *coke scrubber*, and its function is to cool and partly clean the hot smoky gases issuing from the producer. Passing upwards through the coke scrubber the gas was led next into a vessel *g*, containing loose coarse sawdust, termed the *sawdust scrubber*, whereby any remaining tarry vapours were abstracted. The cleaned and cooled gas was passed finally through a shut-off valve *h* into a gas-holder *j*, whence it was delivered through a gas outlet *k* to the heating apparatus, or engine, in which it was burnt.

This plant is termed a *pressure* producer, as the internal pressure throughout is slightly above that of the atmosphere.

Starting was effected by first charging the producer with fuel through the feed hopper indicated at the top, closing the valve *h*, and opening the waste-cock *d*. The fire was then lighted through a fire-door, and quickly raised to incandescence by a hand-driven blower, the first products of combustion passing directly into the air through the vertical waste-pipe shown. Simultaneously steam was raised in the boiler *a*, and the injector *b* being next opened, a combined jet of steam and air was blown into the ash-pit, and the gas-making process thus commenced. The waste-cock *d* remained open until normal gas was produced, when it was closed, and valve *h* opened, whereupon the gas was delivered into the holder *j*. The tar- and dust-laden water from the bottom of the coke scrubber runs away through a simple water seal. The coke becomes gradually charged with impurities and requires occasional renewal; the lower portion becomes foul sooner than the upper, and provision is accordingly made, as indicated, to renew this separately. The sawdust similarly needs renewal periodically.

Prior to the appearance of this apparatus, gas-engines were practically restricted to the use of town's gas, which, although an excellent fuel, is very expensive. Dowson's pressure producer, and, in particular, the suction producer evolved somewhat later (p. 94), gave a great stimulus to the construction of gas-engines of increased power, as they furnished the engineer with a practically unlimited supply of a satisfactory power gas at a very low cost.

Pressure plants of very large size continue to be made, especially for use with bituminous coal in accordance with the inventions of Dr. Mond. In the case of the smaller plants the suction producer prevails, while recently a modification termed the suction-pressure system has appeared, as referred to later (p. 100).

**Action in a Gas Producer.**—Consider firstly that air only is passed through a mass of incandescent carbon. The fuel near the entering air is completely burned to carbon dioxide, each 1 lb. of carbon uniting with

$2\frac{2}{3}$ lb. of atmospheric oxygen to form $3\frac{2}{3}$ lb. of $CO_2$, 14,500 B.Th.U. of heat being evolved in forming the combination. The resulting $3\frac{2}{3}$ lb. of $CO_2$ in passing through a further mass of incandescent fuel is next reduced to carbon monoxide thus:

$$CO_2 + C = 2CO \quad \text{.................... (8)}$$

$$44 \text{ lb.} + 12 \text{ lb.} = 56 \text{ lb.}$$
$$\therefore\ 3\tfrac{2}{3} \text{ lb.} + 1 \text{ lb.} = 4\tfrac{2}{3} \text{ lb.}$$

So that each $3\frac{2}{3}$ lb. of $CO_2$ takes up a further 1 lb. of C and produces $4\frac{2}{3}$ lb. of CO. The net result therefore in this case is that, in theory, all the carbon is converted into carbon monoxide, which is subsequently used in heating apparatus, or in a gas-engine cylinder, and burnt to carbon dioxide.

The 2 lb. of carbon if burned directly to $CO_2$ would evolve $2 \times 14{,}500 = 29{,}000$ B.Th.U., whereas the resulting $4\frac{2}{3}$ lb. of CO which issue from the producer yield only $4\frac{2}{3} \times 4360 = 20{,}347$ B.Th.U. in burning to $CO_2$. Hence in the ideal case the efficiency of such a producer would be only $\frac{20{,}347}{29{,}000} = 0{\cdot}7$ as a maximum; the difference—viz. $(29{,}000 - 20{,}347) = 8653$ B.Th.U. per 2 lb., i.e. $\frac{8653}{2} = 4326$ B.Th.U. per 1 lb. of carbon gasified —must have appeared as free heat in the producer, and have been carried off in the issuing hot gases. Briefly, therefore, it may be said that the net result is that the fuel in the producer is burnt to carbon monoxide, evolving 4326 B.Th.U. per pound of carbon consumed (p. 238).

The issuing gases would have the composition by weight:

| | Pounds. | Percentage by Weight. |
|---|---|---|
| Carbon monoxide .. .. | $4\frac{2}{3}$ .... | 34·3 |
| Atmospheric nitrogen .. .. | $2\frac{2}{3} \times \frac{77}{23}$ .... | 65·7 |

and these figures express also the composition by volume, as CO and N are of equal density.

The earliest, and unsuccessful, attempts to construct satisfactory gas producers contemplated the use of air alone, as just considered, but the 4326 B.Th.U. of heat evolved per pound of carbon gasified—30 per cent of its whole heat—resulted in severe overheating of the producer and the rapid destruction of its refractory lining, together with a very hot issuing gas. All these difficulties have been successfully surmounted, and the efficiency largely increased by passing not air alone, but a *mixture of air and steam* through the incandescent fuel; by this means gas producers are rendered entirely practicable and capable of furnishing a continuous output of inflammable gaseous mixture varying but little in composition, and at a very low cost of production.

An ideal producer of this kind may be regarded as (1) directly producing CO as in the previous case, and (2) utilizing the 4326 B.Th.U. of free heat per pound of carbon to decompose the steam in presence of incandescent fuel. For this second process we have from, equation (6):

$$(1 \text{ lb. water} + \tfrac{2}{3} \text{ lb. carbon}) \text{ yields } (\tfrac{1}{9} \text{ lb. H} + \tfrac{14}{9} \text{ lb. CO}).$$

Now 1 lb. of water decomposed to free hydrogen and oxygen absorbs $\frac{62,000}{9} = 6889$ B.Th.U.; while $\frac{2}{3}$ lb. of carbon burned to CO evolves $\frac{2}{3} \times 4326 = 2883$ B.Th.U. (p. 220). Accordingly 1 lb. water + $\frac{2}{3}$ lb. carbon on the whole *absorb* $6889 - 2883 = 4006$ B.Th.U. in their conversion to $\frac{1}{9}$ lb. of hydrogen and $\frac{14}{9}$ lb. of carbon monoxide.

The amount of heat available is, however, 4326 B.Th.U. per pound of fuel, and this amount will decompose $\frac{4326}{4006} = 1{\cdot}08$ lb. of water, which will unite with $\frac{2}{3} \times 1{\cdot}08 = 0{\cdot}72$ lb. of carbon and yield 0·12 lb. of free hydrogen and $\frac{14}{9} \times 1{\cdot}08 = 1{\cdot}68$ lb. of carbon monoxide.

The resulting gaseous mixture per $(1 + 0{\cdot}72) = 1{\cdot}72$ lb. of carbon burned, and 1·08 lb. of water decomposed, and 5·797 lb. of air supplied, would consist of:

| | | Pounds. | |
|---|---|---|---|
| Carbon monoxide | .. | 2·333 | from direct combustion of carbon. |
| Atmospheric nitrogen | .. | 4·464 | |
| Carbon monoxide | .. | 1·680 | from decomposition of the water. |
| Free hydrogen .. | .. | 0·120 | |
| | | 8·597 | |

The percentage composition by weight would be:

| Constituent. | | Percentage by Weight. |
|---|---|---|
| Nitrogen | .... | 51·93 lb. |
| Carbon monoxide | .... | 46·67 ,, |
| Hydrogen | .... | 1·40 ,, |
| | | 100·00 lb. |

While by volume the percentage composition would be:

| Constituent. | | Percentage by Volume. |
|---|---|---|
| Nitrogen | .... | 43·9 c. ft. |
| Carbon monoxide | .... | 39·5 ,, |
| Hydrogen | .... | 16·6 ,, |
| | | 100·0 c. ft. |

This gaseous mixture would have a higher heat value of $0{\cdot}395 \times 341 + 0{\cdot}166 \times 347$ (Table IV) $= 192$ B.Th.U. and a lower heat value of $0{\cdot}395 \times 341 + 0{\cdot}166 \times 292 = 183$ B.Th.U. per cubic foot at 32° F. and 1 atmosphere, and would contain *all* the heat of the carbon supplied to the producer, as may be verified by observing that the whole of the carbon is gasified, and each cubic foot of the resulting gas contains

$0{\cdot}395 \times 0{\cdot}0783$ lb. of CO, or $0{\cdot}395 \times 0{\cdot}0783 \times \frac{12}{28}$ lb. of carbon, and this amount of carbon has a calorific value of $0{\cdot}395 \times 0{\cdot}0783 \times \frac{12}{28} \times 14{,}500 =$ 192 B.Th.U.

Such an ideal 100 per cent efficiency condition, though convenient for comparison with actual performances, is of course unattainable, as the various chemical reactions can only occur at a high temperature, and some of the heat of the carbon must accordingly be expended in keeping the fuel incandescent, and this is carried away in the gas made; nevertheless in actual practice efficiencies varying from 80 per cent to 90 per cent are often attained.

Thus a test of a Dowson producer * using anthracite, made in 1904 by M. A. Adam, furnished a gaseous mixture of the following composition:

| Constituent. | | Percentage by Volume. |
|---|---|---|
| Nitrogen | .... | 56·24 c. ft. |
| Carbon monoxide | .... | 20·13 ,, |
| Hydrogen | .... | 15·64 ,, |
| Methane | .... | 1·16 ,, |
| Carbon dioxide | .... | 6·09 ,, |
| Oxygen | .... | 0·74 ,, |
| | | 100·00 c. ft. |

From Table IV the higher calorific value of this gaseous mixture is 135 B.Th.U. and the lower value 125 B.Th.U. per cubic foot at 32° F. and 1 atmosphere pressure; and from Table V the volume of oxygen necessary to just burn 1 c. ft. is

$$(0{\cdot}2013 \times 0{\cdot}5) + (0{\cdot}1564 \times 0{\cdot}5) + (0{\cdot}0116 \times 2) - 0{\cdot}0074$$
$= 0{\cdot}19465$ c. ft., corresponding to $0{\cdot}19465 \times 4{\cdot}762 = 0{\cdot}927$ c. ft. of air.

The higher and lower calorific values per cubic foot of mixture-with-air are therefore $\frac{135}{1{\cdot}927} = 70$, and $\frac{125}{1{\cdot}927} = 65$ B.Th.U. respectively. The anthracite used consisted of about 87·6 per cent carbon, 8·8 per cent ash and clinker, and 3·6 per cent moisture, and by calorimeter was found to have a heat value of 13,890 B.Th.U. per pound, and 92·9 c. ft. of gas were obtained per pound of fuel consumed. The efficiency of the producer was accordingly $\frac{92{\cdot}9 \times 135}{13{,}890} = 0{\cdot}90$ on the higher heat value of the gas, or $\frac{92{\cdot}9 \times 125}{13{,}890} = 0{\cdot}836$ on the lower value.

Using small gas coke having a calorific value of 12,477 B.Th.U. per pound, this same Dowson producer, tested by Larter, showed an efficiency of 0·89 (higher value)—practically the same as with anthracite—and yielded 81·6 c. ft. of gas (measured at 32° F. and 1 atmosphere) of 136 B.Th.U. higher value, per pound of gas coke consumed.

* Of "suction" type; see p. 94.

Sir D. Clerk,* writing in 1912, observes: "In recent producer plants of either pressure or suction type the fuel cost with anthracite is roundly 0·135 pence and with gas coke 0·05 pence per 10,000 B.Th.U. of heat supplied. . . . Allowing for costs incurred in connection with the producer, as, e.g., for attendance, water, and plant repair and depreciation, it may be taken that in a 40-b.h.p. plant . . . the cost of supplying 10,000 B.Th.U. is practically the same as from coal gas at *one shilling* per 1000 c. ft." The average cost of coal gas in the London district in 1912 was nearly *three shillings* per 1000 c. ft.; the Dowson producer referred to was installed near London.

The pressure type of producer is adopted in many cases where the gas made is used for heating purposes or for both heating and power, and the fuels ordinarily employed in Great Britain are anthracite or clean gas-coke. On account of its lower cost and greater simplicity the suction type of producer (p. 94) has almost superseded the pressure type in cases where the gas made is required for power purposes only. The suction-pressure producer (p. 100) combines the advantages of both types. For bituminous fuels, however, very large pressure producers are built, operating on the Mond and allied systems, and these are distinguished as "non-recovery" or "recovery" plants, according as the nitrogen is not or is recovered from the gas made, in the form of sulphate of ammonia.

**The Mond Non-recovery Pressure Plant.**—It was for long found impossible to obtain producer gas free from tar with any fuels other than anthracite or coke. In the Mond system, however, the cheapest qualities of non-caking, bituminous small coal and "slack" are ordinarily burned, and even caking coals can be used. In this connection the Power Gas Corporation, Ltd., who supply these plants, state: "The Mond producer is not confined to non-caking coal . . . but can use almost any coal, some of the most caking qualities being in successful daily use"; certain details of design are, however, varied in accordance with the particular kind of coal to be used.

Small plants, gasifying less than about 1 ton of coal per hour, are usually of the non-recovery type, and a sectional view of a typical non-recovery plant is given in fig. 2. The distinctive feature of the process is the large quantity of steam which is mixed with the air blown through the incandescent fuel; in ordinary anthracite or coke plants the weight of steam used is commonly about $\frac{7}{8}$ of that of the fuel burned, whereas in non-recovery bituminous plants it is usually about $1\frac{3}{4}$ times, and in recovery plants up to even $2\frac{1}{2}$ times the weight of the coal consumed. These large quantities are necessary to keep the furnace temperature low so as to prevent caking and clinkering, and, in the case of recovery plants, to prevent the destruction of the ammonia in the gases formed in the producer. Non-recovery plants are built by the Power Gas Corporation in units of 100 h.p. upwards, while the recovery type is built in units of 500 to 30,000 h.p.

Referring to fig. 2 it will be seen that the producer is a fire-brick lined metal cylinder standing on a concrete foundation, with its lower end water-

* *The Gas, Petrol, and Oil Engine*, Vol. II, Clerk and Burls (Longmans).

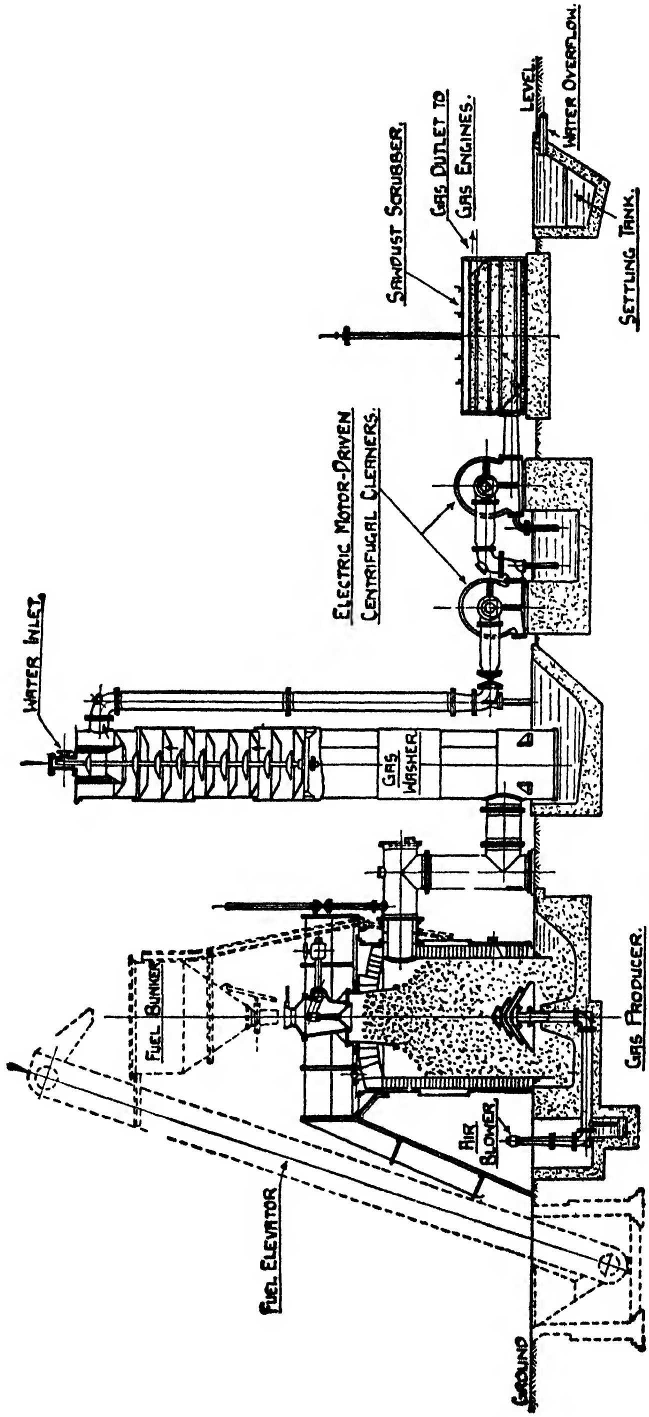


Fig. 2.—Diagrammatic Elevation of Non-recovery Power Gas Plant

sealed as indicated, thus enabling the ashes to be removed with ease without any interruption to the working of the apparatus.

*Grate and Blast.*—The grate is central, conical in form, and built up of several cast-iron sections in such manner as to leave between them passages through which a blast of mixed air and steam is delivered into the producer from the central blast-pipe shown. The blast is often created by a boiler-fed injector of the steam-jet type (cf. fig. 1), but in large plants steam-driven Root's blowers or turbo-blowers are employed.

The large quantity of steam required is obtained in part by passing the air on its way to the producer through a " saturator ", wherein it is brought into intimate contact with the hot water just previously used in the washer for gas cooling (see fig. 3), and is supplemented if necessary by a small boiler and by any exhaust steam from engines, &c.; any constant steam supply at a pressure of not less than about 2 lb. per square inch is suitable for use. In large gas-engine power plants all the necessary steam may be provided by specially designed boilers heated by the exhaust gases from the engines.

*Fuel Adjustment.*—The producer is furnished with a number of horizontal poking holes at about the level of the top of the grate, and with a second series in its top, as indicated in fig. 2, whereby the fuel can be stirred, clinker dislodged, and the formation of holes prevented by the attendant at any time without affecting the production of gas.

*Fuel Feed.*—On the producer top is a feed hopper with hand-operated sliding cover and conical distributing valve, enabling fresh fuel to be supplied without any escape of gas into the atmosphere; an operating platform surrounded by a hand-rail, with access by ladder, is provided as shown.

In large plants fuel-handling apparatus is usually included, as indicated by dotted lines in fig. 2, comprising a power-driven bucket-type elevator by which the fuel, delivered by tip-wagons into a pit or " boot " at the ground level, is hoisted and delivered into a capacious fuel bunker, fitted with a hand-regulated shoot by which the feed-hopper is supplied at necessary intervals.

*Treatment of the Gas.*—Leaving the producer by the outlet branch shown in the upper right-hand side of the illustration, the gas is passed firstly into a " Lymn " gas washer which contains no coke, liable in this type of plant to become quickly tar-fouled, but a series of water-distributing discs mounted upon a vertical central rod or shaft extending from top to bottom of the washer. Water enters at the top of the vessel, and attached to the inner surface of the cylindrical mild-steel casing, water-sealed at its lower end, is a series of equidistant inverted hollow conical frustra or " collectors ", so placed as to receive the thin sheets of water falling from the distributing discs, whence, if no gas were passing, it would return to the top of the disc next below, and so in a zigzag course to the bottom of the vessel and drain away. The hot, dusty, tar-laden gas from the producer enters, however, from the bottom, and passing upwards breaks up the thin sheets of water, pursuing a somewhat undulating path in its ascent; the gas and water are thus brought into intimate contact, the gas is cooled, and the grosser particles of suspended dust and tar removed.

The diameter of the discs being considerably less than that of the holes in the conical collectors, the gas encounters but little resistance, and thus experiences very little loss of pressure. The absence of any solid packing, such as coke, earthenware, tiles, boards, &c., in the washer which would obstruct the flow of the gas and rapidly become tar-clogged is a valuable feature of this apparatus.

From the washer the gas passes next to the first of two centrifugal tar extractors ordinarily arranged in series as indicated; each of these is provided with a water-sealed tar-draining pipe discharging into a trough. Leaving the second tar extractor, the gas passes into a sawdust scrubber, consisting in this case of a rectangular cast-iron box furnished with wooden grids on which are spread layers of wood chips, shavings, and sawdust; this scrubber in general requires refilling only at intervals of two or three months, and arrangements are provided for quickly raising the cover and removing the contents. The fouled material removed may be burned in the furnace of the boiler which provides the steam for the blast.

All liquid drained from the plant is run into a concrete pit or *settling tank* fitted with iron or reinforced-concrete partition plates so disposed as to retain the tar and allow water only to run away. The tar is collected and either burned under a boiler, distilled, or sold.

Leaving the sawdust scrubber, the gas passes to the engines or heating furnaces, or both, an automatic gas-supply governor being usually included when the producer is operated by a power-driven blower. The automatic governor comprises a gas-holder of small capacity, having its bell so arranged that when nearly full it operates a valve controlling the blast, thus regulating the gas production; such an arrangement, operating on the steam valve, however, is indicated in fig. 1.

An ample supply of cold water is necessary for the washer, producer lute, troughs, settling tank, tar extractors, &c. When the supply is limited it becomes necessary to install special water-cooling and circulating plant in order to save as much as possible.

*Fuels Used.*—The fuel most commonly burned in Mond plants is bituminous slack, but plants are also in existence using coke, peat, lignite, coke-breeze, colliery-refuse, wood-blocks, and other cheap fuels.

From a non-recovery Mond plant using a non-caking bituminous coal having a lower calorific value of about 14,000 B.Th.U. per pound, about 150,000 c. ft. of gas were obtained per ton of fuel burned, or, say, 67 c. ft. of gas per pound of fuel. The composition of the gas, by volume, was found to be:

| Constituent. | | Percentage by Volume. |
|---|---|---|
| Hydrogen | .... | 17·36 c. ft. |
| Carbon monoxide | .... | 25·55 " |
| Methane | .... | 1·20 " |
| Oxygen | .... | 0·30 " |
| Carbon dioxide | .... | 5·77 " |
| Nitrogen | .... | 49·82 " |
| | | 100·0 c. ft. |

and its lower calorific value per cubic foot at 32° F. and 1 atmosphere about 149 B.Th.U. per cubic foot, corresponding to a producer efficiency of 71·3 per cent, exclusive of any fuel used in raising steam, &c.

**The Mond Pressure Plant with Ammonia Recovery.**—When it is desired to recover the ammonia in the fuel, it is necessary, as already stated, to keep the producer temperature lower, which is done by increasing the amount of steam blown through the fuel up to as much as 2½ lb. per pound of coal in some cases. This large quantity of steam renders it necessary to utilize carefully all possible sources of spare heat. The efficiency of the producer itself is diminished, but the yield of the valuable ammonia sulphate more than compensates for the diminution. Some early experiments * by Messrs. Bone and Wheeler upon a 2500-b.h.p. Mond recovery plant, using a non-caking bituminous slack, showed that when only 0·45 lb. of steam was used per pound of coal, 38 lb. of ammonia sulphate were obtained per ton of coal gasified, with a producer efficiency of 71·5 per cent. On increasing the steam supply to 1·55 lb. per pound of coal the yield of sulphate was increased to 71·8 lb. per ton of coal, but the producer efficiency had then fallen to about 60·4 per cent. With later plants 90 to 100 lb. of sulphate are obtained per ton of slack gasified.

Many producer plants operating on the Mond and Lymn (see below) methods have been installed by the Power Gas Corporation, Ltd.; by 1929 their normal practice was to combine low-temperature production and efficient gasification in one operation; a comparison of average yields by the earlier and later processes is given in Table IX (see p. 82). This shows that the later process furnishes a somewhat smaller volume of producer gas of higher heat value than the older methods, the total calorific value per ton of coal gasified appearing as 18,492 B.Th.U. by the older, and 21,716 by the newer process.

Generally, the later process consists in maintaining the fuel supplied to the upper part of the producer at the temperature found most suitable for the maximum yield of tar oils, &c. The tar oils and ammonia produced in the second and third (gasification) zones pass on through zones of falling temperature, thus preventing cracking of the oils and decomposition of the ammonia.

The gas yielded by Mond plants is employed in a great variety of industrial processes, including firing coke ovens, reheating furnaces, glass and steel melting, annealing, tempering and chemical furnaces, calcining ores, evaporating brine, melting copper, brass, tin, and lead, drying cores and moulds, firing steam boilers, and for power development in gas engines; in 1929 it was estimated that over 450,000 b.h.p. was thus attained. The tar oils yielded, readily dehydrated by mechanical means to less than 2 per cent moisture content, and the sulphate of ammonia and pitch, are very valuable by-products.

Thus an estimate by the Power Gas Corporation of the annual cost of operating a low-tension recovery and gas-producing plant dealing with

* Made in 1906.

126 tons of coal per day on six days per week showed a total annual cost of £56,328, with a corresponding credit from the sale of the sulphate of ammonia, heavy and light tar oils, and pitch obtained of £39,916, leaving a debit of £16,412 as the cost of the gas produced, which corresponds roundly to 1200 c. ft. of gas for one penny.

TABLE IX *

*Analysis of fuel used* (*on dry fuel*):

| | |
|---|---|
| Moisture | 5·7 per cent |
| Ash | 8·0 ,, |
| Volatile matter | 31·2 ,, |
| Total carbon | 71·1 ,, |
| Fixed carbon | 56·6 ,, |
| Hydrogen | 5·0 ,, |
| Nitrogen | 1·47 ,, |
| Net calorific value, B.T.U.'s per pound | 12·150 |

*Comparative Results* (*Old and New Processes*)--*Long Period Averages*

| *Gas Analysis*: | Old Process. | New L.T. Process |
|---|---|---|
| $CO_2$ | 16·0 per cent | 8·3 per cent |
| CO | 11·0 ,, | 21·0 ,, |
| H | 25·0 ,, | 20·5 ,, |
| $CH_4$ | 2·5 ,, | 4·9 ,, |
| N | 45·5 ,, | 45·3 ,, |
| B.T.U.'s net per cubic foot | 134 | 178 |
| Gas efficiency $\frac{\text{calorific value of gas}}{\text{calorific value of coal}}$ | 68 per cent | 80 per cent |
| Average gas yield per ton of dry fuel gasified | 138,000 c. ft. | 122,000 c. ft. |
| Average ammonium sulphate yield per ton of dry fuel gasified | 95 lb. | 90 lb. |
| Average yield of dry tar and oils per ton of dry fuel gasified | Low-grade quality; value negligible. | 21 gall. |
| Steam per pound of dry fuel gasified | 2½ lb. | 1 lb. |

**The Lymn Pressure Recovery Plant.**—An interesting example of a recent recovery plant of this type is that at the Chelmsford works of the Hoffmann Manufacturing Co., Ltd., on the River Chelmer.† The whole installation comprises two 30-ton units each consisting of two producers, one of which is illustrated diagrammatically in fig. 3.

* Reproduced by courtesy of the Power Gas Corporation Ltd.
† See Patchell, *Proc. I. E. E.*, 1920.

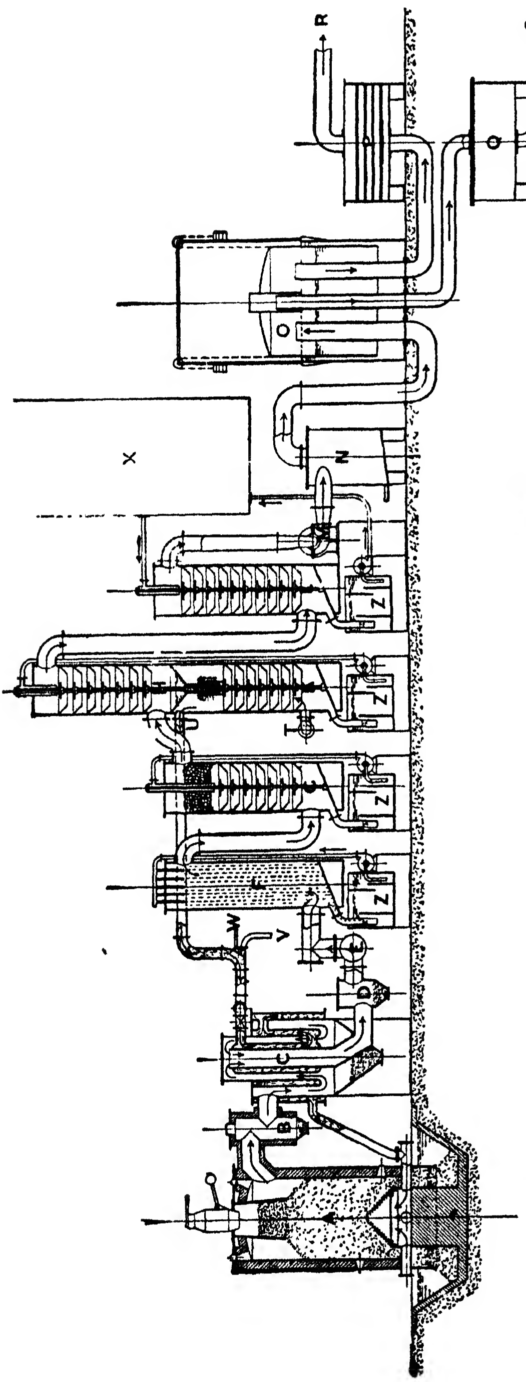

Fig. 3.—Diagram of Lymn Gas Plant at Chelmsford

*The Producer.*—Shown on the extreme left of the diagram, the producer A is a cylindrical mild-steel vessel lined with fire-brick and furnished with the usual type of air-locking feed hopper in the crown. The poke holes will be noticed. The conical grate is fitted centrally, and the whole apparatus is borne on a massive foundation of concrete as indicated.

The lower end of the producer is open but water-sealed. A mixture of heated air and steam is delivered below the central grate as shown, and passes upwards through the mass of ignited fuel, the resulting gases leaving by way of a fire-brick lined branch on the right into the cooling and cleaning plant.

*Fuel and Ashes.*—The installation was laid down during the 1914–18 war, and for some time shortage of coal resulted not only in a considerable variation in the quality of the fuel consumed but also in *mixtures* of different coals being used, which, in general, renders it impracticable to operate a plant to the best advantage. During a period of six months' working, one 30-ton unit gasified 3255 tons of coal of an average calorific value of 11,333 B.Th.U. per pound; the ashes withdrawn were found to contain 11·8 per cent of carbon, representing 3·23 per cent of the coal consumed. The small coal used is delivered by railway trucks into the boot of a bucket-type elevator (cf. fig. 2), from which it is delivered by a hand conveyor to 30-ton bunkers (not illustrated) fixed above each producer.

The ashes and clinker are withdrawn from the producer water lutes by long rakes and shovels on to the surrounding ground, which is so sloped as to drain back any water into the lutes; they are next loaded into 5-cwt. hand-propelled narrow-gauge trucks from which they are finally tipped into carts or railway wagons as desired.

*Products of Gasification.*—Per ton of coal gasified, from 130,000 to 140,000 c. ft. of gas are obtained, together with about 90 lb. of ammonia sulphate and 150 lb. of tar.

The calorific value of the gas produced during three months' working varied only about ± 7 per cent from the mean value; the average composition of the gas from forty-two tests taken during this interval was found to be:

(*From W. H. Patchell*)

| Item. | | Percentage by Volume. |
|---|---|---|
| H | .... | 26·60 c. ft. |
| CO | .... | 10·68 „ |
| $CH_4$ | .... | 2·59 „ |
| O | .... | 0·15 „ |
| $CO_2$ | .... | 15·80 „ |
| N | .... | 44·10 „ |
| | | 99·92 c. ft |

corresponding by Table IV to calorific values per cubic foot at 32° F. and 1 atmosphere of 156 B.Th.U. higher and 138 B.Th.U. lower.

From the formula for ammonia sulphate, viz. $(NH_4)_2SO_4$, it appears that 28 lb. of nitrogen yield 132 lb. of sulphate; the coal used contained about 1·3 per cent of nitrogen, i.e. 29·12 lb. per ton, corresponding to a theoretical yield of 137·3 lb. of sulphate. As only about 90 lb. are obtained, the efficiency of the sulphate recovery plant is 65·5 per cent in this case.

The tar obtained differs in quality from gas-works tar, and contains also considerable quantities (up to 40 per cent at times) of water. In this installation it is used for steam-raising in boilers, and is burned in Kermode sprayers.

*Production of the Gas.*—Air at normal temperature is delivered by Samuelson blowers through a pipe T (fig. 3) into the lower part or "saturator" of a compound Lymn washer K, wherein, passing upwards, it comes into intimate contact with hot water just previously used for cooling the gas in the upper portion of the vessel—rising in temperature about 80° F. and taking up some moisture. Leaving the saturator at U it is next delivered into the superheater C, receiving on its way exhaust steam from certain steam-engines through a pipe V, and the live steam to complete the necessary supply of moisture through the smaller pipe W from boilers heated by the exhaust from the gas-engines.

To avoid overcooling the fuel in the producer by introducing into it air heavily loaded with moisture, the mixture is first passed through a "regenerator" or "superheater" C, consisting of a nest of co-axial cylinders through one set of which the moist air passes towards the producer, while through the other go the hot gases from the producer. Leaving this superheater at a temperature of about 330° F., the moist air is next delivered below the grate, and thence passes upwards through the incandescent fuel in the producer as indicated.

The hot and smoky gases made leave the producer at a temperature of about 930° F., and pass firstly through a dust pocket B wherein grosser particles are deposited, and next through the superheater C, leaving this at a temperature of about 680° F., after heating the moist air as just described. The gases next pass through a second dust pocket D, via a "bus" main E, into the bottom of the first washer and dust extractor F,* which consists simply of an empty mild-steel cylinder fitted with jets in the top cover, whereby it is filled with water spray. Passing upwards through this spray, the gas is somewhat cleaned and cooled down to about 205° F. The water runs off from the sloping bottom of the washer through the water-sealed pipe shown, into a settling tank Z fitted with partitions to facilitate the removal of the tar; from this tank it is continuously circulated through the washer at the rate of 200 gall. per minute by the small centrifugal pump shown.

*The Ammonia Absorber.*—From the washer F the gases pass next to the bottom of a mild-steel Lymn ammonia absorber G containing a series of distributing discs and collectors, as described on p. 237, together with a layer of coke resting on a grid in the upper portion of the vessel as indicated. Through this an acid solution of ammonia sulphate is sprayed by a small centrifugal pump of bronze, the ammonia in the gases uniting with the acid; the liquid drains from the sloping bottom of the absorber through a water-sealed pipe into a tank Z. The process is continuous, and when the liquor becomes sufficiently concentrated some is removed from the tank and replaced by a 4 per cent solution of sulphuric acid and water.

Ammonia absorbers are usually lead-lined to prevent corrosion by the acid liquor; in this case, owing to war conditions, the vessel was made of mild steel. Steel Lymn absorbers had, however, been successfully at work in Germany for upwards of three years prior to 1914. In Messrs. Hoffmann's plant no ammonia was recovered until the plant had been running for some weeks to enable the absorber to acquire a good coating of tar before any acid liquor was circulated through it. Excepting only a few bolts and rivets, no trouble has arisen from corrosion.

*Cooler and Saturator.*—Leaving the ammonia absorber at a tem-

* One each of F, G, H, K, &c., to each *pair* of producers.

perature of about 165° F., the gases pass next to the upper portion H of the compound Lymn cooler and washer mentioned above; the water from the vessel H passes through a simple water-seal as indicated, from which it sprays into the lower portion, or air saturator, K, already described.

*Final Cooler.*—Leaving H as shown, the gases pass next through another cooler and washer L, the water from which, instead of being circulated round and round, as in the three preceding cases, is passed, by the small centrifugal pump shown, through a water cooler X. From L the gases, reduced now to a temperature of about 110° F., are passed through two centrifugal tar separators, one of which is shown at M; each consists of a centrifugal fan 25 in. in diameter, driven at about 1550 r.p.m., and capable of delivering 4500 c. ft. of gas per minute against a pressure of 5 in. of water. Each fan requires about 11 h.p., and into the centre of each is injected about 500 gall. per hour of cold water. The mixed tar and water drain off from the fan outlets into a settling tank from which the tar is periodically skimmed off. The great difficulty in the use of bituminous fuels is the extraction of the suspended tarry matter in the gas made, and especially of the *tar fog*, which, if allowed to pass through, quickly chokes the engines and burners. The last traces are due to lighter tars of low condensing temperatures, and hence it is important to reduce the temperature of the gases as much as possible on their arrival at the tar extractor. These centrifugal extractors, driven at a high peripheral speed, are very efficient, tests of a Crossley tar extractor in 1909 showing an average efficiency of extraction of 96 per cent, 100 c. ft. of gas containing from 6 to 12 gr. of tar on entry having only ¼ to ½ gr. on exit.

*Water Separator.*—From the tar extractors the gases, heavily charged with moisture, are passed next through a *cyclone*-type water separator N, by which they are dried, and pass next into the *gas-holder governor* O, which is a small-capacity holder from which proceed pipes for supplying the gas-engines and furnaces respectively. The gas-engines are supplied from the central pipe, and it will be observed that if the supply of gas should fail, the holder in its descent cuts off first the connection with the engines, thus preventing any suction from coming on the furnace burners, which would extinguish them and might possibly cause an explosion if the gas supply were suddenly restored.

*Sawdust Scrubbers.*—From the gas-holder O the gases, now reduced to a temperature of about 85° F., pass through sawdust scrubbers P and Q, and issue at R and S to the furnaces and engines respectively.

Each scrubber is 13½ ft. long, 6 ft. wide, and 5 ft. deep, and contains four sets of trays covered with sawdust and shavings, and finally a sheet of sacking or blanketing to prevent any of the sawdust being carried forward to the engines, &c. Draining pipes are provided to remove any water or tar that may collect in the scrubbers.

The diagram shows the usual arrangement with the gas passing upwards, in which case the lowermost layer of sawdust is the first to become fouled; arrangements were later made to pass the gases either upwards or downwards; in a downward-flow scrubber the uppermost layer becomes fouled first, and is readily removed and replaced with clean sawdust without disturbing the trays beneath.

*Pressures throughout the Plant.*—In Mond plants it is usual to keep the pressure of the gases always slightly above atmospheric; in this Lymn plant the pressure is a little *below* atmospheric from its entry into the ammonia absorber to its entry into the centrifugal tar extractors; actual pressures at several points are about as given below:

| | |
|---|---|
| Blower delivery into saturator .. .. | 16 in. of water. |
| Below grate of producers .. .. .. | 12 in. ,, |
| Gases on leaving producers .. .. | 3½ in. ,, |

Thence the pressure falls steadily until at entry into the tar extractors it is 7½ in. of water *below* atmospheric. From the tar extractors it is delivered into the gas-holder at a pressure of about 9 in. of water above atmosphere, and finally leaves the sawdust scrubbers at about 3 in. of water above atmosphere. 1 in. of water corresponds to a pressure of 0·036 lb. per square inch.

*Steam Used.*—A considerable amount of steam is necessary for the producers for evaporating the ammonia sulphate liquor, and for heating tar troughs and tanks to maintain the tar used in burning in a sufficiently fluid condition. By using boilers heated by the exhaust from the gas-engines, and burning the tar under other boilers, the actual coal required for producing all the steam necessary is reduced to 6½ per cent of the coal gasified in the producers.

*Performance of the Plant.*—From observations taken over six months' working of one 30-ton plant, Mr. Patchell states that the total coal gasified was found to be 3255 tons, and that of the gas produced 42·7 per cent was used by the engines and 57·3 per cent by the furnaces. Hence $0{\cdot}427 \times 3255 = 1390$ tons of coal were required to supply the gas-engines alone during this period. Crediting the whole of the steam obtained from the engine-exhaust-heated boilers to the engines, and 42·7 per cent of the steam obtained by burning the tar actually produced, taking as equivalents:

| | |
|---|---|
| 1 b.h.p.-hour in engines | = 2 lb. steam from exhaust boilers, |
| 1 lb. coal under Lancashire boilers | = 6 lb. steam, |
| 1 lb. tar under ,, ,, | = 7½ lb. steam, |

he obtains valuable figures relating to actual performance, some of which are given below:

### Lymn Plant at Chelmsford

Some results from six months' working at one 30-ton plant
(From W. H. Patchell)

*Power Production only*

| Item. | Tons. |
|---|---|
| Coal gasified in producers .. .. .. .. .. | 1390 |
| Equivalent coal corresponding to surplus steam made .. .. | 147 |
| Net coal debited against power plant .. .. .. | 1243 |

The exhaust-heated boilers *alone* supplied more steam than was necessary to operate the power plant, and the excess, plus that obtained by the burning of 42·7 per cent of the tar produced, is the origin of the 147 tons above.

During the period under observation the engines generated 1,834,375 kw.-hr., corresponding to an average engine load factor of 73·6 per cent on running time; the coal gasified per kilowatt-hour was therefore $\frac{1243 \times 2240}{1,834,375}$ $= 1{\cdot}51$ lb.; and, as 1 kw.-hr. corresponds to 3412·5 B.Th.U. per hour, the over-all thermal efficiency of the power plant, from the coal fed into the producer to the electrical energy appearing at the switchboard, was $\frac{3412{\cdot}5}{11,333 \times 1{\cdot}51} = 0{\cdot}199$, or 19·9 per cent.

**The Lymn-Rambush Pressure Producer.**—For the gasification of bituminous fuels of widely varying type, to prevent clinkering, and to minimize labour, in the Lymn-Rambush producer a slowly rotating gear-driven grate is fitted; this grate is circular in plan, and is furnished with spiral-topped quarter sections mounted eccentrically within the producer. The mixed blast of air and steam is introduced into the fuel through the vertical grids at the ends of the quarter sections, which remain free and open as the direction of rotation of the grate moves them always away from the mass of the fuel. The eccentrically mounted spiral top of the grate also communicates to the fuel in the producer a continuous combined slow vertical and lateral motion, thus preventing the formation of adherent clinker and keeping the fire always in order. Mechanical arrangements are provided, in addition, for the removal of the ashes from the water lute at the base of the producer.

**The Kerpely Pressure Producer.**—This producer also has an eccentrically mounted revolving grate, and in conjunction with a "Bentley" mechanical fuel feed, is illustrated diagrammatically in fig. 4.*

*Supply of Moist Air.*—The blast of combined air and steam enters beneath the grate into the water-sealed space AA′A′ and passes upwards into the ignited fuel through the "Kerpely" grate BB. The moisture-laden air is delivered separately and quite independently to the central and peripheral portions of the grate, the inner chamber A conveying the blast to the central portion, while from the outer chamber or blast pit A′A′ the peripheral portions are supplied, as indicated by the arrows.

*The Grate.*—The Kerpely grate BB is built up of a number of massive interlocking sections of hematite iron plate showing jointly a conical disposition in vertical section, but appearing in plan as a somewhat elongated twelve-sided polygon. The grate rests on stout castings CC, which in turn are supported upon the bottom of the water trough D, the whole being borne upon a ball-bearing turn-table EE, and continuously slowly rotated by a worm acting upon the worm-wheel FF.

This slow rotation of the eccentric grate and trough prevents the grate

* By courtesy of Messrs. E. G. Appleby & Co., Ltd.

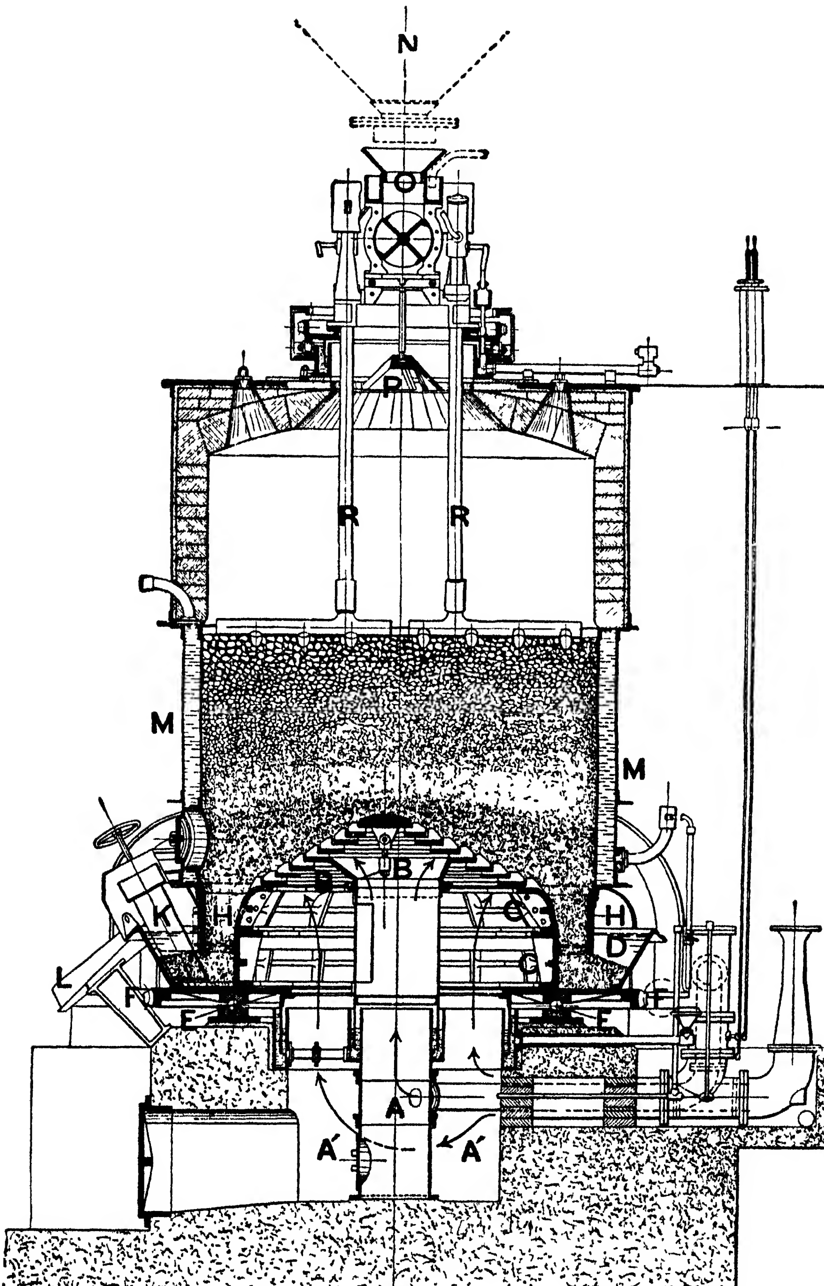


Fig. 4.—Kerpely Gas Producer

spaces becoming choked, ensures that the blast passes equally throughout the mass of fuel, and breaks up the clinker by crushing it sideways against the water-seal apron HH. The rotation of the trough causes the ashes to pile up against the (adjustable) fixed ash scoop K, and thus discharge themselves automatically and continuously into the ash shoot L.

*Body of Producer.*—The body or *shell* of the producer is borne by four heavy cast-iron brackets standing upon a substantial foundation of concrete; these are indicated in fig. 4, but are shown much more clearly in fig. 5. The lower half of the shell is water-jacketed, as shown at MM, a constant circulation of water being maintained through it. By this means the adhesion of clinker is prevented, the labour of excessive poking avoided, and the damage to brick lining eliminated, while there is no longer any need to introduce an excessive amount of steam into the producer when burning caking or clinkering fuels. The upper portion and crown of the vessel are protected by the usual fire-brick lining.

*Fuel Feed.*—The fuel, raised by an elevator and delivered by a band conveyor into 15-ton overhead bunkers N, is thence transferred by gravitation into the hopper O of a Bentley mechanical fuel feeder in which a charging drum, as indicated in section, is slowly rotated, thus delivering the fuel at a uniform and adjustable rate via a distributing cone P into the body of the producer. In the design illustrated in fig. 4, the whole device is borne upon roller bearings and rotated by a power-driven worm actuating a water-sealed worm-wheel, as shown in section. Attached to the frame of the feed device, and rotating with it about the axis of the producer, are the two fuel agitators RR, of solid-drawn hydraulic tubing, each fitted with a horizontal arm to which are attached *fingers* of a special heat-resisting alloy. The tubes RR, and the horizontal arms, are water-cooled.

The agitators are balanced on ball-bearings and are adjustable, so that they may be set to work either at or somewhat below the surface of the fuel in the producer as found preferable. They are removable through the hole in the producer crown, and the fingers are readily replaced when necessary.

The constant slow movement imparted to the mass of fuel by the moving grate and the agitators jointly, considerably hastens and improves the gasification of the fuel, prevents unequal burning of different parts of the fire, and tends to produce uniformity in the quality of the gas yielded; the fuel movement, in conjunction with the water-jacket MM, by preventing the formation of adherent clinker, renders it possible to use much less steam than is usual in the blast, resulting in the production of a drier gas, which is a feature of value.

An external view of a pair of Kerpely producers is shown in fig. 5. The two blowers supplying the blast for the central and peripheral portions of the grate will be noted in the foreground; two of the four heavy cast-iron brackets supporting the producer body and the worm-wheel through which the grate is rotated are also shown clearly. On top is seen the mechanical feed apparatus with the large fuel bunkers suspended above from a roofed-over open-sided and open-ended frame of rolled-steel double-tee joists.

*Performance.*—From results observed on a producer as just described,

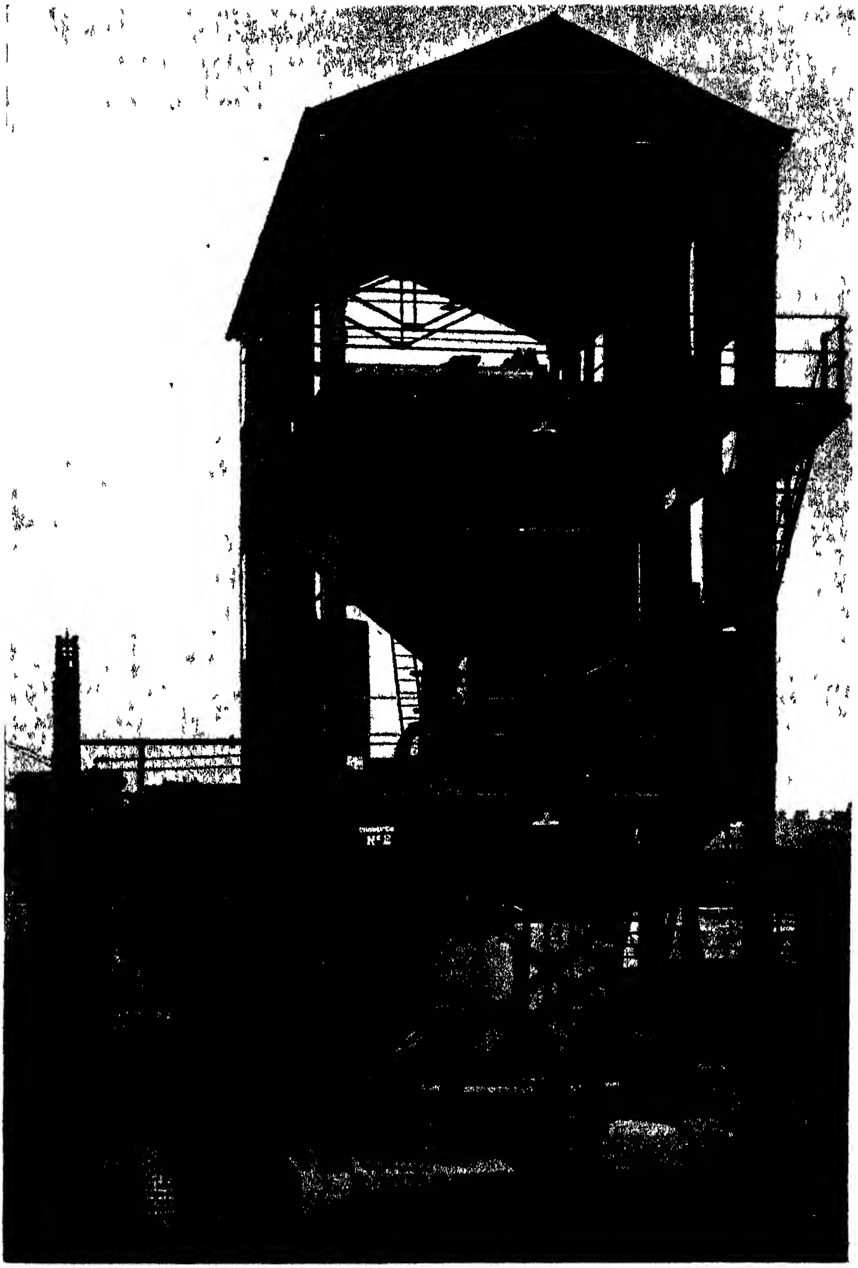


Fig. 5.—External View of a Pair of Kerpely Gas Producers with Bentley Mechanical Feed

during ten working days, using as fuel a Mansfield coal, the following average figures have been deduced:

Steam pressure at blowers:

| | | |
|---|---|---|
| For outer blower | .. .. | 24·0 lb. per square inch. |
| For inner blower | .. .. | 47·0 " " |
| Saturation temperature of air | .. | 137° F. |
| *Carbon in ash* | .. .. .. | *2·77 per cent.* |

*Analysis of gas yielded:*

| | | Per Cent. |
|---|---|---|
| H | .. .. .. | 13·22 c. ft. |
| CO | .. .. .. | 28·20 " |
| $CH_4$ | .. .. .. | 3·68 " |
| N | .. .. .. | 51·54 " |
| $CO_2$ | .. .. .. | 3·36 " |
| | | 100·00 c. ft. |

Proportion of combustible in gas made .. 45·1 per cent

Calorific value of gas in British thermal units per cubic foot at 32° F. and 1 atmosphere:

| | | |
|---|---|---|
| Higher value | .. .. .. | 181 |
| Lower value | .. .. .. | 169·5 |

**The Kerpely "High-pressure" Producer.**—For effectively dealing with low-grade fuels, as, e.g., coal dust, coke breeze, blast-furnace coke refuse, anthracite *slimes*, &c., it is necessary to employ a blast pressure considerably higher than usual, on account of the great resistance offered to the passage of the mixed air and steam by such closely packing fuel. The Kerpely high-pressure producer overcomes this difficulty while retaining the various automatic features already described by the provision of an enclosed water-seal, which permits the introduction of air above the seal at the same pressure as that introduced below the grate, so that any necessary blast pressure can be employed without risk of blowing the seal.

Upwards of one thousand Kerpely producers have been installed, including both recovery and non-recovery plants.

The reader will have noted the large amount of cleaning plant necessary when low-grade bituminous fuels are used, and particularly the difficulty of removing the tar and tarry vapours suspended in the gas. Much attention has been devoted to the problem of preventing the formation of tar when gasifying good and medium grades of non-caking bituminous fuel, and since 1910 many makers of gas plant, e.g. Campbell, Crossley, Dowson, &c., have constructed special producers, in most cases of the suction type, in which such fuels are successfully dealt with, the gas yielded requiring only normal cooling and scrubbing. In several of these the formation of tar is minimized by removing the gas from the hot zone of the producer, and arranging for the supply of steam and air through the fuel both upwards and downwards (see fig. 11). The tarry ingredients in the fuel are first volatilized, and on passing through the fire zone are next burnt and converted into fixed gases; cleaning and cooling is then effected by the usual wet-coke scrubber and sawdust scrubber, sometimes with the addition of a small centrifugal separator between them.

**Blast-furnace Gas.**—In the discharge of its function of producing pig-iron, the blast-furnace yields enormous quantities of gas of low calorific value which is nevertheless an excellent fuel for use in gas-engines, and a valuable heating agent for use in steam-raising, heating coke ovens and blast air, calcining ore, &c. A diagrammatic section of a blast-furnace is shown in fig. 6. The apparatus is of great size, up to 100 ft. in height and 25 ft. in internal diameter, and consists essentially of a brick-lined cylindrical steel casing, within which is a second lining of fire-brick some 18 in. in thickness. The furnace is charged through a cup-and-cone feed hopper A with a mixture of ore, limestone, and coke, and combustion is maintained by air at a temperature from 900° to 1400° F. blown in through *tuyeres* B, C at a pressure ranging from 6 to 12 lb. per square inch. The result of the actions within the furnace is that molten iron collects in the bottom at E, covered with a layer of molten slag, and that large quantities of gas issue from D, which may be used for heating, drying, and power purposes; the yield of gas may be roughly estimated as 1½ millions of cubic feet per furnace per hour.

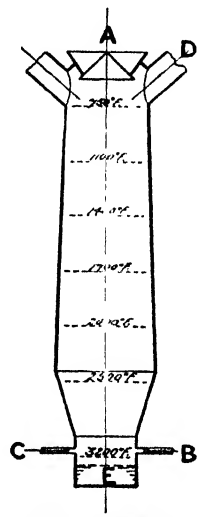

Fig. 6.—Diagram of Blast-furnace

As no steam is used the gases should contain no hydrogen; actually a small amount is found, arising from the moisture contained in the air blast and in the furnace ingredients.

The blast-furnace may thus be regarded as a large gas producer of the type referred to on pp. 72–73, and accordingly very high temperatures are attained within it, as indicated in the diagram.

Roughly, average blast-furnace results may be taken as follows:—

To produce 1 ton of iron about 1¼ tons of coke and 5 tons of blast air are required, and roundly 180,000 c. ft. of gas are evolved (estimated at 32° F. and 1 atmosphere). The composition of this gas is somewhat variable, depending upon the nature of the ore and amount of blast air introduced, and accordingly its calorific value ranges from about 85 to 120 B.Th.U. per cubic foot at 32° F. and 1 atmosphere. A typical average composition may be taken as:

| | Per Cent. | |
|---|---|---|
| H .. .. | 3·0 c. ft. | 30 per cent combustible. |
| CO .. .. | 27·0 „ | |
| $CH_4$ .. .. | Trace | |
| N .. .. | 60·0 „ | |
| $CO_2$ .. .. | 10·0 „ | |
| | 100·00 c. ft. | |

Such a gas has, by Table IV, a higher calorific value of 102·5, and lower of 101 B.Th.U. per cubic foot, and by Table V requires for just complete combustion 0·7 c. ft. of air per cubic foot. Thus the calorific value per cubic foot of mixture with air averages about $\frac{100}{1 \cdot 7} = 60$ B.Th.U.

*Cleaning.*—The greatest practical difficulty in the utilization of blast-furnace gas for power purposes proved to be that of effectually removing the large quantities of grit and dust suspended in it, and particularly in eliminating the light impalpable powder, so fine as in some cases to pass through a thick layer of felt. The gases are now satisfactorily cleaned in several ways, as, e.g., in large centrifugal washers. Space does not permit of a description being given here, but the reader will find a full and valuable account in Donkin's treatise.*

**The Suction Producer.**—In this type of gas producer advantage is taken of the suction created by the piston of the gas-engine during its charging stroke to draw the mixed air and steam through the ignited fuel in the producer. The advantages gained are: (1) the separate high-pressure boiler (30 to 50 lb. per square inch) and gas-holder of fig. 1 are dispensed with altogether; (2) the pressures within the apparatus are throughout rather less than atmospheric; and (3) that gas is only made as required for the immediate use of the engine.

The first suction producer was built by M. Leon Bénier of Paris in 1894, but it was not until about 1903 that all practical difficulties were finally overcome, when Dowson and others commenced to build them on the commercial scale. They are now manufactured by a large number of firms, and have largely superseded the pressure type of producer except in certain special cases, as, e.g., where low-grade bituminous fuels are available in large quantities.

A diagrammatic section of a suction producer plant of the earlier closed-hearth type is shown in fig. 7. When using anthracite or coke as fuel the whole apparatus is, as will be noted, very simple, comprising a closed producer A, wet coke scrubber F, and sawdust scrubber G. The branch H leads to the engine, the suction from which draws through the producer air from B and steam from C, as shown by the arrows.

The steam is supplied from the small vaporizer DD mounted in the upper part of the producer, and kept supplied with water as necessary through E; this vaporizer is heated by the gases leaving the producer, which pass through holes within it as indicated.

The results of tests made on a plant of this type have already been given on p. 77.

*Suitable Fuels.*—For suction producers of this type the most suitable fuels are: (1) dust-free anthracite of Welsh or other good quality, used in pieces from $\frac{5}{8}$ to 1 in. in size; (2) washed gas coke, in pieces from $\frac{1}{2}$ to $\frac{3}{4}$ in. in size; and (3) charcoal of good quality, in pieces from $\frac{5}{8}$ to 1 in., is a good fuel when available. With inferior anthracite or coke it is usually necessary

* *Gas and Oil Engines* (Griffin, 1911)

to install a tar separator as part of the cleansing plant; this often consists of a connected series of baffle plates fitted in a box through which the gas is passed; the tar deposits on the plates, which are readily taken out and at once replaced by a spare cleaned set at intervals of a few days. The fouled plates are cleaned by scraping, burning off, or immersion in paraffin. The tar extractor is sometimes of the centrifugal type as already described.

*Starting the Producer.*—The feed hopper being opened, a small

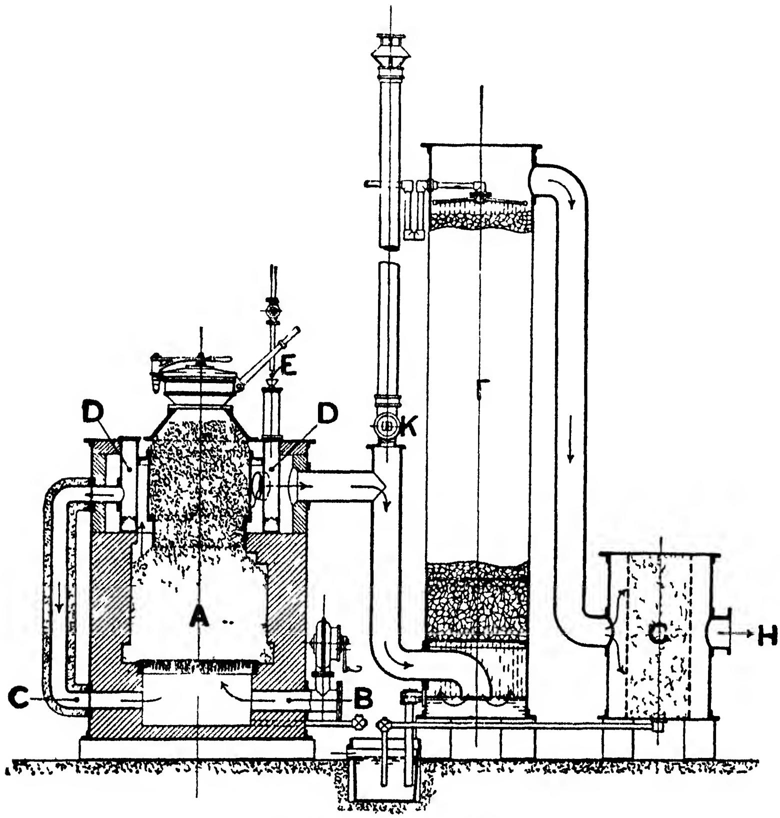


Fig 7.—Dowson Gas Plant. Closed-hearth Suction Type

quantity of shavings, firewood, and coal, sprinkled with paraffin, is dropped on to the hearth and lighted; if the producer be out of doors the hopper may be left open with advantage. The waste cock K being opened and the vaporizer charged with water through E, the fire is next gently blown up by means of the hand fan shown, a further small quantity of fuel being added as soon as the first is well alight. The use of the hand fan is continued and the charge of fuel gradually completed, until a definite incandescent zone is established. Gas now begins to come off, the hopper is closed, and the quality of the gas tested at a small cock fitted in the gas main near the producer, until it burns steadily with an orange-red flame.

In the gas main near the engine is a second waste cock and second test cock which are now opened, while the cock K is closed and the water spray turned on in the coke scrubber F. The use of the hand fan is continued until good gas, as determined by this second test cock, reaches the engine. All waste and test cocks may now be closed and the engine started, as described in the article on " Gas-engines ", section " Starting ", p. 21.

*Working the Suction Plant.*—When using anthracite the producer must be charged every three to four hours; with gas coke as fuel every two to three hours. The fuel is never allowed to fall below the level of the bottom of the charging chamber.

The ashes are removed and the fire-bars cleared of clinker, &c., every three to four hours through the ash-pit doors, which must be closed as quickly as possible. Doors are also fitted in the producer giving access for cleaning the fire above the grate; these must not be allowed to remain open more than a few seconds at a time, or the quality of the gas will deteriorate. Poking is necessary from time to time to prevent the fire burning into holes and to dislodge clinker.

The vaporizer supply should be such that a few drops fall constantly from its overflow pipe; it is then certain that the quantity of water within it is adequate; the temperature of the water should be at least 140° F.

The spraying water supplied to the coke scrubber should be sufficient to maintain the upper part of the vessel quite cool. The scrubber is carefully hand packed with washed gas coke of the best quality in pieces from 4 to 5 in. in size, with a 6-in. top layer of pieces about 2½ in. The lower portion of the scrubber is made of cast iron, as this resists corrosion better than mild steel, and the coke in this portion is removed through a door provided in the side at intervals of about a month; the upper portion needs renewal in general only once every six to nine months.

When good anthracite is used as fuel the sawdust scrubber is frequently omitted, and the gas is passed from the coke scrubber into an *expansion chamber*, consisting of a simple empty cylindrical vessel introduced to diminish the irregularity in the gas flow due to the engine suction, and obtain a steadier current of steam and air through the producer; an expansion chamber is illustrated in figs. 8 and 9.

*Standing-by and Cleaning.*—During the night the gas admission valve to the engine and scrubber spray cock are closed, the fire is cleared of clinker, the producer is recharged, and the waste cock K is opened. The ash-pit door is then opened, so as to provide sufficient air to maintain very slow combustion of the fuel throughout the night. It is also usually necessary to completely empty the producer once or twice a week, in order that any clinker adhering to the fire-brick lining may be carefully removed.

**The Anderson-Grice Producer.**—In cases where non-stop runs for long periods have to be made, two producers are frequently installed; either can then be temporarily laid off for cleaning or repair without necessitating stoppage of the engine. The Anderson-Grice double-suction producer is illustrated in fig. 8; it will be observed that this firm's producers are square in section, which permits ordinary square fire-bricks to be employed for the linings. These are procurable anywhere and can be set by unskilled labour. The lower halves of these producers are made in cast iron, as this material resists corrosion better than mild steel.

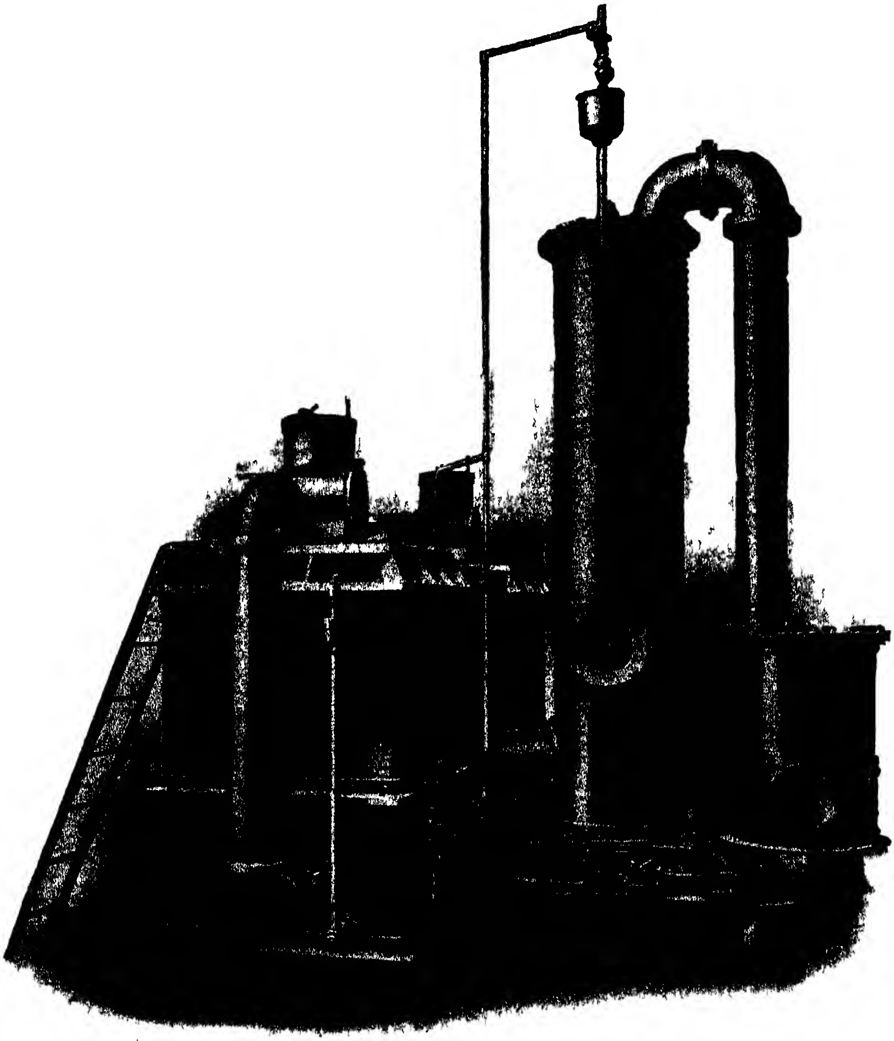

Fig 8 —Anderson-Grice Double-suction Producer

**Open-hearth Suction Producers.**—With the closed-hearth suction producer it is difficult to use low-grade small coal fuels, as the fire-bar spaces have to be very narrow, and clogging up is frequent; the necessity of often clearing the grate also renders the quality of the gas yielded liable to frequent impairment. Accordingly in such plants, as has been seen, it is usual to gasify only washed and dustless anthracite or coke, or charcoal in pieces varying in size from ½ to 1 in. cubes.

Very large quantities of such inferior fuels as small coal, " breeze ", and locomotive smokebox " char " are obtainable, and an important recent advance in design enabling such cheap and plentiful fuels to be used successfully is that of the open-hearth suction producer.

A simple plant of this type, rated at 50 b.h.p., is shown in fig. 9, which

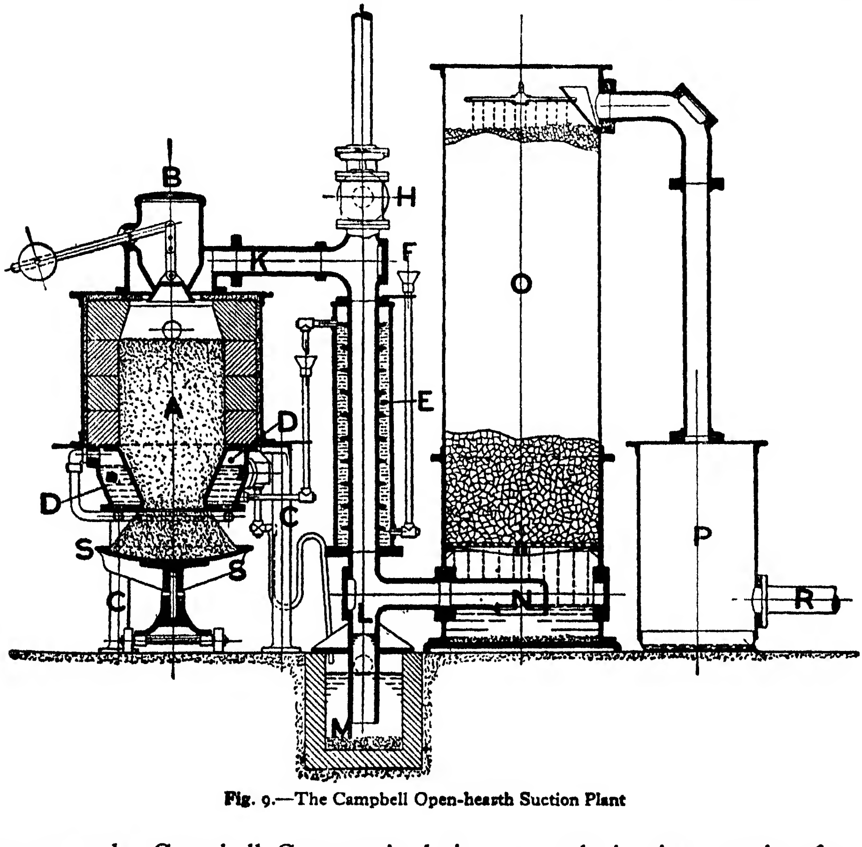


Fig. 9.—The Campbell Open-hearth Suction Plant

illustrates the Campbell Company's design as made in sizes ranging from 50 to 550 b.h.p. The rating is based on the use of anthracite, in pieces from ¼ to ⅝ in. in size, and a consumption of 1 lb. of this fuel per brake horse-power hour at sea level and normal air temperature. The type will successfully gasify anthracite or coke so small as just not to pass a ⅛-in. sieve; also coal and coke " breeze " and locomotive " char ", the rating of the plant being, however, reduced with the two last-named fuels.

The inclusive dimensions in feet of the 50 and 550 b.h.p. plant are approximately as follow:

| Plant. | Length. | Width. | Height. |
|---|---|---|---|
| 50 b.h.p. | 13 | 4·5 | 12 |
| 550 b.h.p. | 33 | 20·0 | 20 |

Referring to fig. 9, the producer A consists of the usual fire-brick lined cylinder of mild-steel plate surmounted by a feed hopper B of the lid and balanced-cone type. The whole is supported from an extension plate by three cast-iron columns CC as shown.

The vaporizer DD in this design is located at the lower end of the producer, and surrounds the hot zone of the fuel; the water is thus effectually vaporized, while the relatively cool walls prevent the adhesion of clinker. The water, supplied at F, receives a preliminary heating in the jacket E of the dust separator L, before passing to the vaporizer; the gases are thus somewhat cooled and some heat is saved.

There is no grate, but the ashes and fuel rest upon a shallow basin-shaped *dead-plate* or hearth S, supported upon a four-wheeled truck, enabling it to be withdrawn quickly when desired. The dead-plate is of such diameter as to contain the base of the *cone of repose* of the ashes as shown.

The steam from the vaporizer passes to a perforated circular pipe surrounding the upper portion of the fuel cone, whence it is drawn, together with the necessary air, into the producer.

The gas leaves the producer by a branch K in the hopper casting, and descends first through the water-jacketed dust separator L, turning sharply at right angles near the bottom and passing through a water-seal N into the usual wet coke scrubber O. The grosser particles of dust and grit fall through the water-sealed bottom of L into the seal-box M, from which they are scooped from time to time as necessary.

Leaving the coke scrubber the gas—if made from clean anthracite—is passed next into the expansion chamber or *gas-box* P, which acts as a reservoir from which the engine takes its supply through R, and reduces the intensity of the suction impulses upon the producer. The waste cock H, which is opened when standing by, as usual, will be noted; also the several cleaning doors provided in the gas main, giving ready access to the interior for the removal of accumulations of dust. The starting fan (not shown in fig. 9), in plants of the open-hearth type, is connected to the gas main beyond the point R, and *draws* air through the producer and cleaning plant in order to create the necessary draught after lighting up. The test cock is also placed beyond R and near the engine.

**The Crossley Open-hearth Producer.**—The Crossley open-hearth producer is illustrated diagrammatically in fig. 10. This design is characterized by a stepped grate, as indicated at AA, which breaks up the fuel cone into several smaller cones, leaving the fire, however, always open and readily accessible to the attendant for stirring, cleaning, and clinkering at all times. The producer is carried by brackets supported on four short stout columns BB. The necessary air is drawn in through the open grate by the engine suction, while steam is supplied all round the fire from the perforated circular steam distributor CC. This steam is raised by the gas yielded by the producer, which is passed through a vaporizer consisting of a nest of gilled tubes surrounding the dust separator and located as at E of fig. 9, but having a much larger and more effective heating surface.

The Crossley rotary charging valve may be noticed; this consists of a hand-operated air-tight hollow cylinder D having a portion of its circumference removed. The fuels used in these open-hearth plants range from clean anthracite and washed gas coke to non-caking bituminous slack and wood refuse.

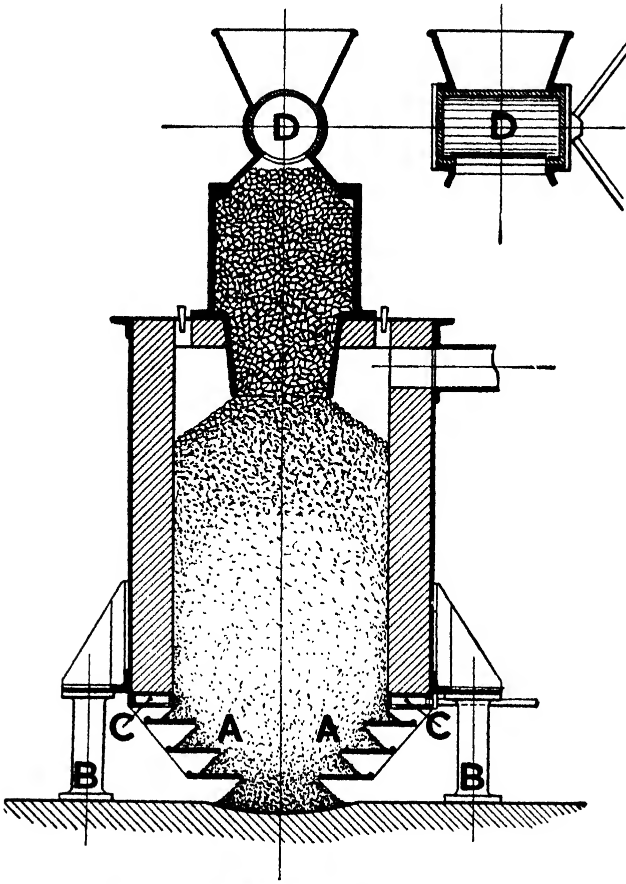


Fig. 10.—The Crossley Open-hearth Suction Producer

**The Suction-pressure Gas Producer.**—The use of the suction producer was for long practically restricted to cases where the gas was used for driving engines, but the advantages of the type are now extended to other cases by the suction-pressure producer, which merely consists of a normal suction plant with the addition of a belt-driven centrifugal exhausting fan between the coke scrubber and the sawdust scrubber or gas drier. The exhauster draws from the producer through the dust separator and coke scrubber and delivers the gas under slight pressure to the sawdust scrubber. In many cases, especially when both engines and burners are to be supplied from the same plant, a regulator is installed between the exhauster and the sawdust scrubber or gas drier, by which the pressure of the gas is maintained constant and its production automatically varied in accordance with the demand (see figs. 1 and 3).

An interesting installation of a Dowson & Mason anthracite suction-pressure producer yielding 28,000 c. ft. per hour of gas of 143·5 B.Th.U. (lower) calorific value is that at Messrs. Taunton's of Birmingham. The gas is drawn from the producer through two coke scrubbers and an oxide purifier by a " booster " which next delivers it at a pressure of 80 in. of water direct to the system of mains; a pressure regulator is fitted whereby gas is by-passed back to the suction side should the pressure at any time rise above the normal. The cost of production with anthracite at 40*s.* per ton, inclusive of depreciation and interest, is roundly 4½*d.* per 1000 c. ft., and 3½ c. ft. of producer gas are required to replace 1 c. ft. of the previously used town gas. Thus the producer gas is the cheaper so long as town gas costs more than 1*s.* 3¾*d.* per 1000 c. ft. The producer plant, using the old town-gas mains, supplies three engines aggregating 150 h.p., 130 tinman's stoves, 16 japanning stoves, lacquering stoves, rings, and other details; and, suitably enriched, may be employed for brazing hearths.

**Bituminous Suction Plants.**—Much attention was devoted for years to the problem of successfully gasifying the very plentiful and cheap low-grade bituminous fuels in suction producers, and plants in which non-caking varieties are utilized have now for some years been built regularly by several firms, as, e.g., Messrs. Dowson, Morton, Crossley, &c.

In most of these the producer is specially designed with a view to burning the tar produced by the distillation of the coal, and so rendering unnecessary the elaborate and expensive cleaning apparatus otherwise required (see, e.g., fig. 3) by providing a yield of gas as tar-free as that from a good anthracite plant.

In many designs this is done by taking the gas from the producer at the hot zone and arranging for both an upward and downward draught simultaneously through the mass of fuel (see fig. 11). The producer is open at the top, through which it is charged with coal, and air constantly passes downwards from above, as indicated by the arrow. In addition, air and steam supplied by the pipe A also pass upwards through the fuel. The gas is taken off from a central belt BB surrounding the zone of incandescent fuel. The upper part of the fire burns downwards; the volatile and tarry matters distil out, and are then converted in the hot zone into fixed gases. The fuel passes slowly downwards towards the grate; such as passes the incandescent zone is coked, and being next met by the upward current of air and steam is converted into ordinary producer gas.

The formation of tar is very largely prevented, and the gas yielded requires practically no more cleaning than when anthracite is used; its calorific value is usually rather less than that of anthracite-made gas, and the fuel required averages about 1¼ lb. per brake horse-power hour. The producer has a water-sealed grate.

The illustration shows clearly the manner in which the mixed air and steam are supplied to the lower part of the producer.

The suction in the producer is sometimes created directly by the engine, but is more usually maintained by an exhausting fan, as described in the paragraph on " The Suction-pressure Producer ", p. 100. After leaving the vaporizer the gas, in Messrs. Dowson's plant, is passed through: (1) a wet wood grid scrubber, (2) a wet coke scrubber of normal type, and (3) a saw-dust or *wood-wool* dry scrubber. It is thence delivered by the exhausting fan either into a governor regulator if there is one engine only, or, if there

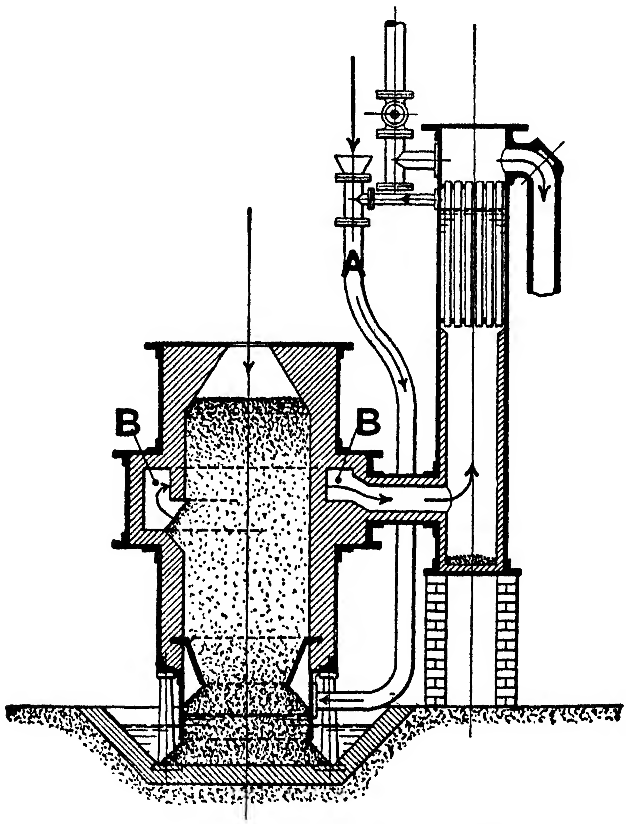


Fig. 11.—Dowson Bituminous Suction Producer

are several engines and perhaps also burners in addition, into a full-sized gas-holder. This type of plant successfully gasifies practically any non-caking or slightly caking bituminous coal containing not more than 30 to 35 per cent of volatile matter; such coal costs in many cases only about one-fourth as much as good anthracite, so that a great reduction in the expense of fuel results from its use. Up to 25 per cent of wood refuse, or sawdust, may be mixed with the coal if desired; and lignite may also be employed alone. Messrs. Dowson build these bituminous suction plants in units ranging from 25 to 800 b.h.p.; the largest type occupies a space $41\frac{1}{2}$ ft. × 31 ft. × 34 ft. in height.

**Gasification of Industrial Wastes.**—An advance of the highest importance has been made in recent years in the completely successful gasification of a great variety of industrial refuses, including, in addition to the low-grade waste coals already referred to, brown coal, waste wood, sawdust, peat, straw, cotton-seed husks, coco-nut shells, coir dust, spent tan, coffee husks, tea prunings, sugar-cane refuse, rubber seed, mealie cobs, olive refuse, almond shells, rice husks, cork dust, fruit stones, sunflower seed husks, dung cake, and dried manure. The production, at exceedingly low cost, of copious quantities of gas for heating, drying, power, and general purposes, using native labour only, and on the sites where the waste materials are actually produced, is an immense boon, and was economically impossible prior to their utilization, as the cost of anthracite or coke delivered at producers so located would, in general, have been prohibitive.

These industrial refuse fuels contain usually a considerable proportion of water, not infrequently amounting to 50 per cent or even more, and accordingly in many cases no vaporizer is required, the fuel itself providing all the steam necessary; fuels containing more than about 50 per cent moisture usually undergo a preliminary drying before use.

On account of their low calorific value and of the considerable variations that are found to occur in their quality and condition, it is necessary, for a stated output, to install producers of larger size than would be required if anthracite or coke were burned; special arrangements of the grate are also sometimes required to suit particular fuels. Some rough average figures for a selection of fuels are given hereunder for comparison.

| Fuel Gasified | Average Consumption in Pounds per B.H.P. Hour. |
|---|---|
| Anthracite .. .. .. .. | 7/8 to 1 1/4 |
| Coke .. .. .. .. .. | 1 ,, 1 1/2 |
| Locomotive smoke-box char .. .. | 1 1/2 ,, 2 1/2 |
| Wood waste and sawdust .. .. | 2 ,, 4 |
| Coco-nut shells .. .. .. .. | 1 1/2 ,, 3 |
| Olive refuse .. .. .. .. | 2 |
| Sugar-cane refuse .. .. .. | 2 to 4 |
| Peat (containing 35 per cent water) .. | 3 |
| Mealie cobs .. .. .. .. | 3 1/2 |
| Rice husks .. .. .. .. | 3 to 4 1/2 |
| Spent tan (up to 50 per cent water) .. | 4 1/2 |

**Water Consumption.**—With high-class fuels about 1 pt. of water per brake horse-power hour is required by the vaporizer; the scrubbers require, in addition, from 1 1/2 to 2 gall. per brake horse-power hour; and if a tar extractor is installed a further small amount is required.

With many refuse fuels, however, the cooling and cleaning water necessary amounts to from 6 to 8 gall. per brake horse-power hour. Water is usually plentiful, but where this is not the case a reservoir tank is installed and the water, cooled and reasonably freed from tar and dirt, is used over and over again, the make-up due to loss by evaporation, &c., amounting in general to from 1 to 2 gall. per brake horse-power hour.

**Waste-fuel Producers.**—In the gasification of these waste fuels no new principle is involved; the producers employed are merely of substantial construction and capacity, and, so far, almost always of the closed-hearth

Fig 12.—Campbell Sawdust Plant at Swindon. External View

suction type; standard plants are constructed by Messrs. Campbell, Cross ley, Ruston-Hornsby, The National Gas Engine Company, The Powe Gas Corporation, and other British firms.

**Sawdust Plant at Swindon.**—Figs. 12 and 13 illustrate respectivel the general external appearance and interior of the fuel loft of a Campbe closed-hearth suction plant installed in the G. W. R. sawmills at Swindo in 1913. The wood refuse, consisting of mixed chips, shavings, and saw dust, is taken from the wood-working machines and delivered by a *cyclon* elevator into the fuel loft, through the floor of which the top of the fee hopper just protrudes, thus enabling a lad to rake it into the hopper at minimum of cost and labour. The height from the ground level to top o feed hopper is 28 ft., and the gas yielded is used to drive a four-cylindered vertical engine giving a maximum output of 350 b.h.p. at 220 r.p.m.

**Portable Plant.**—An interesting design of a very small portable suction power plant is shown in fig. 14; this is built by the National Gas Engine Company and comprises a single-cylindered horizontal engine,

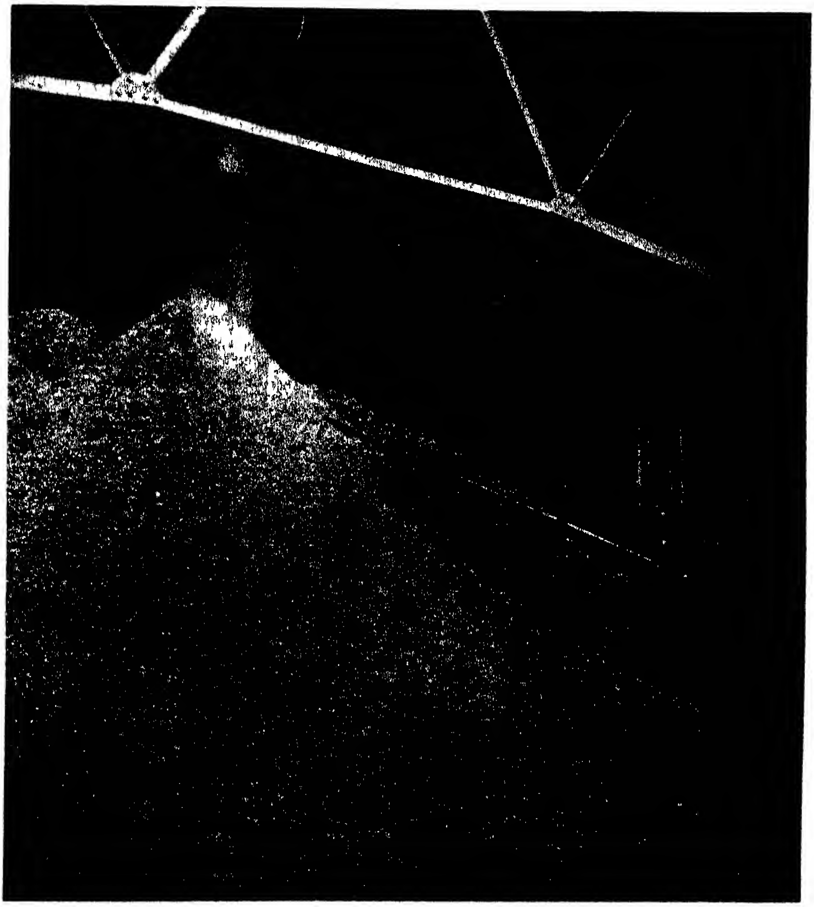

Fig. 13.—Fuel Loft of Campbell Sawdust Plant at Swindon

giving 5 b.h.p. at 400 r.p.m., and a wood-refuse suction plant, all compactly mounted on a small four-wheeled truck. The producer is here a simple tapered cast-iron shell, fire-brick lined at its lower end; the fuel, which may consist of wood refuse, sawdust, coco-nut fibre, cotton-seed husks, and many other carbonaceous refuses, is supplied through the producer top, becoming dried and carbonized as it descends. This small producer is arranged to work with a down draught in order to eliminate the formation of tar as much as possible; air is also delivered through a small pipe terminating in the combustion zone, an additional small supply being also allowed to enter through small holes formed in the producer shell; no vaporizer is necessary, as the fuels employed in general contain sufficient moisture to provide all the steam required.

The fire is cleaned by a hand lever by means of which the grate bars may be shaken, thus causing the ashes to fall into a water-sealed pit. The gas made leaves at the bottom of the producer, and thence travels upwards through a coke scrubber, passing next to a dry scrubber, and thence to the engine. This small unit is stated to be capable of threshing and cleaning 80 bus. of wheat per hour.

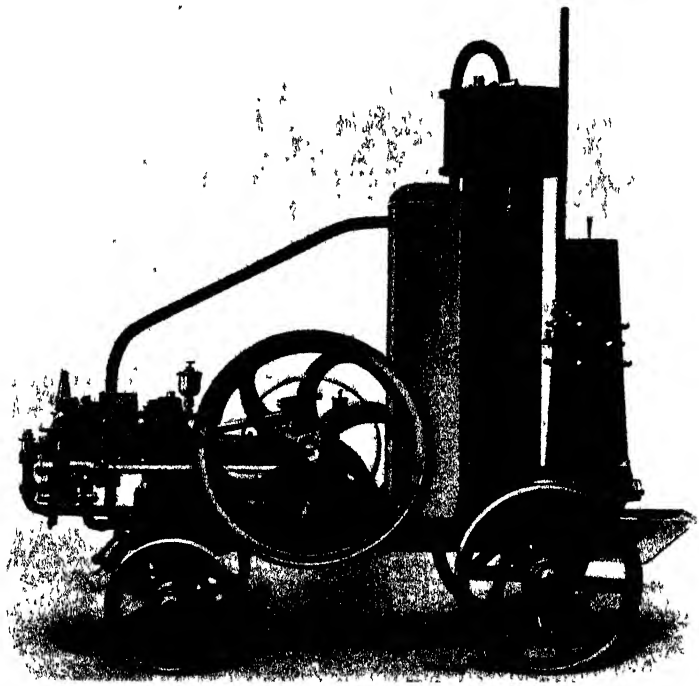

Fig. 14.—"National" Portable Suction Power Plant

**The Lonely Reef Plant.**—A producer plant of considerable interest is that of the Lonely Reef Gold Company of Rhodesia supplied by Messrs. Crossley. This large installation, at an altitude of 4200 ft. above sea-level, replaced a steam plant using 3000 cords of wood per month obtained from the local bush. This large consumption had cleared the bush to a radius of 11 miles from the steam plant, and it was realized that in a very short time the economic limit would have been reached.

The Crossley wood-fuel suction plant reduced the consumption of wood to one-fifth of its previous amount, and effected great economy in operation. The producer plant comprises four closed-hearth suction producers each rated at 350 h.b.p. and capable of an overload to 450 b.h.p., one of which is normally used as a stand-by. As fuel wood blocks about 10 in. dia. × 24 in. long are used.

The gas is employed to drive four horizontal Premier gas-engines, three being of 300 b.h.p. each and one of 225 b.h.p.; the three larger engines are direct-coupled to A.C. generators, while the smaller engine drives an air compressor. A 24-hr. test by Prof. Buchanan furnished the following leading results:—

| | |
|---|---|
| Average moisture in fuel as used .. .. .. | 13·5 per cent |
| Fuel per b.h.p. hour at 80 per cent of maximum load | 2·39 lb. |
| Temperature of cooling water at inlet .. .. | 84° F. |
| Cooling and cleaning water used per b.h.p. hour .. | 4·0 gall. |

Prof. Buchanan reported that the engines ran remarkably smoothly and that the producers possessed capacity sufficient to take the normal load with one held in reserve; also that about 200 gall. of tar were produced by the plant in the 24-hr. test.

**The Tin Dredger "Cambria".**—Another interesting application is that of the Crossley-Premier refuse wood-fuel suction gas plant erected

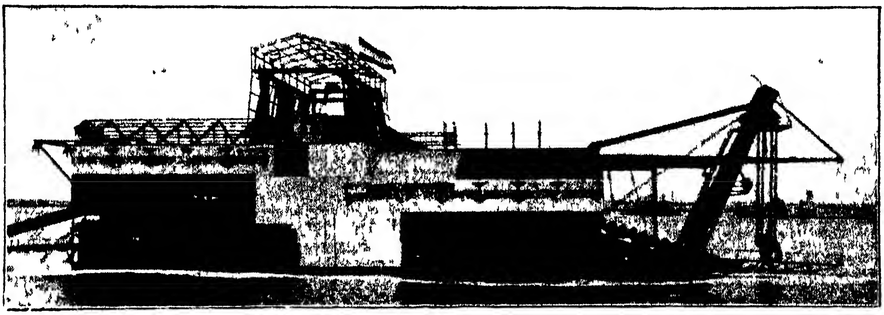

Fig. 15.—Suction Gas Plant operated in the deck of the dredger *Cambria*

on the deck of the tin dredger *Cambria* (fig. 15), owned by the Tavoy Tin Corporation, and operating in Malaya. The gas produced is employed to drive two Crossley-Premier gas-engines, one of 400 b.h.p. and the other of 200 b.h.p.

**Portable Gas Producers for Road Vehicles.**—Much important work has been done in recent years in the direction of driving the engines of heavy motor-lorries by gas supplied by specially designed small suction producers (see below) using anthracite, coke, charcoal, peat, maize cobs, coco-nut shells, refuse wood blocks, &c. &c., as fuel. In England, owing to high labour costs and the lack of a sufficiency of suitable refuse for fuel, the producer-driven lorry has not been much used, but in France and Belgium they are extensively employed, and annual competitions are carried out in those countries under the auspices of their respective governments.

Messrs. Thornycroft built a number in England and as early as 1922 conducted tests at Basingstoke; over an aggregate of 1579 miles the average fuel consumption was only 2·6 lb. of anthracite per vehicle mile. In a special road test made in 1922 with a lorry of 7½ tons gross weight, an average speed of 12 m.p.h. was maintained on a consumption of 2·51 lb. of anthracite and 1·61 lb. of water per vehicle mile. The Thornycroft producer was a very small brick- or corundum-lined cylinder within a square casing which formed a jacket for heating the air on its way to the ash-pan, and served also as a lagging to the apparatus. The bulk of the steam required by the producer was generated in a boiler or " muffle " surrounding the engine exhaust, so that the amount of steam produced was proportioned to the engine output. This boiler worked in conjunction with a water-jacketed pipe terminating in a plain cylindrical dust collector, and by this much of the heat of the gas was removed. From this plain dust collector the gas passed to a " cyclone " dust separator and thence through a cooling pipe led round the chassis frame to a dry " scrubber " packed

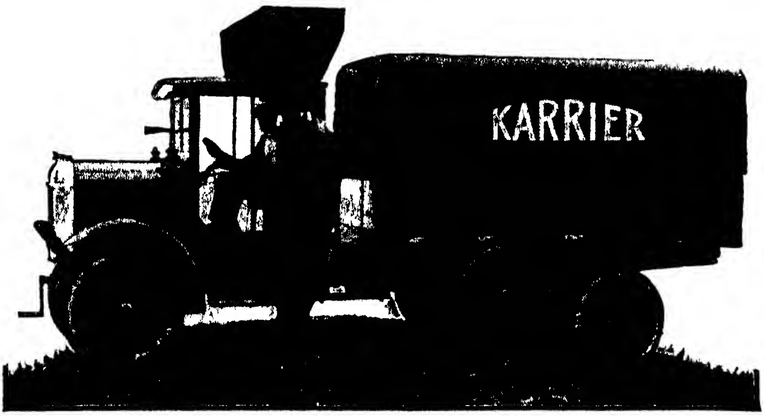


Fig. 16.—Karrier Lorry with Compound Gas Producer

with wood wool, or other suitable material. On exit from this scrubber it was sufficiently clean and cool to be delivered to the engine.

**The Smith Producers.**—Mr. David Smith devoted close attention to the problem of the producer-gas driven lorry for several years, and his small producers have proved entirely successful, using many waste vegetable substances as fuel; many are in service in South America, East Africa, and the Balkan States, tended only by unskilled or native labour; sisal poles, cut into short lengths, have proved particularly suitable as a fuel in East Africa.

As early as 1922 a Caledon lorry carrying a useful load of $2\frac{1}{2}$ tons averaged $15\frac{3}{4}$ m.p.h. on a 25-mile road trial, with a total fuel consumption of 67 lb. of anthracite.

The engines of lorries operated by Compound producer gas are of the ordinary 4-stroke single-acting petrol motor type with no other alteration than an advance of the " spark " on account of the slower explosion of the mixture.

An external view of a Karrier lorry fitted with a Compound gas producer is given in fig. 16. This vehicle, driven by a normal 4-cylinder engine of 5-in. bore × 5-in. stroke, carried a net load of 3¼ tons. The square producer will be seen mounted on the left-hand side of the driver, with a rather large fuel hopper above capable of storing hard-wood fuel blocks sufficient for a 200-mile run.

In fig. 17 is shown a sectional view of the "Compound" suction gas producer using untreated refuse wood and other vegetable wastes as fuel, which is employed on road lorries with complete success. The producer has no moving parts, requires no water (as the fuel contains sufficient moisture), and has but one pipe connection to the engine of the lorry. The scrubber, or gas cleaner, encircles the combustion vessel as shown, and through the annular space between them the air-supply passes down, cooling the gas in the scrubber and becoming itself heated before entering the combustion vessel. The producer is styled "Compound" because the gases are passed through two incandescent zones of fuel, in the main and inner grates respectively, by which all heavy and tarry substances in suspension are reduced, or "cracked", giving a comparatively clean gas which, after passing through the scrubber, is in a fit condition to be delivered to the engine. No hand blower is needed for starting, which is effected by natural draught, and the plant can be brought into operation, from all cold, in from 10 to 15 min.

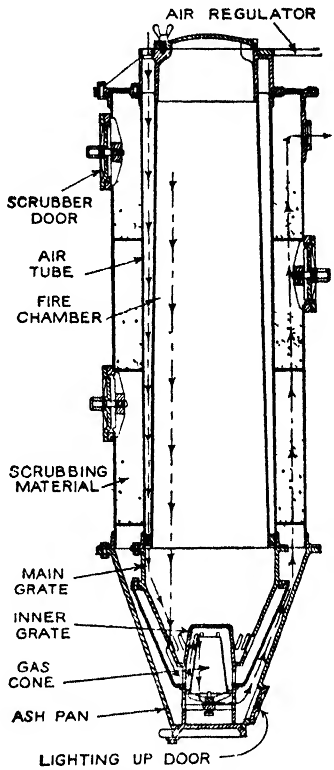


Fig. 17.—Sectional View of the Karrier Producer

The direction of gases formed shown by line of arrows

The composition of the gas as delivered to the engine varies somewhat with different fuels, but an average range, using waste wood, is as follows:

| Constituent. | Percentage by Volume. |
|---|---|
| Nitrogen .. .. .. | 48–52 |
| Hydrogen .. .. .. | 18–15 |
| Carbon monoxide .. .. | 24–20 |
| Marsh gas ($CH_4$) .. .. | 4–4 |
| Carbon dioxide .. .. | 4–8 |
| Oxygen .. .. .. | 2–1 |
| | 100 % |

On ordinary roads a 4-ton lorry requires about 3⅓ lb. per mile of hard wood waste; when using light woods, however, e.g. sisal poles, which have proved an excellent fuel for this purpose, the consumption increases to about 4 lb. per mile.

**Modern Types.**—A number of modern types of vehicle gas producer are described in an extensive paper by Messrs. Bosworth Goldman and N. Clarke Jones read before the Institute of Fuel in December, 1938, and the following illustrations and descriptive matter are taken from this paper by permission.

There are four distinct types of gas producer: the up-draught type, in which the fuel and air move in opposite directions, generally provides the best gas. The down-draught can be used with advantage for dirty or tarry fuels. The cross-draught is the simplest in construction, and relies on a very high temperature in contrast with other types. The double-draught and double-zone types seek to combine the production of good quality gas with the ability to use low-grade fuels. Analysis of gases made from various fuels indicates that the up-draught type of producer yields the best gas.

In Great Britain all types of producer are available, but the up-draught has predominated until recently, when cross-draught types have increased in number under the influence of French design.

Water is used with plants of British design originally developed for charcoal but now able to use low-temperature coke and anthracite equally well. Conversion is the line generally followed both with trucks and tractors, but a new vehicle specially designed for producer gas has been built and its development is proceeding.

Fuels for gas producers preferably should be reactive, of suitable size and grade, with low moisture and ash, and have sufficient volatile constituents. Anthracite has a high volume/calorific value ratio, is not very reactive, and has little volatile matter, ash or moisture. Wood is bulky, but reactive, has high moisture but little ash. Charcoal is less bulky than wood and is highly reactive; its ash, moisture and volatile vary within wide limits unless it is the product of retorts. Low temperature coke is homogeneous and otherwise has good characteristics, like charcoal. Although suitable fuel is available, the lack of distribution of standardized and classified fuels is a problem to be tackled before extended use of this type of vehicle becomes practicable.

The producer plant will include (*a*) Generator, combined with fuel-hopper, (*b*) Coolers, (*c*) Scrubbers and/or filter, (*d*) Air-inlet device, (*e*) Gas-throttle and, in some cases, starting fan and water regulator.

Fig. 18 shows the general lay-out of the Koela up-draught producer. The plant comprises a cylindrical hopper superposed on a combustion chamber. The hopper ends in a spout which feeds fuel automatically into the combustion chamber. This chamber is provided internally in its base with an annular chamber for air distribution, the chamber also serving to support a grate of heat-resisting steel and a refractory liner immediately within the generator shell.

The refractory extends to a position slightly higher than the lowest point reached by the hopper spout. On top of the refractory a baffle plate is arranged, thus forming a gas collecting chamber around part of the hopper spout. The collecting chamber is connected to a preheater, separately mounted on flanges. Through the centre of the preheater passes the air inlet pipe, open at the top and connected at the bottom to the annular air chamber at the bottom of the combustion chamber. The air inlet pipe is fitted on the top with a hand or power pressure fan for starting up purposes, and into this pipe a controlled water-supply is fed when the producer is in operation. This double piping arrangement is an important feature, since it serves the purpose of heating incoming air and vaporizing water

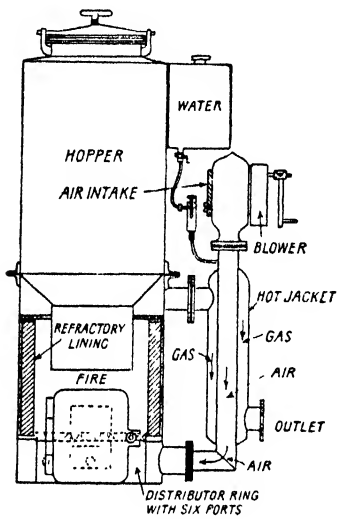


Fig. 18.—Koela Up-draught Producer

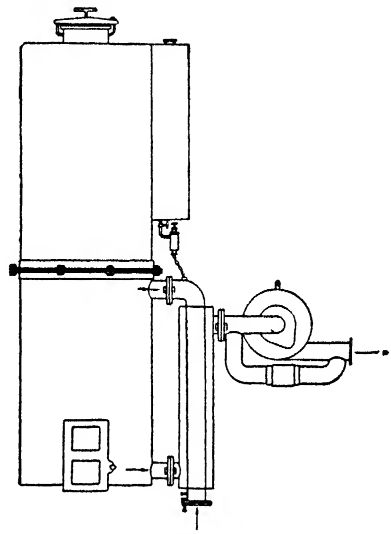

Fig. 19.—Koela Down-draught Producer

while cooling the gas as it leaves the producer. A door for cleaning out the ash is arranged so as to communicate with the ashpit and the combustion chamber above the grate. Provision is made to meet the varying requirements of different fuels by differing grate designs and different fire depths.

Down-draught producers are characterized by their ability to destroy tarry matter and fuel dust, but suffer to some extent from uneven thermal balance. Water is not used to any extent with down-draught producers, but certain manufacturers claim that charcoals containing up to 25 per cent of moisture can be used efficiently. More than 8 per cent is, however, rarely used successfully.

The Koela producer used for down-draught purposes (fig. 19) is substantially the same as the up-draught type (fig. 18). The air heating and gas cooling arrangement is, however, reversed and a suction fan is employed instead of the pressure type. The water-supply is also fed into the top of the air inlet pipe and in counter direction to the air flow, whereas in the up-draught type the air and water flow in the same direction.

Fig. 20 shows the chief features of the H.S.G. producer. The gas exit is located above the air tuyere and direct water injection is carried in the air blast. The combustion zones take pear-shaped forms, the base being at the tuyere. There is not, as might be expected, a tendency for the fire to rise into the fuel hopper. The water flow should be too great rather than too small, otherwise the power falls off appreciably. Any surplus falls into the producer base, where it remains as a reserve against a sudden demand on the producer.

Immediately below the tuyere is a platform which can be lowered by a handwheel outside the circumference of the generator. The platform extends only half-way across the producer and the whole assembly is carried in a drawer, the inner part having a top which still supports the fuel if the drawer is only half withdrawn. Thus it is claimed clinker can be withdrawn during a halt without emptying the remaining fuel from the hopper.

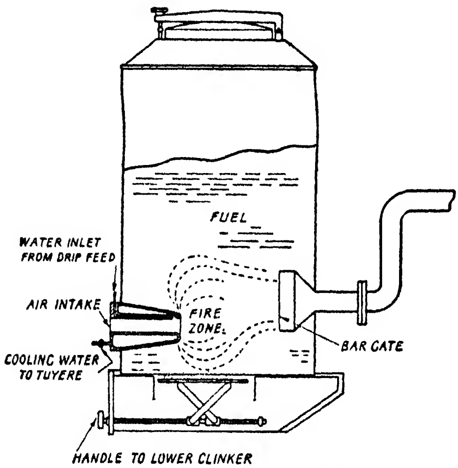

Fig. 20.—H. S. G. Producer

In the Koela double-draught producer the generator consists of a circular combustion chamber lined with refractory material which rests upon an annular chamber (fig. 21). Within this chamber is a slide-grate of which modified forms are available to suit different kinds of fuel. The chamber is provided with slots in the side wall through which air may be admitted for supporting the combustion when up-draught is employed or for gas withdrawal when down-draught is used. A pipe connection is made to the chamber for the passage of air or gas.

A fuel hopper is arranged over the combustion chamber and at the bottom is fitted with an inclined base which is provided with an extension made of a special non-corrosive heat-resisting steel, a narrow passage being left between it and the refractory liner. The tubular portion acts as an upper or auxiliary furnace, and is provided with a controlled air-feed tube passing through the hopper wall. The annulus communicates with a chamber surrounding the inclined base of the fuel hopper, and to this chamber is connected a pipe which serves for air entry when running down-draught or as the gas exit when operating the plant on the combined up- and down-draught method.

The composition and quantity of dust drawn from a producer depends upon the type of producer employed and the fuel used. The gas-cooling and purifying plant used varies with the type and make of producer, but comprises the following units:—

1. An expansion box or first dust separator.
2. One or more cyclone or centrifugal separators.
3. Cooling tubes or boxes.
4. Baffle separators.
5. Filtering and purifying apparatus.

For descriptions of construction and arrangement of various types of the above, reference should be made to Journal No. 63 of The Institute of Fuel.

Tests of Koela producers show that the starting time is 5 to 8 min. for charcoal, 8 to 10 min. for low-temperature coke, and 10 to 12 min. for

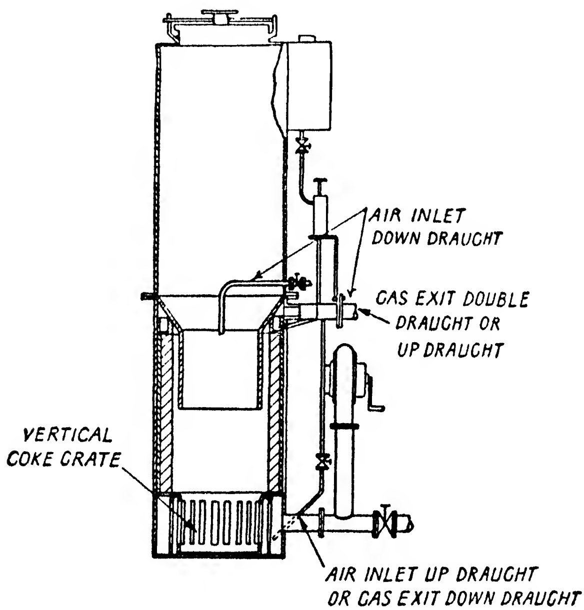


Fig. 21.—Koela Double-draught Producer

anthracite. The calorific values (B.Th.U. per c. ft.) of the gas produced vary from 140 for charcoal to 150 for anthracite. Compared with petrol, the engine needs to be about 20 per cent larger for producer gas, for the same power.

Under war conditions, with consequent shortage of petrol for commercial vehicles, although it is not practicable to fit producers to existing vehicles designed for running on petrol, special trailers fitted with gas producers using home-produced fuel have been fitted as an experiment to a number of omnibuses. The results (November, 1939) are quite encouraging and no doubt other vehicles will be similarly fitted, but obviously such an arrangement can only be a temporary expedient which will be discontinued when normal conditions are re-established. Should, however, the experiment show that the advantages of cheap fuel in this connection are not counterbalanced by disadvantages of weight, space and attention required, together with problems associated with frequent starting-up and acceleration, it is probable that a larger number of vehicles specially designed for producer gas will eventually appear on the roads.

**Conclusion.**—Abundant and cheap heat and power may be regarded as necessities in modern civilized communities, and in this country the chief source of these is coal. Since this is a continuously diminishing asset the importance of measures for its conservation cannot be over-assessed. The burning of raw coal in open domestic fires is well known to be wasteful and inefficient, apart from the question of atmospheric pollution, and is responsible for the waste of millions of tons annually.

The methods of treatment of coal on a large scale which make full use of its potentialities and provide at the same time sufficient fuel for domestic purposes would include: (1) its distillation, yielding coal gas, available for heating, lighting and power, and furnishing also numerous and important by-products, such as coke, retort carbon, sulphate of ammonia, benzol, carbon bi-sulphide, cyanogen and tar, also the *tar derivatives* such as creosote, carbolic acid, naphthalene, anthracene, dyes, &c. (the process of low-temperature carbonization yields a fuel which is more reactive than ordinary coke, and quite suitable for domestic use); (2) the use of coke directly as a smokeless fuel or, by means of producers, to obtain a further quantity of inflammable gas.

The hydrogenation process, in which coal is heated in conjunction with hydrogen in closed retorts under high temperature and pressure, yielding fuel oils and motor spirit, is still largely in the experimental stage, but large plants have been erected for production on a commercial scale, and there is every indication that the process will eventually become financially successful.

Many industrial refuse fuels are also capable of being used and of yielding valuable by-products, and where it is possible to install plants of sufficient size, these may be recovered with profit.

# THE OPERATION OF GAS-ENGINES

BY

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# The Operation of Gas-engines

**Controlling Instruments.**—The operation of a gas-engine plant as far as the production and cleaning of the gas is concerned has been dealt with in a previous section, and the following remarks are chiefly confined to the procedure in the gas-engine house itself. An intelligent man new to the work of running gas-engines will soon realize that the points to be looked to in a gas-engine plant are quite different from what obtains in the case of steam-engines or steam-turbines. The fundamental differences have previously been indicated. If the gas-engine attendant has only had previous experience with steam-engines, he will have to form quite new conceptions of what really happens in the interior of a gas-engine cylinder, and learn that there are more variables to be considered than in the case of steam. He will have to familiarize himself with the difference between blast-furnace gas, coke-oven gas, and producer gas. Generally speaking, the staff in the engine-house has very little to do with the production of the gas used; its function is to deal with the gas after it has been delivered to the engine-room, but it is incumbent upon it to see that the gas is delivered in the best possible condition for use by the engine. But if the gas-engine staff has no direct control over the production of gas, it would at all events protect itself by insisting that the gas be delivered to it up to a standard calorific value. Where gas-producers are used to generate the gas, considerable fluctuations in the calorific value may occur unless the producers are of the self-poking, self-cleaning type. This is sometimes due to the variable nature of the fuel, but more often to want of proper supervision in the poking and cleaning operations. A modern, well-equipped gas-engine house should contain the following instruments, preferably of the recording type: (1) A sensitive gas calorimeter to indicate the calorific value of the gas actually delivered to the engines; (2) A set of water gauges or the equivalents should be provided for showing the pressure of the gas; (3) *Recording tachometers* should be fitted to each engine, and, as a convenience, *revolution-indicators* should also be used, so as to check the revolutions and thus, indirectly, the amount of gas used. Where two-stroke engines are used and there is a liability for some " mixture " to escape with the exhaust, a quick method of sampling the constituents of the exhaust gas should be provided. In addition to these appliances, the man in the engine-room should be able to tell whether the gas contains more than an average amount of tar or other impurity. Tar is the greatest difficulty to be overcome in using producer-gas. You may cool it or wash it and pass it to scrubbers, but it will still come over in the form of attenuated vapour and form cumulative deposits on the valves, cylinders, pistons, and piston-rings, and it will also deposit on the ignition devices and collect in any recesses or salient points in the cylinder or piston, and will often cause pre-ignition. A rough method of estimating the amount of tar present in the gas is to cause a jet of unburnt gas to impinge on a disc of absorbent paper placed at a definite distance from a discharge nozzle. More accurate results can be obtained by the absorption method, in which the gas is passed through cotton-wool. In some cases, a rough idea of the amount of tar present is obtained by burning the gas under a kind of Welsbach mantle, and the operator soon gets to know what the indications mean. The gas-engine operator cannot be too insistent in getting clean gas. With producer-gas, in particular, there has been a great deal of trouble from dirty gas. The failure of the large gas-engine plant at Johannesburg some years ago turned out to be primarily due to the excessive amount of tar in the gas, which was never properly removed from it, and came over and broke the mechanism actuating the gas pumps. Some of the gas-engine makers stipulate that the gas must not contain more than a specified amount of sulphur, but the author has never been able to trace that this is a very serious matter, and where this is really of importance, *oxide-scrubbers* can be used as in gas-works.

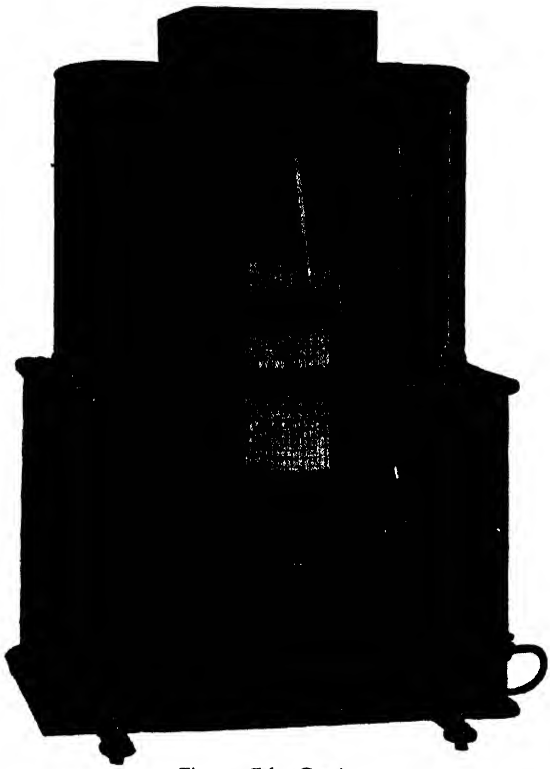


Fig. 1.—Calor-Graph

The calorific value of the gas may be obtained in two ways, either by analysis of its constituents or by burning a known quantity of the gas at a uniform pressure and measuring the heat produced. The *method of analysis* should be used from time to time to check the calorimeter. There are several different kinds of reliable and accurate calorimeters on the market.

Most of the continuous types of instrument are arranged to record the heat imparted to a known volume of water by the combustion of a measured quantity of gas. This type requires a certain amount of attention in connection with the flow of water and variations in the pressure of gas. Messrs. Alexander Wright & Co. have introduced a new design which is made under Simmance & Abady's Patents. The design is termed the *Calor-Graph.* A measured quantity of gas is burnt and the heat generated is observed by means of a differential thermometer, one member of which is exposed to the ordinary temperature while the other member is acted upon by the products of combustion. These thermometers are in the form of vessels with metal diaphragms, and are suspended at opposite ends of a balanced beam, and variations are shown on the chart by means of an ordinary recording pen. If the instrument is placed in a small cupboard in which the temperature is kept constant, the Calor-Graph gives the absolute calorific value of the gas and is self-regulating as far as variations in pressure are concerned. An illustration of one of these instruments is shown in fig. 1. Another apparatus extensively used is the *Sarco Recording Calorimeter.* This instrument is illustrated in fig. 2, and the principle is that the gas is burned in a well-radiated chimney, and the working of the instrument depends upon the fact that the average rise in temperature of this chimney is proportionate to the heat developed by the flame. This chimney contains an annular chamber which forms one limb of a U-tube, the other limb being a simple tube maintained at atmospheric temperature. The U-tube is filled with oil, and as both limbs terminate in tanks, it is possible to record the differences in level between the oil in the two elements, as the expansion of the oil due to the heat applied takes place on one side only. A very ingenious device for maintaining pressure and uniform flow of the gas is used in this instrument.

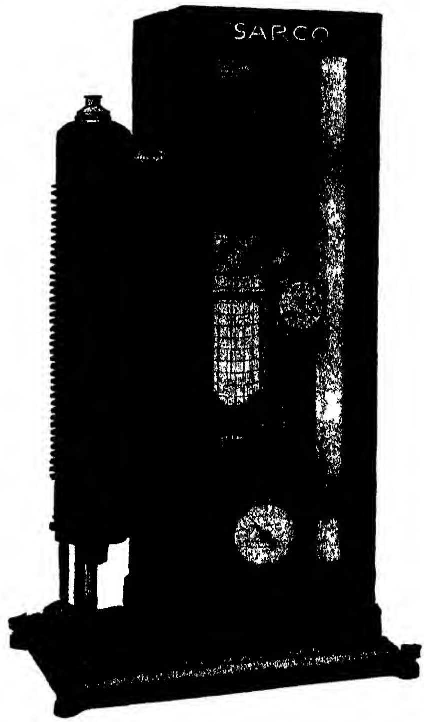


Fig. 2.—The Sarco Recording Gas Calorimeter

A calorimeter to be of much use in the engine-house should be placed where it is under the constant observation of the attendant, but in arranging this a difficulty is often met with. The connections leading to the instrument itself are of comparatively small size compared to the pipes supplying the engine, and therefore a throttling action takes place which causes a sort of time-lag, and the attendant only gets to know of any change in the value of the gas after it is too late to adjust the engine to meet it. Apparently the only way to obviate this is to place the calorimeter in the large main of the gas-supply, and transmit any changes of value to the engine-room by some sort of electrical indicator.

**Starting Up Engines.**—Various methods have been proposed and put into practice for starting up gas-engines. In small engines, petrol injected at the back of the piston with the exhaust-valve temporarily arranged for half-compression was at one time largely used. In larger engines, coupled to continuous-current generators, the engine can be started by making the generator act as a motor; but extra switch-gear is required, and nowadays it is almost universal to start with compressed air. Most of the compressed-air installations are far too small for the job, or rather they have not sufficient storage capacity. A gas-engine may start off at the first effort, or it may be necessary to make a number of attempts, and, in that case, after several tries it will probably be found that the pressure in the reservoirs has fallen so low that the whole system will have to be pumped up again, involving a very considerable loss of time. It is the custom now to use compressed air at about 300 lb. per square inch. The valves admitting the compressed air to the cylinders are generally worked off the shaft which actuates the inlet and exhaust valves, and are arranged to come out of action automatically as soon as the engine gets into normal running. Attention should be given to these air valves to make sure they " bed " properly on their seats; otherwise they may cause pre-ignition. Before attempting to turn on the compressed air, the engine should be moved into the most suitable position for starting, either by means of its own barring gear, or by means of a sling connected to the overhead traveller, if there is one available. After standing for any length of time the engine should be barred round for at least two revolutions before turning the compressed air on.

**Ignition Plugs.**—Prior to starting up, the ignition plugs and contacts should be examined and cleaned. If the sparking plug is of the jump type, like the Lodge, the working condition of this can be tested by inserting it in a closed vessel, where its behaviour can be observed either through a thick toughened glass window, or through mica. Where the ignition is derived from accumulators the engine generally starts up without trouble, whatever the exact sparking arrangement is; but where the spark or arc is obtained directly from magnetos, it may be necessary to flick the armature once or twice by hand until the magneto attains its normal speed. Where battery-ignition is used, the current is generally distributed to the various plugs by a kind of revolving commutator with adjustable brushes, so that the actual point of ignition can be varied. Care should be taken to keep this distributing mechanism very clean, as any short-circuiting may lead to considerable trouble, and slow the engine down so that it gets out of step or stops altogether. Where the engine is large enough to justify the expense, it is a matter of precaution to fit two independent systems of ignition, one being worked from a battery, while the other plugs are directly connected to magnetos. In considering the operation of a gas-engine, it may be truthfully stated that ignition is the vital spark. The nature of the ignition apparatus is fixed by the engine-maker, and all the attendant can do is to adjust it to the best advantage. Various views are held as to the desirability of producing a fat spark, but the subject of electric ignition is really a very complex one. A most instructive paper read before the Institution of Mechanical Engineers by Mr. J. D. Morgan was published in January, 1922, and a list of very useful references is given at the end of the paper. Vols. 17, 20, 21, 22 and 23 of the *I. A. E. Proceedings* should also be consulted.

**Water-cooling Circuits.**—In large gas-engines, where the piston and piston-rods have to be water-cooled, the supply of water for this purpose is kept distinct from the supply to the jackets, cylinder covers, and exhaust valves—where these are water-cooled. In a large Nurnberg engine, the cooling water enters at the centre of the cylinder jacket on the bottom side, and discharges from both ends of the jacket into the bottom of each cylinder cover, the final discharge being from the top of each cover into a common discharge pipe. The cooling water reaches the jackets at a pressure of 45 lb. per square inch, and is generally fed from an overhead tank which is kept pumped up. The pistons themselves are supplied by an independent pump giving a pressure from 85 to 100 lb. per square inch, this pressure being necessary to overcome the inertia of the water in the reciprocating pistons. The most usual way to convey the water to and from the pistons themselves is to make the piston-rods annular, and use jointed pipes attached to the middle slipper. The rocking joints of these may give some trouble, and if any water escapes on the fine threads at the junction, corrosion will take place. In some engines this water is led to the pistons by pipes working over one another like a trombone. The discharge water-pipes are turned into a common tun-dish where they are all visible. The cooling water should leave the engine between 105° and 115° F., but care should be taken that the piston-rods are not cooled down to a point where sweating begins, due to deposits of moisture from the atmosphere, otherwise wasting away as a result of corrosion follows.

**Stopping the Engine.**—When it is desired to stop the engine the following procedure should be followed.

The main gas valve must first be closed and then the separate cut-off valves on the manifolds. Unless there is any obvious reason for stopping the engine as quickly as possible, the ignition circuit should not be interrupted, and the ignition plugs should never be tested by short-circuiting them, since a charge of unburnt gas is thus discharged into the exhaust pipe and silencer. The hot exhaust gases from a succeeding charge will ignite this, and serious explosions have resulted. All gas-engines require a general examination about every nine months, and with dirty gas more frequently. When this examination is made the alignment of the brasses should be looked into and compensated for. In an engine with three bearings it is somewhat difficult to see that the whole three wear down equally.

**Lubrication.**—It is obvious that the lubrication of a gas-engine must be continuous and reliable, as any interruption may cause serious damage. Different qualities of oil are generally used for the cylinders and exterior parts (like the main bearings, crank-pins, crosshead, and slippers). This latter supply is usually taken from an overhead tank, and after doing its work, and being passed through a cooler and oil filter, is then pumped back to the supply tank, fresh oil being added from time to time, and the sludge removed from the filter. The whole of this oil ought to be changed periodically. For the lubrication of the cylinders, the stuffing-boxes, and the valve spindles, *screw-down lubricators of the Mollerup type* are generally used, and a group of these is shown in fig. 3. These are worked by a ratchet movement derived from the lay shaft, and, as long as they are supplied with oil, rarely give any trouble. Certain small parts of the gas-engine are lubricated with grease, the feed being regulated by hand, and, in some inaccessible parts, by hand oil-feeding. In gas-engines of the vertical type, the oil is conveyed under pressure to the main bearings, and to the big ends and gudgeon-pins by copper pipes attached to the connecting-rods, and as these pipes are liable to break and come adrift a sensitive pressure gauge should be placed close to the oil pump, and a strict watch be kept to see that the oil pressure is equally maintained through the system. In vertical engines the oil pumps are generally of the valveless type. Where the vertical type of engine is used, the crank-case is usually closed in, and the lubrication is partly on the splash system,

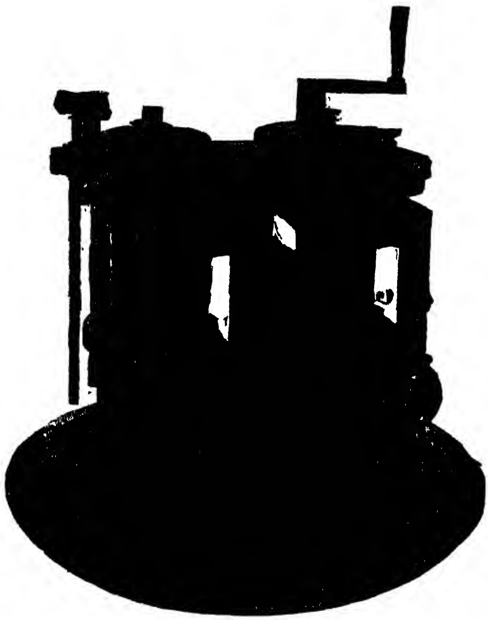


Fig. 3.—Mollerup Plunger Lubricator

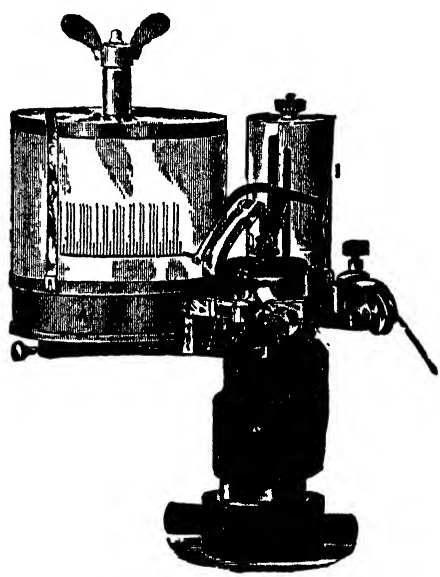


Fig. 4.—Dobbie-M'Innes Indicator

and partly under pressure. The continuous agitation of the oil in the bottom of the crank-case produces a kind of mist which is liable to explode if any flame passes downwards past the piston rings. The author has known quite serious explosions arise from this cause. It is now the custom to fit relief devices in the crank-case, usually in the form of fibre discs which yield if an explosion does take place, but most engines with enclosed crank-cases are now fitted with mechanically driven fans, which draw off the oil vapour and deposit it outside, or some of it may be returned to the manifold where the gas enters the engine. For further information on the lubrication of running gear and cylinders, see " The Operation of Oil-Engines (p. 304).

The tachometer or speed indicator should be lubricated with pure neat's-foot oil free from any acid, and, if the engine has been standing any time, should be cleaned with petrol or benzene before starting up. The tachometer should be checked by counting the actual revolutions of the engine.

**Indicators.**—To know whether the engine is working to the best advantage, the attendant will have to take into account the quality of the gas as indicated by the calorimeter, and where possible to ascertain the amount of gas consumed by each individual engine, and, moreover, indicator diagrams should be taken frequently. Messrs. Dobbie, M'Innes & Clyde, Ltd., of Glasgow, have specialized in indicators for internal-combustion engines, and, in addition to their well-known "Dobbie-M'Innes" external-spring parallel-motion gear type of indicator for such work, are also makers of continuous explosion and pressure recorders and the Hopkinson flash-light engine indicator. This latter apparatus is valuable for very high-speed engines, automobiles, or aeroplanes, but need not be described in detail here.

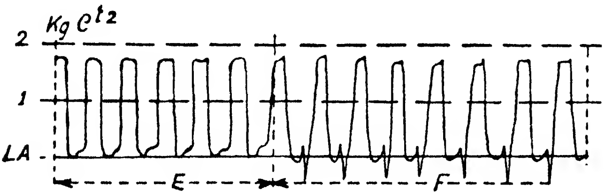
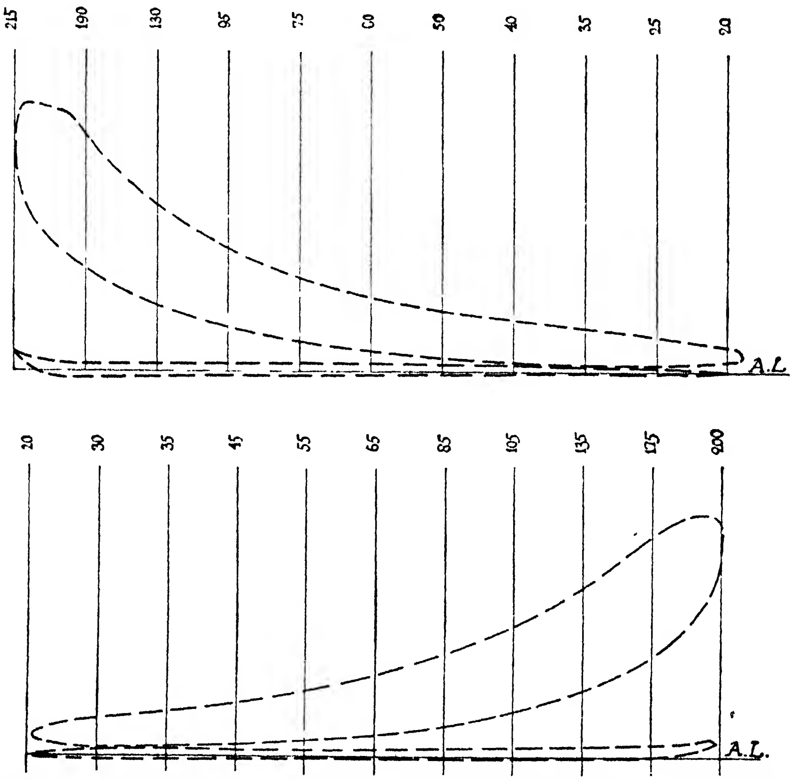



Fig. 5.—Diagram showing the rate of compression and resistance to inlet and exhaust. This diagram was taken on a motor-car engine driven by a dynamo running at a speed of 1500 revs. It can easily be seen that the resistance to inlet and exhaust is on the increase, while the total amount of compression is diminished.

The most serviceable form for use in the gas-engine house is the Dobbie-M'Innes modification of the Mathot apparatus. This is illustrated in fig. 4. It will be seen there are two drums, one of which, marked A, continuously rotates and keeps a record of the sequence of explosions, while the other drum, I, permits diagrams to be taken of the working cycle. Fig. 5 shows a diagram taken with such a recording instrument. For gas-engine work, the spring is generally placed outside the cylinder of the indicator, where it is only subjected to atmospheric temperature. Fig. 6 shows diagrams from a four-stroke gas-engine. Fig. 7 shows light-spring diagrams " stopped off ". This stopping off enables the suction stroke and the toe of the exhaust to be better studied. Fig. 8 shows a diagram from a two-stroke Körting engine. In each of these cases the engine is working on producer gas, and a high compression pressure is used in order to ensure rapid combustion.

Although anyone can soon learn to take a diagram, it requires a great deal of experience and judgment to draw the right conclusions from it. There are certain elements of error in every indicator, such as the friction of the piston, the inertia of the moving parts, and possible stretching of the cord connections, but a practical operator knows how to allow for these.

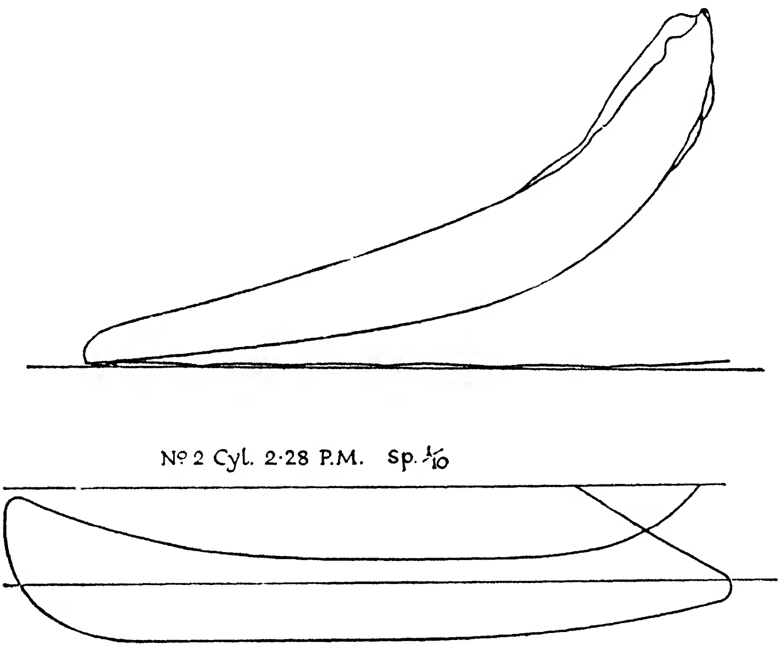



Note.—Engines are 4-Cycle, 2-Cylinder Tandem. Diameter of Cylinder, 4′ $1\frac{7}{16}$″. Stroke, 4′ $3\frac{5}{32}$″. Normal Speed, 94 R.P.M. Piston Rod Diameter, 1′ 0″. Spring, $\frac{1}{160}$.

Fig. 6.—Indicator Diagrams, Nurnberg Engine No. 252

Great care should be taken that there is no lost motion in connection with the reducing gear, as the position of the indicator piston must synchronize with the position of the main piston. Care must be taken to see that the cord from the reducing gear is led off in a direction parallel to the axis of the engine, a guide pulley being used if necessary to take the cord to the indicator drum. If this precaution is not observed a distorted indicator diagram will result and wrong conclusions will be drawn. The author has seen cases of this kind where correction of the reducing gear has produced a perfectly normal diagram. The Dobbie-M'Innes instruments can be fitted with an electrical attachment by which diagrams can be taken simultaneously from both ends of the cylinder.

The technical head of a gas-engine station should satisfy himself as to suitability of the indicator " rig up ", and closely examine the diagrams himself. In two-stroke engines where the air- and gas-charging pumps are used, the " negative work " absorbed by these should be deducted from the work manifested in the main cylinder diagram. The *Okill pressure indicator* is a useful appliance for testing compression and explosive pressures.



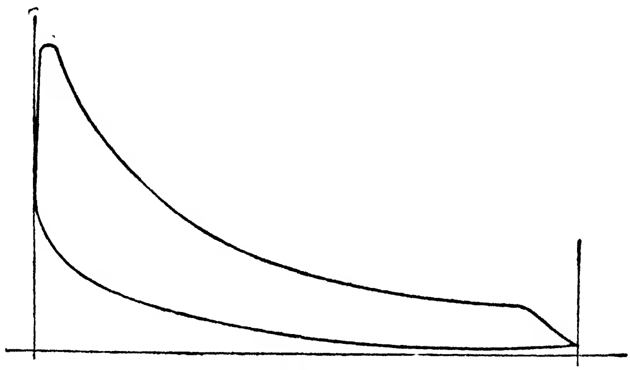

Engines are 4-Cycle, 2-Cylinder Tandem. Diameter of Cylinder, 4′ $1\frac{7}{16}$″. Stroke, 4′ $3\frac{5}{32}$″. Normal Speed, 94 R.P.M.

Fig. 7.—Indicator Diagrams, Nurnberg Engines Nos. 251-253

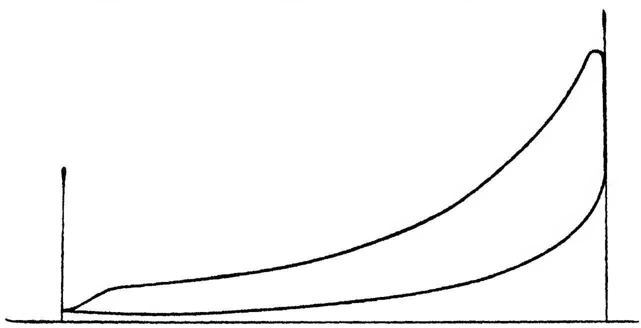
Care should be taken that when the indicators are removed the connections are stopped up by plugs, otherwise pockets will be left and pre-ignition may occur.

Directly after the indicators have been used, they should be taken apart and any tar or deposit removed by means of some solvent. The indicator is to the engineer what the stethoscope is to the surgeon. It does not show everything, but it can be made to indicate whether the inlet and exhaust valves are acting properly, and the influence of compression, temperature speed, and time of ignition, and the best proportions of air and gas.

**Gas Consumption.**—The amount of gas used for each individual engine is not very easily determined, as the pulsations during the suction strokes tend to upset the readings of the meter, and the only real way to arrive at the actual consumption is to have a *calibrated gas-holder* which can be connected to any desired engine. This may be made to contain a supply of gas to last half an hour or an hour, or longer, and does really give the value consumed. If the cubic contents of the cylinders are known and the revolutions per minute counted, and if the indicator-diagrams show that the pressures are normal, it can be judged fairly closely how much gas the engines are



Körting No. 1 (back)
Spring, $\frac{1}{300}$. Average Pressure, 80·6 lb. per square inch. Amps. 2200, volts 201. R.P.M., 85.



Korting No. 1 (front)
Spring, $\frac{1}{200}$. Average Pressure, 69·5 lb. per square inch. Amps. 2200, volts 201. R.P.M., 85.

Fig. 8.—Indicator Diagrams from 200 B.H.P. Körting Horizontal Gas-Engine

using; but generally the engineer in the power-house is content to take the readings of the main or *bulk-meter* and divide this reading by the number of engines running. In gas-engines up to a moderate size, it is the custom in many cases to use gas-bags or similar anti-pulsating devices, but these are liable to be damaged if the engine backfires, and in most modern layouts attempts are made to meet the pulsation difficulty by making the gas pipes of ample size. Where gas-engines are used to drive continuous-current generators, the attendant can generally regulate the revolutions so that peaks in the speed variation do not synchronize with different engines. With a three-phase generator, the matter becomes more difficult and considerable fly-wheel effect is necessary.

**Air for Mixture.**—The intake for the air to make the mixture should be arranged so that it gets the air as clean as possible, because dirty gas is bad enough, but dirty air with dirty gas may give a lot of trouble. In large engines, some sort of air-filter is generally used, but it must not be allowed to get choked up. Water-washers are sometimes employed, but although from one point of view they are advantageous in increasing the density of the charge and cleaning the air, it is very difficult to prevent the washer-water being carried over into the cylinder. The same general remarks apply to the operation of two-cycle engines as with four-cycle engines. In the two-cycle Körting type, the attendant has the gas and air pumps to look after in addition to the working cylinder proper, and he will have to see that the by-passing arrangements controlled by the governor are properly set; but if the gas is reasonably free from tar and without any gross impurities, the charging pumps give but little trouble.

**Valve- and Governor-gear.**—It is essential that the valve-gear and the governor-control of gas-engines should be kept in perfect order. Where the valves are worked by eccentrics, the wear is less than with cams and rollers, but it is transferred from the cams to the rocking levers. When the engine is quite new the makers should be asked to furnish hardened steel templates of either the cams or the rocking levers, and when the engine can be stopped for inspection these templates should be applied to show the actual wear, which, if appreciable, should be compensated for.

The number of men it is necessary or desirable to employ in a gas-engine house depends upon several things, e.g. the size of the engines, whether they are horizontal slow-speed engines of the Nurnberg type, or whether they run at a higher speed like the National or other designs; and a good deal depends upon the general layout of the plant, which in turn is generally determined by the space available for the building. For horizontal engines above 1000 h.p. it is usual to employ one driver to each engine, and have a somewhat superior type of cleaner who is competent to temporarily relieve the driver if required, but whose regular work is to look after the basement plant, the circulating water pump, the behaviour of the exhaust valves and the exhaust steam-boilers. In vertical engines of tandem type there are generally two platforms, one to each line of cylinders with connecting gang-ways between, and as the control is all concentrated at the floor, one driver can look after two engines, and the cleaner can promenade the upper plat-forms and occasionally look at the exhaust boilers where fitted. Care should be taken to have the gas-engine house well ventilated. Gases will escape, and although perhaps not in dangerous quantities, they reduce the efficiency of the men.

It is to be hoped that the attendant who is taking up the running of gas-engines will not be deterred from doing so by the apparent number of difficult points that have to be dealt with. The writer has known many men brought up with steam-engines who, in a very short time, grasped the different conditions obtaining in a gas-engine house, and most of them seem to take a pride in being able to adapt themselves to the new conditions. In conclusion, it should be said that the successful working of the gas-engine station depends to a considerable extent upon how far the management affords the attendants opportunity of acquiring the special technical knowledge. Men are as a rule eager to learn, and the expenditure of, say, £20 in books for a small circulating technical library would generally be amply repaid in increased output and freedom from breakdowns. An occasional lecture from the chief engineer would also have a stimulating effect, and interchange of visits between various installations increases everyone's knowledge. Working models and explanatory diagrams should also be available, and it is to the employer's interest that the operator should become thoroughly competent at his job, and he will hardly do this by unaided efforts. The internal-combustion engine, in one form or other, has a wide future before it, and the student will find that a thorough knowledge of the subject will prove a very good investment.

# OIL-ENGINES

## (Including Marine Oil-engines)

BY

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# Oil-engines

## (Including Marine Oil-engines)

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## CHAPTER I

## Intioductory

**De Rochas and Otto. the Four-stroke Cycle.**—The amazing development in the use of the internal-combustion engine during the past half-century received its " starting impulse " by the enunciation in 1862 of the essentials of successful working by M. Alphonse Beau de Rochas, who first clearly laid down the sequence of operations now universally known as the " four-stroke cycle ", comprising the suction, compression, expansion, and exhaust of the working charge within one cylinder.

Fourteen years later, in 1876, the de Rochas cycle was realized in the famous " Otto Silent Gas-engine ", which first placed the internal-combustion engine on an economic footing, and thus inaugurated the modern world-revolution in prime movers.

The earliest Otto " Silent " gas-engines were of the horizontal single-cylinder single-acting four-stroke type illustrated diagrammatically in fig. 1. Within an open-ended cast-iron cylinder AA slides a gas-tight piston B driving a crank-shaft C through a connecting-rod D. When the piston is in its innermost position, as indicated by dotted lines, a volume K, termed the " combustion chamber " remains, and in this are fitted valves E and F and an ignition device H. In the figure the piston is making its first out-stroke, or suction stroke; the inlet valve E is open, and through it fresh charge is entering the cylinder. At the end of this stroke the inlet valve is closed, and the return of the piston compresses the charge into the combustion chamber K; ignition is then made by H, explosion instantly results, and the piston is driven forcibly outwards, thus performing the expansion or working stroke, at the end of which the exhaust valve F is opened. The piston finally returns, expelling the remaining exhaust gases through F. This cycle is continuously repeated during the running of the engine.

In the four-stroke cycle engine there is but one working stroke of the piston in every four, and this infrequency of working impulses involved cumbrous and heavy designs. Accordingly the attention of engineers was

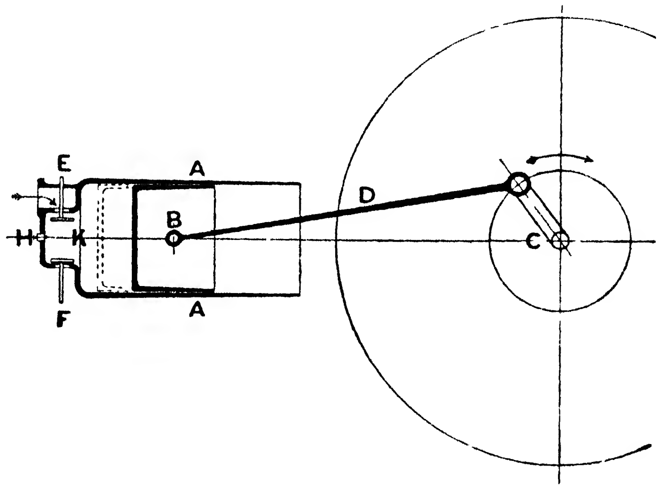


Fig. 1

soon directed to the problem of increasing the impulse frequency; this of course, can most obviously be done by increasing the number of cylinders acting upon the same crank-shaft, and in modern designs of large internal-combustion engines as many as eight cylinders actuating one crank-shaft are not uncommon.

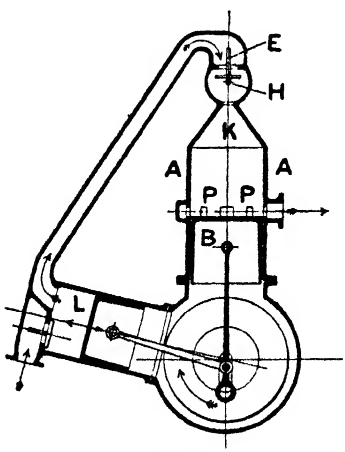


Fig. 2

**The Clerk Two-stroke Cycle.**—A second means of increasing the impulse-frequency is by modifying the working cycle in each cylinder, and a fundamentally important advance in this direction was effected by Sir D. Clerk, who exhibited at the Paris Exhibition of 1881 the first single-cylindered internal-combustion engine giving onc working impulse in every revolution of the crank-shaft. This type, operating on the famous "Clerk" or "Two-stroke" cycle, is illustrated by the diagram fig. 2, the general arrangement of which follows that of the modern marine "Dolphin" Clerk-cycle engine, and comprises a working cylinder AA and piston B which, near the end of its outstroke, overruns a belt of ports PP, thus permitting the escape of the burnt gases into the atmosphere. A charging-pump L, driven from a pin on the connecting-rod as indicated, delivers the working charge through an automatic inlet valve E while the piston is near the end of the out-stroke, its subsequent return compressing the charge into the combustion chamber K, when it is fired by the plug H, causing the working stroke to be described. Thus in this engine every down stroke of the piston is a working stroke, and the impulse-frequency is twice as great as with a four-stroke cycle. The conical combustion chamber facilitates the expulsion, through the ports PP, of the burnt gases by the fresh charge with a minimum of commingling, and the spherical top ensures a rich and readily explosive mixture at the ignition plug, so ensuring regular firing.

**The Daimler Fast-running Engine.**—A third method of increasing the power output of a given engine is by increasing the revolution speed. The early gas-engines were all of the horizontal stationary single-cylindered, or occasionally two-cylindered, four-stroke single-acting type, running at not more than about 200 r.p.m.; they were thus necessarily bulky and heavy in relation to their power output, as much as 1100 lb. per b.h.p. being a not uncommon figure.

To Gottlieb Daimler (1834–1900) is mainly due the credit of the invention of the small high-speed internal-combustion engine which has so profoundly modified the conditions of modern life through the consequent development of the road automobile, the aeroplane, and of small, light, high-speed water craft.

Daimler retired from the Deutz Gas Engine Company in 1882 in order to devote his attention to the invention of a practicable small quick-revolution internal-combustion engine, and as early as 1886 a small Daimler motor was fitted to a bicycle, while in 1887 a small car was similarly fitted; it is of peculiar interest to note, however, that Daimler formed the opinion that his little engines were unsuited to road vehicles, and accordingly he turned his attention to the propulsion by their means of launches and canal boats, and in this service they have been, and continue to be, largely employed.

By 1883 Daimler's engines were running at 800 r.p.m., and in the hands of the motor-car pioneers, notably Messrs. de Dion, this was soon more than doubled. Daimler's 1883 engine was of the familiar hand-started vertical single-acting four-stroke type with enclosed crank-chamber and "splash" lubrication. The valves were in a pocket at the side of the cylinder, the automatic inlet being located vertically above the cam-operated exhause: ignition was by hot tube.

The problem of finding a suitable fuel gave considerable trouble, and Daimler experimented with many liquids before finally deciding upon "light petroleum spirit", i.e. "petrol", as the best; he used a "surface" carburettor in which, during suction, warmed air was caused to pass through a constant thickness of liquid petrol.

**Day's Two-stroke Engine.**—In 1891 Day effected a very important simplification in the small quick-revolution two-stroke motor by dispensing altogether with the separate charging-pump L (fig. 2) of the Clerk engine, and utilizing the enclosed crank-chamber and piston of the motor itself to effect the charging. The Day engine is illustrated in fig. 3; there is the usual cylinder A, with piston, connecting-rod, and crank-shaft; the crank-chamber is enclosed and gas-tight. Assume the engine running, and that the downward stroke is just completed; when near the bottom of the stroke the piston first uncovers a belt of exhaust ports B, thus allowing the escape of the burnt gases, and immediately afterwards overruns a belt of inlet ports C, thus permitting a fresh charge to flow from the crank-chamber M into the cylinder, as indicated; this fresh charge is caused to enter the crank-chamber through an automatic valve E during the previous up-stroke of the piston creating a partial vacuum in it, while the subsequent

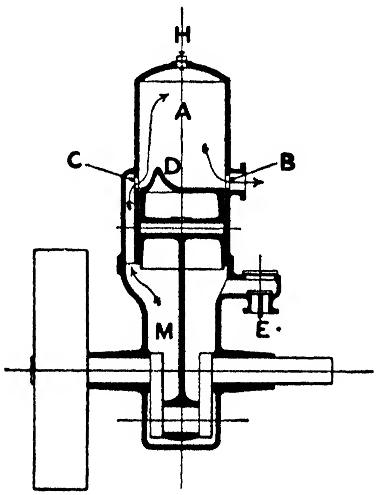

Fig. 3

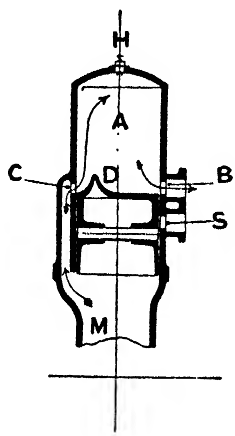

Fig. 4

descent of the piston compresses the charge to some 3 or 4 lb. per square inch, and thus causes it to rush into, and fill, the upper part of the cylinder immediately the port C opens. A "hump" D on the piston deflects the entering charge upwards, thus minimizing loss by short-circuiting straight across to the exhaust port B. The piston next ascending first cuts off the ports B and C and then simultaneously compresses the entrapped mixture into the combustion head, and sucks a fresh charge into the crank-chamber below; when the piston reaches the top of its stroke the compressed charge is fired by the ignition plug H, and the working stroke is then described.

Thus every down stroke is a working stroke, and in this engine only one valve, and that of the automatic type, is needed. This engine is known as the "two-port" type, while fig. 4 shows a still further simplification in which the motor becomes entirely valveless, and of the greatest possible simplicity, the only moving parts being the piston, connecting-rod, and crank-shaft. This is known as the "three-port" type, and the valve E is eliminated by providing a third port S which is overrun by the *lower* edge of the piston when nearing the *top* of its stroke, its ascent prior to this overrunning creating a partial vacuum in the crank-chamber, so that as soon as s is opened fresh mixture rushes in and fills it.

The reader will note that both these two-port and three-port type engines will run equally well in whichever direction they may be started, and this is a valuable property in marine service.

**Liquid Fuels:** *Petrol.*—The volatility and extreme inflammability of petrol necessitate careful precautions when used in watercraft, and in the earlier applications of the internal-combustion engine to boat propulsion fires and explosions were disconcertingly frequent. Experience has, however, taught that when suitably designed and carefully used, petrol-driven boat engines are reasonably safe, and this liquid fuel is now largely used in craft varying from the smallest outboard-engined boats up to such considerable vessels as e.g. the 65-ft. cruising yacht *Frefada* (by Gale & Co., Cowes), propelled by two 90-h.p. petrol engines, the fuel tank of 280-gall. capacity being situated right aft, and separated from the cabins by a steel bulkhead.

Compression-ignition engines are, however, now used for all of the larger and many of the smaller craft.

*Kerosene.*—The greater safety and lower cost of the heavier petroleum products soon led inventors to devise means of utilizing these as fuels for internal-combustion engines, and the earliest of the " paraffin- " or " kerosene "-engines to achieve success was that of Priestman of Hull in 1885. This engine operated on the single-acting four-stroke cycle and used kerosene of specific gravity about 0·8 and flash-point about 100° F. as fuel. The fuel was forced by air pressure from a reservoir and delivered in the form of a misty cloud through a fine spraying jet into an exhaust-heated vaporizer wherein it at once became gaseous; during the suction stroke of the piston this oil gas, mixed with the proper volume of air from an automatic air inlet valve, was drawn into the cylinder, compressed, and fired in the usual manner. The engine was started by a preliminary heating of the vaporizer by a blow-lamp, the air pressure for the oil spray being provided by a hand-pump; the flywheel was then briskly turned by hand, and the engine started.

The preliminary heating necessary to gasify the fuel causes the mixture delivered to the engine to be raised in temperature, and so diminishes the volumetric efficiency of the engine. Further, the " kerosenes ", " paraffins ", or " lamp oils " are prone to detonate easily, and to avoid this low compression pressures must be employed in the engine cylinder. The result is that the power output is reduced to 80–85 per cent of that obtainable with petrol as fuel, notwithstanding the rather larger calorific value per pound of the heavier fuel.

A great many paraffin marine engines of Priestman type have been constructed for river and coastal passenger-carrying boats, one (1929) case, for example, being that of a 60 ft. × 16 ft. × 5½ ft. cabin passenger vessel for use along the south coast of England, fitted with two 8-cylindered Parsons paraffin engines each developing 56 b.h.p. As in the case of petrol-engines, however, kerosene engines are now rarely used for any but very small craft.

**"Heavy Oils": Akroyd Stuart.**—An extremely important advance in the development of the heavy-oil engine was made by Akroyd Stuart (1886–90), who first utilized successfully the heated surface of a special portion of the combustion chamber to vaporize the heavy oil fuel, and also, the vapour being mixed with the correct quantity of air for complete combustion, to cause *automatic* explosion at the required instant.

The essential features of Akroyd Stuart's invention are shown diagrammatically in fig. 5; a single-acting four-stroke cycle cylinder AA and piston B with combustion chamber K, all water-jacketed, has an uncooled vaporizer V in constant communication with it through the narrow passage N. An inlet valve E and exhaust valve F open into the combustion chamber and an oil delivery pipe J into the vaporizer. To start the engine the vaporizer receives a preliminary heating by a blow-lamp for a few minutes; the engine flywheel is then turned, and the piston during the suction stroke draws a charge of fresh air through the valve E, directly into the combustion chamber K. Simultaneously a charge of oil is injected into the vaporizer or "hot bulb" V by a small force-pump driven from the engine. During suction the charge of oil vaporizes and diffuses through V, being mixed, however, only with the residual exhaust gas from the preceding cycle. On the compression stroke of the piston air passes from K to V through the narrow neck N, and mixes with the oil vapour. The mixture is at first too rich to ignite, but matters are so adjusted by trial that just as compression is completed a correct explosive mixture is created; the heat of the vaporizer wall then causes ignition, and the piston is driven outwards, thus performing its working stroke; at the end of this stroke the exhaust valve F is opened, and during the second in-stroke of the piston the burnt gases escape into the atmosphere; the cycle then recommences.

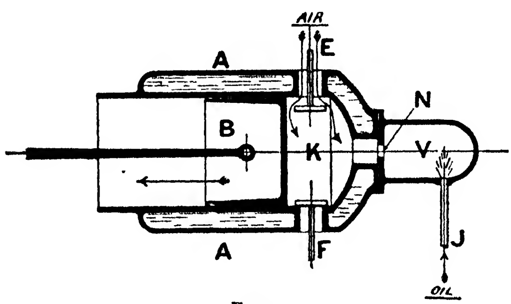


Fig 5

In the Otto, Clerk, and Day engines already described, some special ignition apparatus is necessary, but it will be noted that in this engine, usually known as the "Hornsby-Akroyd" engine, this is no longer needed, the firing being entirely automatic. This engine of Akroyd Stuart is the foreunner of the many "hot bulb" heavy oil engines formerly largely used, but now replaced by cold starting compression-ignition engines.

As fuel the Hornsby-Akroyd engine used heavy petroleum oils of specific gravity ranging from about ·825 to ·86, and flash point from 88° F. to above 200° F.

Three ways of increasing the power output of the original Otto type four-stroke cycle engine have already been mentioned, viz. (1) by increasing the number of working cylinders, (2) by increasing the impulse-frequency, and (3) by increasing the speed of revolution. A fourth way is by increasing the mean effective pressure in the cylinder during the working stroke. Material limitations, however, soon render this impracticable with the types of engine so far described, as in all these the combustion of the working charge is practically instantaneous or " explosive ", and effectually to raise the mean pressure on the piston during the working stroke would necessitate an extremely high initial explosion pressure

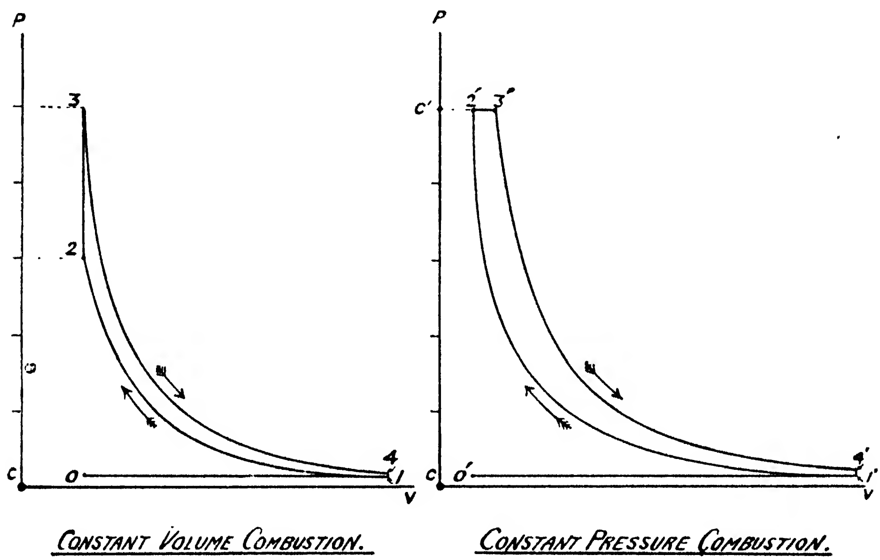


Fig. 6

which would be difficult to provide for in design, except in the case of very small bore engines such as are used for motor vehicles and aircraft.

These types are known as " constant-volume combustion " types, as the rise of pressure on ignition of the charge occurs so suddenly that the piston has no time to move perceptibly; the typical indicator diagram is shown on the left of fig. 6.

**The Diesel Engine.**—Dr. Diesel's book, *The Rational Heat Motor*, appeared in 1893, and the first successful Diesel engine was built by the Augsburg Co. in 1897; the cycle finally decided upon after much experimentation was as follows:

(1) A suction stroke of air only.

(2) Compression of the air to the maximum working pressure (500–600 lb. per square inch).

(3) A regulated injection of the fuel charge into the compressed, and consequently highly heated, air, causing spontaneous ignition, and burning

*at constant pressure* while the piston performs the first portion of its working stroke.

(4) Subsequent expansion of the burning charge to the end of the working stroke.

(5) Release of the burnt charge into the atmosphere at the end of the working stroke, and expulsion of the residual gas during the second instroke of the piston.

Thus the engine worked on the four-stroke cycle, its two distinguishing features being: (1) the employment of a very high compression ratio, and (2) the gradual injection of the fuel into the combustion chamber, thus causing combustion to take place at constant pressure while the piston performed the early portion of its working stroke. The Diesel cycle is accordingly often referred to as a "constant-pressure combustion" cycle. Indicator diagrams of "constant volume" and "constant pressure" type are contrasted in fig. 6. Consider firstly the constant-volume diagram: at o the piston is at the top of its stroke, co representing the volume of the combustion chamber; during the suction stroke a charge of mixture at atmospheric pressure is sucked into the cylinder and the line o1 is traced on the diagram; the inlet valve closes at 1, and the returning piston next compresses the charge into the combustion chamber, the increase of pressure with reduction of volume being shown by the compression curve 12. At 2 the compressed mixture is fired, explosion takes place, and the pressure rises instantly to 3, as shown by the vertical line 23, the volume of the charge remaining unchanged. The piston is then driven outwards, thus performing its working stroke, the pressure of the working gas falling as its volume increases as shown by the expansion curve 34; at 4 the exhaust valve opens, the burnt gases exhaust into the atmosphere, and the pressure in the cylinder drops to atmospheric, as shown by the line 41. Finally the piston in performing its second in-stroke expels the remaining burnt gases as indicated by the line 10. The cycle is then repeated.

Now in the Diesel cycle (right-hand diagram of fig. 6) there is similarly a suction stroke o′1′, and a compression stroke 1′2′; but the compression in this case is carried much farther than before, viz. up to the maximum pressure attained. At 2′ regulated admission of the fuel begins, and the mixture spontaneously ignites and burns *at constant pressure* while the volume increases from 2′ to 3′. At 3′ the fuel supply ceases, and the working gases then expand at falling pressure as indicated by the expansion curve 3′4′; the exhaust opens at 4′, the burnt gases escape, the pressure falls to 1′ (atmospheric), and during the second in-stroke of the piston, 1′o′, the remaining burnt gas is expelled. The cycle is then repeated. The average vertical height of the theoretical constant-pressure diagram (i.e. the mean indicated effective pressure) exceeds that of the corresponding constant-volume diagram by some 10 per cent. On the test bed mean effective pressures of 140 lb. per square inch have been attained with large four-stroke marine Diesel engines, but in actual everyday service conditions the mean pressure employed ranges from about 90 to 105 lb. per square inch. A further important advantage of the Diesel cycle is the high compression ratio that is practicable (1) owing to air only being compressed, and (2) owing to the combustion at constant pressure enabling the compression to be carried up to the maximum pressure attained. In ordinary gas, petrol, and other engines in which during suction the working charge, consisting of gas or vapour mixed with air, is drawn into the cylinder and compressed, the risk of a premature spontaneous ignition of the mixture limits the compression ratio (i.e. $\frac{\mathrm{IC}}{\mathrm{OC}}$, fig. 6) to a maximum value of about six in ordinary use; in the Diesel engine, however, a compression ratio of twelve, or even somewhat more, is commonly employed; and the thermodynamic efficiency of an internal-combustion engine increases with the value of the compression ratio.

Accordingly, although the theoretical efficiency of the constant-pressure cycle is somewhat less than that of the constant-volume cycle for a given compression ratio, the net result of the several advantages of the Diesel in practice is that it exhibits the greatest economy of fuel in normal running of any internal-combustion engine.

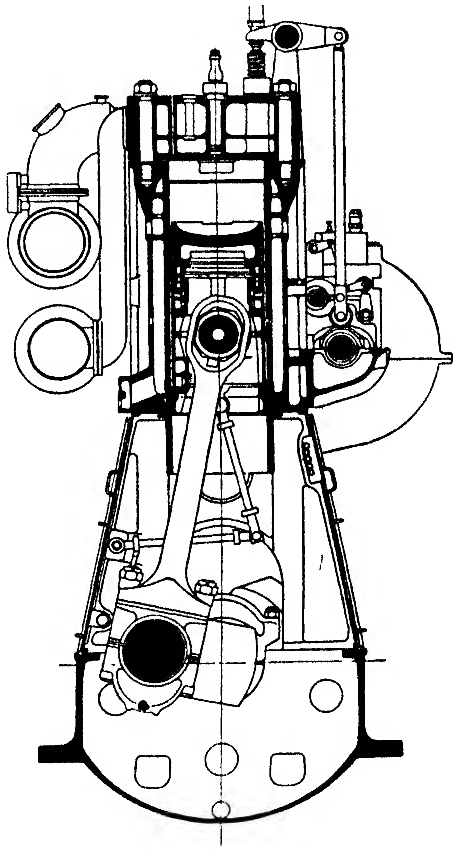

Fig. 7.—Section through Cylinder of 700-b.h.p. M.A.N. Trunk Piston Airless-injection Engine

A section of a characteristic type of four-stroke single-acting trunk-piston marine Diesel engine is given in fig. 7; this is one of a six-cylinder 700-b.h.p. unit running at 145 r.p.m., built by the Maschinen-Fabrik Augsburg-Nürnberg Co. (usually referred to as the M.A.N., or Augsburg Co.). The cylinder bore is 480 mm. (18·9 in.), and the stroke 700 mm. (27·6 in.). When it is noted that a pressure of 500 lb. per sq. inch implies a piston load of roundly 60 tons applied about seventy times per minute during running, the reason for the massive strength exhibited by Diesel engine designs becomes apparent. The well water-jacketed cylinder is fitted with a detachable head of deep section, also water-jacketed; the head contains (1) an automatic air inlet valve (not shown); (2) a cam-operated exhaust valve, indicated on the right;

(3) a fuel injection nozzle in the centre; (4) a small valve for the admission of compressed air for starting purposes; and (5) a relief valve. Further details of design are dealt with later.

It will be noted from fig. 7 that in its general arrangement this single-acting four-stroke Diesel follows the well-tried type of the earlier gas-engine, with trunk piston and without crosshead; though this type has been very largely used in marine service on account of its efficiency, relative simplicity, and minimum vertical height, it is rapidly being displaced by the crosshead type engine in which the cylinder is relieved of the side thrust of the connecting-rod, with consequent great reduction in piston friction and wear; one of the chief troubles in trunk piston Diesels has been the loss of compression resulting from wear of the piston rings and "ovalling" of the cylinder bore through side thrust, and the advantages of the crosshead design are now generally realized. Great practical difficulties were at first encountered with the Diesel engine arising from the high compression pressure employed and consequent high temperatures developed, and many failures due to cracked and seized pistons and cylinders were recorded. Lubrication details and the preservation of gas-tightness in valves and pistons also necessitated much research and experimentation. In the early Diesels, moreover, the charge of fuel was always blown into the combustion chamber by a very high-pressure air blast supplied from an air reservoir in which it was stored at from 800 to 1000 lb. per square inch, and the maintenance of this apparatus in efficient working condition demanded close study. In recent years great progress has been made in the simplification of the fuel injection by replacing the high-pressure air blast by "solid" or "airless" injection of the fuel by means of a small force pump; instances are described later. Trouble has sometimes arisen, again, through pre-ignition owing to leakage of oil past the fuel inlet needle valve, or to this valve sticking up; this results in oil being present during compression, and the inflammable mixture thus formed pre-igniting, the pressure being augmented by the normal oil charge injected when the top of the stroke is reached; in this way momentary pressures in the combustion chamber of 1000 lb. per square inch and more have been created, with resulting disastrous rupture of the cylinder. On account of this possibility relief valves were in some cases fitted in cylinder heads, and were usually set to lift whenever the pressure in the combustion chamber exceeded the normal maximum by about 40 per cent.

A further source of trouble to the early Diesel engineers arose from want of uniformity in the fuels available; experiments were conducted with a great variety of substances including petrol, kerosene, gas oil, crude Russian, American, and German oils, residual oils as Astatki, shale oils, coal-tar oils, lignite oils, palm and nut oils, castor oil, fish oil, alcohol, coal gas, producer gas, and coal dust. It is of interest to note here that Dr. Diesel originally proposed to use coal dust as fuel, but all the early experiments proved unsuccessful; the possibility was not, however, lost sight of, and it was stated* early in 1929 that, since 1917, an 80-h.p. single-cylindered air-injection engine of 16·5-in. bore and 24·8-in. stroke, running at 160 r.p.m., had worked regularly not only on finely pulverized coal from Silesia, but also on lignite coal dust, charcoal dust, rice mill dust, sawdust, and even the dust of metallurgical coke! But for marine purposes lengthy experience has shown that the most suitable fuels are thoroughly clean petroleum oils of specific gravity ranging from about ·9 to ·97, flash-point 220° F. to 250° F. (closed), and net calorific value of roundly 18,500 B.Th.U. per pound.

To bring even the smaller single-cylindered single-acting four-stroke trunk-piston Diesel engine to a successful issue proved a lengthy and costly process, largely on account of practical difficulties arising from the creation of great forces and high temperatures and of obtaining uniform and complete combustion of the fuel; in its final practical form it involved the exercise not only of great skill in design but also of great refinements in exactitude of manufacture.

**The Two-stroke Diesel Engine.**—With the successful application of the Diesel engine to marine propulsion soon arose an urgent need of increasing the power output per cylinder in order to save space, and this caused attention to be focused upon the production of a serviceable two-stroke type. This raised again acutely the problem of overcoming over-heating difficulties, and again much costly and lengthy experimentation had to be incurred before success was attained.

During the charging of ordinary small two-stroke engines of Clerk or Day type the working charge of gas, or vapour, mixed with air is received by the cylinder, and it has been found impossible with these simple motors to prevent loss of some of the fresh mixture by short-circuiting through the exhaust ports (see figs. 2, 3, and 4). Further, the expulsion of the burnt gases, assisted by the fresh charge entering under slight pressure, has to take place during the short interval elapsing between the unmasking of the exhaust ports by the piston, and their subsequent early closing on its return. This results in considerably more residual exhaust gas being mixed with, and heating, the fresh charge than is the case with a four-stroke engine wherein the exhaust period extends over a much longer time, and the gases are definitely expelled by the returning piston during its exhaust stroke. Thus a smaller quantity of somewhat heated fresh charge is received by the cylinder, and the exhaust products are not so completely voided as in the four-stroke case. This is usually expressed by saying that the " volumetric efficiency " of the engine is reduced, and that the " scavenging " is imperfect.

If the ordinary two-stroke motor were of equal volumetric efficiency, and were scavenged as effectively as the four-stroke, it would give, size for size, just twice the power output of the four-stroke at the same speed. Actually its performance usually falls markedly short of this ideal; thus in

* *v. Gas and Oil Power*, Feb., 1929, p. 84.

some experiments on a small three-port two-stroke petrol motor Watson and Fenning obtained the following results.

| At Revolutions per Minute. | Percentage of Fresh Charge lost through Exhaust. |
|---|---|
| 600 | 36 |
| 1200 | 20 |
| 1500 | 6 |

The volumetric efficiency of the engine was only about 40 per cent, and the conclusion was reached that the output of this two-stroke motor was only about 47 per cent greater at 900 r.p.m., and only 29 per cent greater at 1500 r.p.m., than that of a corresponding four-stroke engine. Even with a Dolphin Clerk-cycle marine engine (fig. 2), with separate charging-pump and specially formed conical combustion chamber, the output compared with the corresponding four-stroke was only 57 per cent greater at 400 r.p.m., falling to 31 per cent at 1200 r.p.m. Notwithstanding these fundamental drawbacks, however, the great simplicity and ready reversibility of the small two-stroke motor have caused it to be used very extensively in small motor-boats and similar craft.

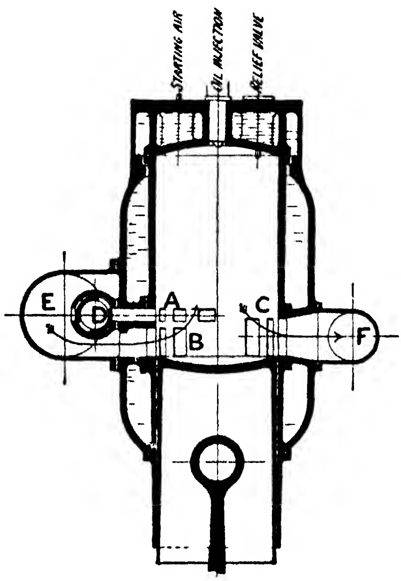

Fig 8

In employing the two-stroke cycle the Diesel engine starts with the important advantage that as air only is taken in during charging, any loss through short-circuiting is quite immaterial, and in fact in at least one very successful design a special supply of air is provided in order to improve the scavenging to the utmost. Thus, fig. 8 is a diagrammatic view of an older two-stroke Diesel of Messrs. Sulzer Bros., who have devoted attention to this type of engine since about 1907. The diagram illustrates a single-acting well water-jacketed cylinder fitted with a detachable head of deep section, also water-cooled, and carrying three valves, viz. (1) the fuel injection valve in the centre, (2) a small valve for the admission of compressed air for starting purposes, and (3) a relief valve.

At the lower end of the cylinder is a belt of exhaust ports C, which are first opened by the piston, thus permitting the escape of the burnt gases into the exhaust pipe F, and a belt of inlet ports B, through which fresh air, at a pressure of some 3 lb. per square inch, enters the cylinder from the air-supply or scavenging pipe E.

A special feature of the design is the third belt of small air ports A placed above the inlet ports B, and controlled by a rotary valve D situated within the scavenging pipe. This valve opens after the piston, in rising, has just cut off the ports B, but before ports C are closed; the air entering the cylinder through the "auxiliary" scavenging ports A, which extend round one-half the circumference, not only completes the scavenging of the exhaust gases through C, but also causes the air in the cylinder to be at slightly above atmospheric pressure (some 2 lb. per square inch), and thus effects a supercharging of the cylinder. In later marine Sulzer two-stroke engines, in place of the mechanically operated rotary valve D, automatic disc valves control the belt of auxiliary scavenging ports; these valves rise as soon as the pressure within the cylinder falls below that in the main scavenging reservoir E.

The supply of low-pressure air to the scavenging reservoir E was effected in earlier designs by a large scavenging pump directly driven from the after end of the crank-shaft; in later designs the scavenging air is supplied by a separately driven turbo blower, by which the shaft horse-power of the main engine is increased, and the total weight of the installation reduced. The well-known double-piston Oechelhauser gas-engine also appears as a single-acting two-stroke Diesel in the Doxford design, and again in the ingenious two-cylinder four-piston unit of the Fullagar engine; instances of these and other designs are described later.

**Double-acting Diesel Engines.**—A fifth means of increasing the power output of an internal-combustion engine is by making the cylinder double-acting. In all the engines so far described the working cycle takes place at one end only of the cylinder. By closing the cylinder at both ends and causing the cycle to be carried out on both sides of the piston, the power output per cylinder is obviously doubled. The evolution of a quite satisfactory double-acting Diesel engine naturally proved a matter of great difficulty, not only owing to cooling problems but also because, in four-stroke cycle engines, it was found very difficult to arrange the valves in the bottom cylinder cover on account of the presence of the large stuffing-box of the piston-rod.

With double-acting engines it is manifest that water or oil cooling of the piston becomes necessary; the engine must, of course, be of the cross-head type. A further point to be noted is that each four-stroke cylinder provides two working impulses in one revolution, followed by an idle revolution, so that although each cylinder gives an *average* of one working impulse per revolution, the crank-shaft turning effort per cylinder is very un-uniform; this disadvantage is, of course, minimized by employing multi-cylindered engines.

The difficulty of arranging the valves in the bottom cover of the cylinder has been overcome by Messrs. Burmeister and Wain by providing at the lower end a separate casting bolted to it and forming the combustion chamber; in this casting all the valves are located, the air inlet being at the top, the exhaust at the bottom, and the oil fuel and air starting valves nearly horizontal between them. By 1928 double-acting four-stroke

"B. & W." engines had been built for a twin-screw installation of 18,000 b.h.p., each engine having eight cylinders, the normal running speed being 100 r.p.m. Later developments are discussed in Chapter IX.

**Double-acting Two-stroke Diesel Engines.**—It is clear that each cylinder of this type furnishes two working impulses per revolution, and thus equals the impulse-frequency of the double-acting steam-engine. Much costly and risky experimentation was embarked upon some years before the war of 1914–18 in Germany and elsewhere, in the attempt to produce a satisfactory two-stroke double-acting Diesel engine for warships. The thermal problems encountered proved formidable, and extensive research was necessitated not only in connection with design detail arrangement but also in regard to the qualities of material required for construction. The problem may, however, be regarded as practically solved, and the multi-cylindered two-stroke double-acting Diesel engine—the last step in internal-combustion engine design—is now an accomplished fact; e.g. in 1928 Richardsons Westgarth built for a 3000-ton tanker a three-cylindered 1200 b.h.p. double-acting two-stroke airless injection Diesel engine running normally at 90 r.p.m. Engines of much larger size have been constructed in recent years (see Chap. IX).

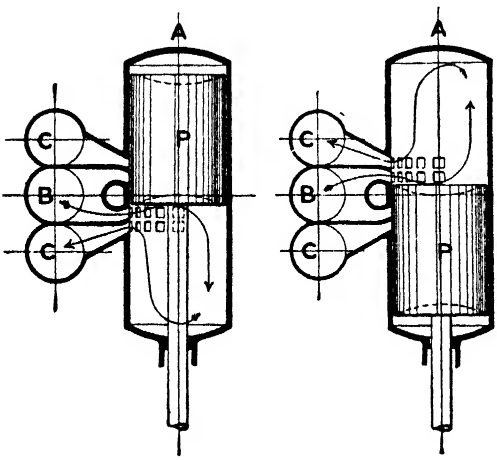


Fig. 9

A diagrammatic view of the M.A.N. double-acting two-stroke arrangement appears in fig. 9; the closed cylinder is fitted with a piston P, and piston-rod, and the exhaust and air supply ("scavenging") ports are arranged on one side of the cylinder as shown, the two belts of scavenging ports being near the middle, while the exhaust belts at the top and bottom ends of the cylinder are above and below these respectively; these belts of ports extend about half round the cylinder, and are inclined as shown in order to cause the scavenge air and exhaust gases to take the general course indicated by the arrows; the scavenging with this arrangement is said to be very satisfactory. It will be noted that the scavenging ports are not uncovered until after the exhaust ports have been fully opened; the scavenge air reservoir is B and the exhaust pipes CC. Instances of the two-stroke double-acting type in more detail are given later.

Next to be briefly noticed are a number of smaller quick-speed engines, commonly referred to as "high-speed Diesels", which in general mode of operation resembled the large Diesels, though the combustion is usually, at least in part, of the constant-volume type.

"**High-speed" Diesel Engines.**—Large marine Diesel engines directly employed in ship propulsion are run at speeds ranging from about 80 to 200 r.p.m., but for many other purposes, e.g. direct-coupled to generators, smaller Diesels running at up to 900 r.p.m. are installed. These occupy less space and are less costly than the slower-running type, and are usually of the vertical two- to six-cylindered four-stroke single-acting type with enclosed crank-chamber and forced lubrication (15–30 lb. per square inch). Power output is sometimes as high as 600 h.p. Among well-known builders of this general type may be mentioned Belliss, Brotherhood, Burmeister and Wain, Carels, Fraser and Chalmers, McLaren-Benz, Sulzer, and Werkspoor.

A number of British firms, including Crossley Bros., L. Gardner & Sons, and The National Gas and Oil Engine Co., now manufacture high-speed compression-ignition engines.

As pistons have usually to be removed from the top, extra head room must be left with the vertical type. Horizontal auxiliary marine high-speed Diesels are accordingly sometimes installed; e.g. in 1928 the refrigerating machinery of the first five motor-ships of the Nelson line was driven by a horizontal single-acting four-stroke four-cylinder vis-à-vis Diesel engine of 16·5 in. bore and 24 in. stroke, with enclosed crank-chamber. The vis-à-vis, or opposed cylinder, arrangement results in a very compact design. Multi-cylinder vertical engines are, however, now almost universal for auxiliary machinery.

**The Ricardo-Brotherhood High-speed Diesel Engine.**—Early in 1929 Messrs. Brotherhood completed the construction of a six-cylindered single-acting four-stroke cycle trunk-piston airless injection Ricardo-Diesel engine designed to run at the unusually high normal speed of 900 r.p.m. An external view of this engine, direct-coupled to a generator, appears in fig. 10; it will be seen that it is of the vertical type with enclosed crank-chamber. This engine is unique in being operated on the single-sleeve Burt-McCallum method, in which the working barrel of each cylinder is a case-hardened and ground ported steel sleeve to which a small motion partly reciprocating and partly angular is communicated from the crank-shaft by suitable intermediate mechanism; the sleeve ports register at the correct instants with corresponding ports in the surrounding fixed cylinder casing, and the inlet and exhaust portions of the cycle are thus effected. The pistons are of aluminium alloy, with five spring rings and one "scraper" ring; the connecting-rods are alloy steel stampings, tubular in section and machined all over. The cylinders have a bore of 7·5 in., the stroke is 12 in., and the power output is 300 b.h.p. at 900 r.p.m.; the corresponding piston speed has the high value of 1800 ft. per minute. Lubrication is forced; from a bed-plate oil reservoir a pump delivers pressure oil to the main and big-end bearings, lay-shaft, and sleeve-operating gear. The pump suction is through a gauze filter, and before delivery into the supply-pipe system the oil is circulated through an Auto-Klean strainer. From the bearings the oil collects in the sump, whence it is drawn by a second

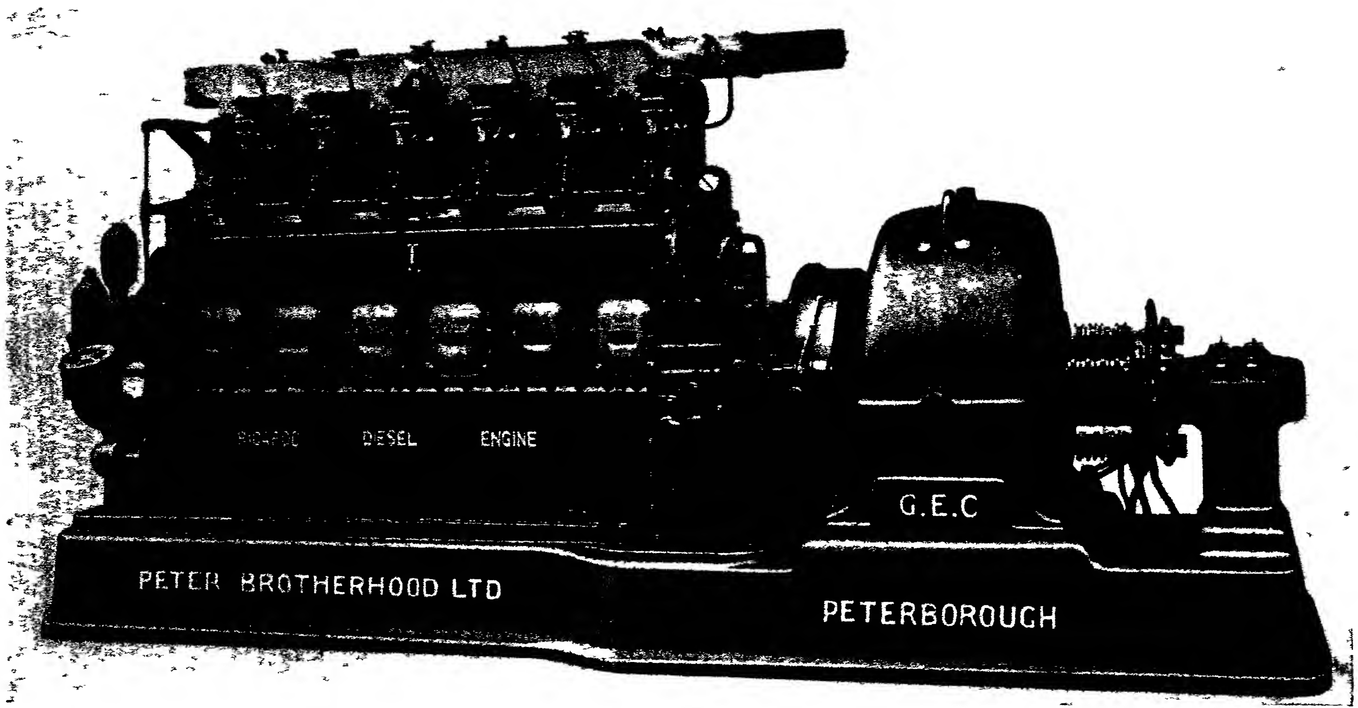


Fig 10 —Ricardo-Brotherhood High-speed Diesel Engine oil-pump, passed through a system of cooling pipes in a water-jacketed portion of the bed-plate, and thence returns to the bed-plate oil reservoir mentioned above. The weight of the six-cylinder engine illustrated, with flywheel, is about 47 lb. per brake horse-power only; and the engine is 9 ft. 6 in. in length, 4 ft. 5 in. in overall width, and 5 ft. 10 in. in total height.

A Vickers-Ricardo eight-cylinder engine giving an output of 400 b.h.p. at 900 revolutions per minute is illustrated in fig. 11, and a sectional view

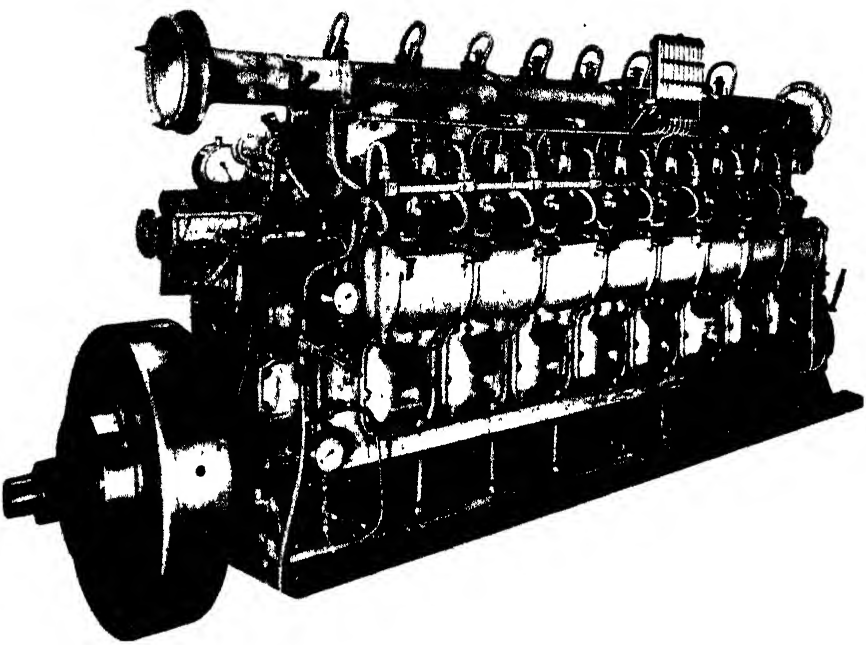

Fig. 11.—Vickers-Ricardo Eight-cylinder Engine, 400 b.h.p. at 900 r.p.m.

of the engine in fig. 12. The weight of this engine is approximately 6½ tons. The following is a description of some special features incorporated in this design.

It is of the four-stroke cycle type utilizing the Vickers-Ricardo sleeve-valve arrangement, which eliminates the usual valves in the cylinder head.

*Sleeve Valve.*—In place of poppet valves for induction and exhaust with their attendant springs and levers, each cylinder is provided with a special cast-iron sleeve with circumferential ports in its upper section matching with corresponding ports in the cylinder barrel. The sleeve is operated from a layshaft by means of a lever and crank mechanism, which gives it a combined reciprocating and rotary motion, resulting in rapid opening and closing of the induction and exhaust ports without noise. Suitably shaped induction ports give a rotational turbulence to the air charge, enabling good combustion to be obtained.

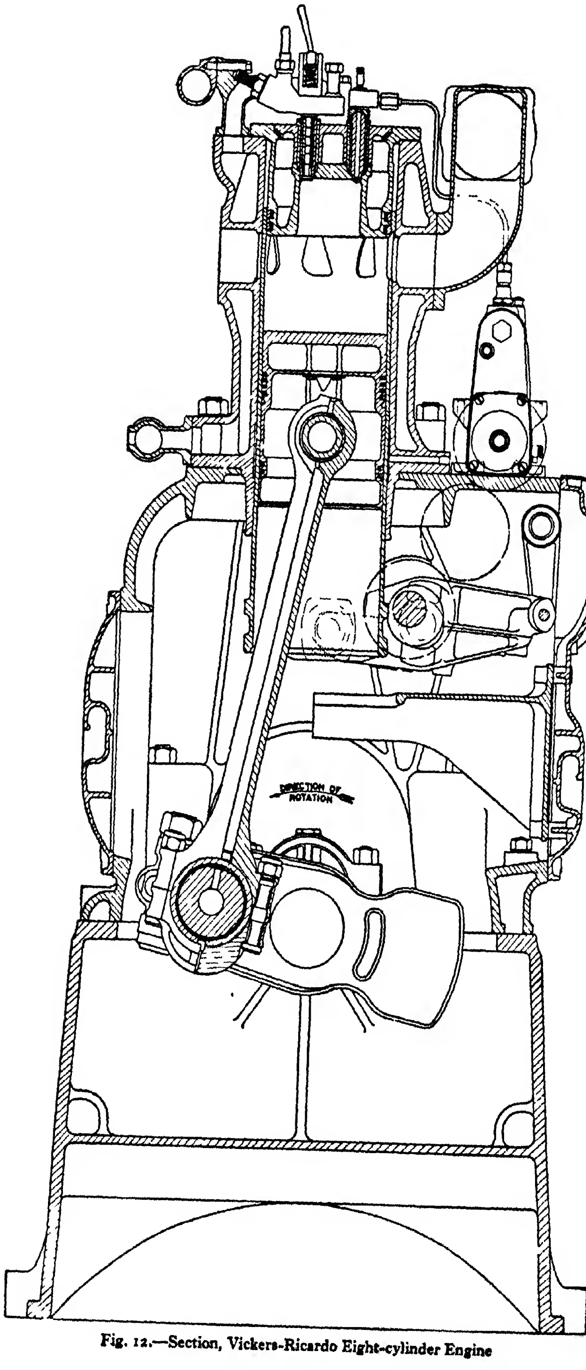

Fig. 12.—Section, Vickers-Ricardo Eight-cylinder Engine

The motion of the piston relative to the sleeve is very different from that of the piston to the liner in the normal engine, resulting in better lubrication of the cylinder and elimination of the excessive wear usually found at the top of Diesel engine liners.

*Cylinder Cover.*—As the cover has only to accommodate the fuel and air-starting valves, its design is simple and allows a two-part forged steel and cast-iron construction to be used, thereby facilitating internal cleaning. The combustion space is shaped with special reference to the rotational turbulence created in the cylinder.

*Spray Valve.*—The fuel spray valve is of the automatic type, with no moving parts other than a spring loaded valve. The nozzle has only one hole of comparatively large size, thus preventing choking. The valve is fitted vertically at one side of the combustion chamber; the spray issuing into the rapidly rotating air contents ensures complete burning of the fuel.

*Fuel Oil Pump.*—Each cylinder has its own fuel injection pump of the constant stroke type, with spill action under hand and governor control.

*Lubrication.*—Forced lubrication is supplied to the principal bearings, the oil being cooled and filtered before passing to the system.

The evolution of the quick-revolution or high-speed Diesel engine involved difficult problems connected with the efficient combustion of the oil fuel, and the prevention of excessive bearing pressures. Its reduced size and weight appeal strongly to the marine engineer as a desirable auxiliary for generator driving, but it is above all essential that complete reliability shall be assured.

*Definition.*—It has been found convenient to enunciate as follows the official definition of a Diesel engine: *A Diesel engine is a prime mover actuated by the gases resulting from the combustion of a liquid or pulverized fuel injected in a fine state of subdivision into the engine cylinder at or about the conclusion of the compression stroke. The heat generated by the compression of air within the cylinder is the sole means of igniting the charge. The combustion of the charge proceeds at, or approximately at, constant pressure.*

**"Semi-Diesel" Engines.**—The true Diesel is a heavy and costly type of engine characterized by a very low fuel consumption and high reliability. The success attending its use soon caused engineers to devote their attention to the production of lighter and less expensive engines of high fuel economy but without the very high compression pressure and air-blast injection of the normal Diesel, and great success has been achieved in this direction. It has been found convenient to classify these types under the term "Semi-Diesel", the accepted definition of the type being as follows:

*A Semi-Diesel engine is a prime mover actuated by the gases resulting from the combustion of a hydrocarbon oil. A charge of oil is injected in the form of spray into a combustion space open to the cylinder of the engine at or about the time of maximum compression. The heat derived from an uncooled portion of the combustion chamber, together with the heat generated by the com-*

*pression of air to a moderate temperature, ignites the charge. The combustion of the charge takes place at, or approximately at, constant volume.*

These semi-Diesels use as fuel the same range of oils as the Diesels, and among them are included "hot-bulb" or "surface-ignition" engines, and also a number of designs with no uncooled portion of the combustion chamber, and thus approximating more nearly to true Diesels; such are the high-compression semi-Diesels in which a somewhat lower compression pressure is employed than in the true Diesel, while the combustion is usually partly at constant volume and partly at constant pressure; these engines are capable of being started from cold, they use heavy oils as fuel, with airless injection, and in economy of running differ but little from the true Diesel constant-pressure-combustion type.

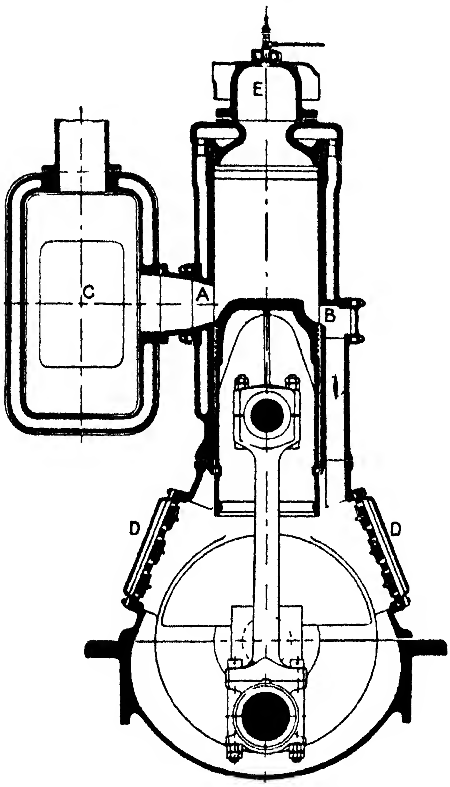

Fig. 13.—Bolinders Engine

Progress in recent years in design and reliability of cold starting compression - ignition engines has rendered the "hot-bulb" type of engine practically obsolete.

**The Bolinders Engine.**—A typical hot-bulb semi-Diesel is the sturdily built "Bolinders" engine largely used for the propulsion of trawlers, tugs, dredgers, yachts, small freighters, &c., and in other cases where only semi-skilled attention is available. The type is built with one, two, or four cylinders, and the output ranges from 8 to 600 h.p.

Illustrated in section in fig. 13, it is seen to consist of a normal Day type trunk-piston single-acting two-stroke two-port engine with crank-chamber compression. The cylinder is water-jacketed, with an unjacketed hot bulb E of Akroyd Stuart type forming a prolongation of the combustion chamber; compare figs. 3 and 5.

The piston first overruns the exhaust port A, the burnt gases immediately escaping into a (water-cooled) silencer C, and next opens the inlet port B, permitting *fresh air* at a pressure of 3 or 4 lb. per square inch to enter the cylinder from the crank-chamber. The air supply to the crank-chamber during each up-stroke of the piston is through groups of automatic valves DD, located in the wall of the crank-case.

The bulb E having received a preliminary heating from a blow-lamp

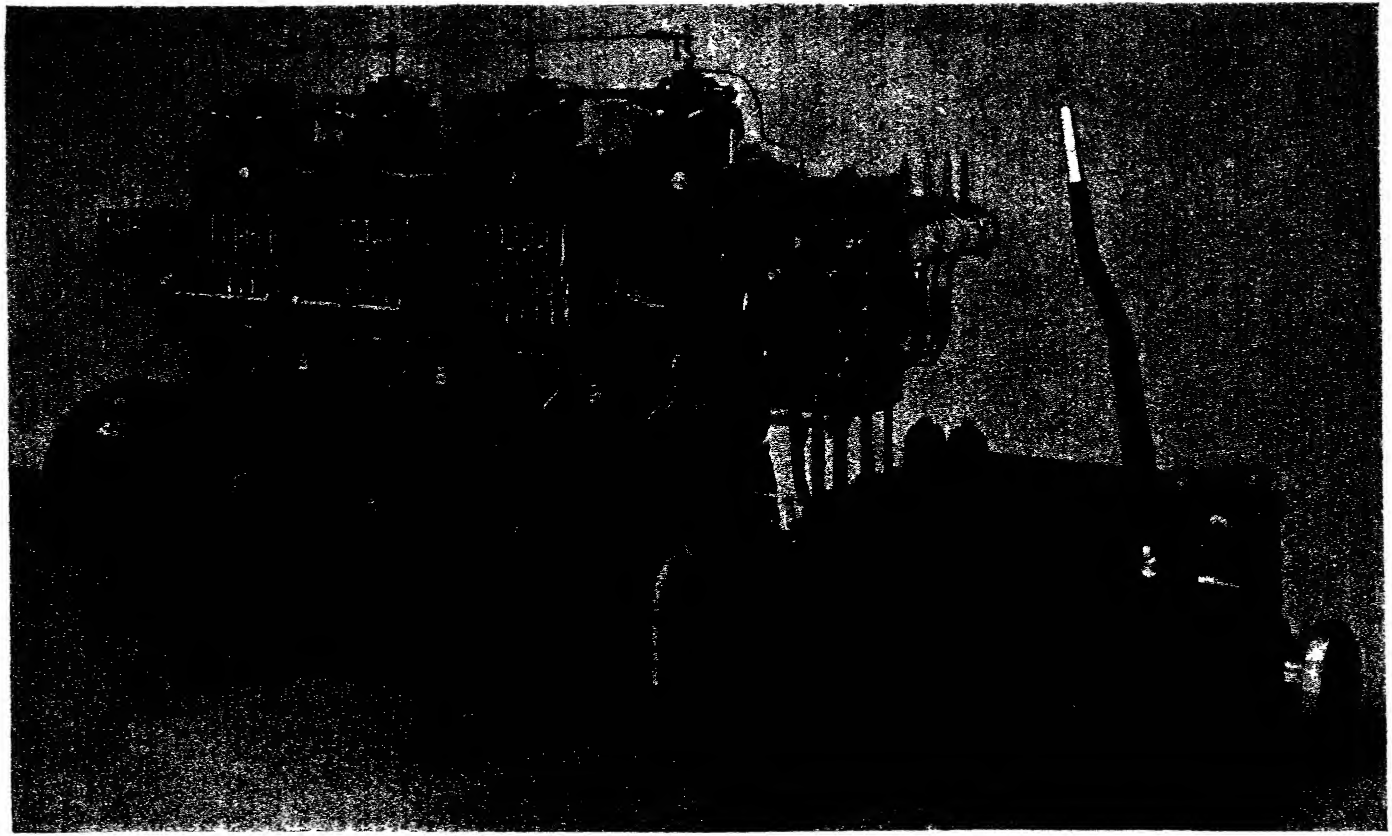

Fig. 14.—600-b. h. p. Bolinders Engine, Four-cylinder

for 10 to 15 min., and the engine then turned or "barred" round by hand until the piston has just passed the top of its stroke, air at a pressure of 120–180 lb. per square inch is admitted from an "air bottle" through a small starting valve and the engine at once moves off. The fuel supply is then admitted and the motor takes up its working cycle.

As soon as the cylinder is well warmed up the blow-lamp may be extinguished, the subsequent ignition being entirely automatic as in the Hornsby-Akroyd engine. The fuel pump injects the working charge of oil into the hot bulb when the piston has completed about half its up-stroke, and ceases just before the completion of this stroke; the oil charge is gasified in the hot bulb and, mixed with the compressed and heated air, forms a mixture which explodes automatically just as the piston passes the upper dead point. The compression pressure employed is 125–150 lb. per square inch.

*Reversing vessels driven by the Bolinders Engine.*—The reversal of the vessel is effected in three different ways: (1) in the first method the engine drives the propeller through a friction clutch; to reverse, the engine is de-clutched and slowed down; a counter-impulse is then given to the rising piston by providing an early injection of the oil charge, which stops the piston momentarily and then drives it downwards, thus reversing the direction of rotation of the crank-shaft; the fuel injection timing is at once adjusted, and the engine speeds up in the reverse direction. It has already been pointed out that the Day type two-port and three-port two-stroke engines will run equally well in whichever direction they may be started.

An external view of the four-cylindered 600 h.p. Bolinders engine is shown in fig. 14; in this large engine the clutch is operated by compressed air supplied from the starting air-bottle. In sailing-vessels using the engine only as an auxiliary, the clutch has the incidental advantage that by de-clutching the propeller is enabled to revolve freely and thus offer no resistance to the vessel's motion when proceeding under sail.

(2) In the second method of reversal the engine runs always in the same direction, but hand-operated reversing gear is fitted between the engine and the propeller shaft. This method is used in small craft which have to be manoeuvred single-handed, as e.g. in barges, small tugs, and pleasure craft, the engine control being brought up to the helmsman.

(3) In the third method, the propeller itself is reversible, the blades being turned to any desired position; the engine runs always in the same direction. All vessels so provided are fitted with a clutch between the engine and propeller shaft. This method is of special value in fishing-vessels where speeds must be varied within wide limits. When sailing with the engine also running, the two propeller blades can be so set as to give the best combined effect, and when sailing only the blades can be set so as to offer minimum resistance.

**The Gardner Heavy-oil Engine.**—A typical example of the modern high-compression semi-Diesel is furnished by the "J-type" Gardner marine engine, which is illustrated in section in fig. 15. Here again is a two-stroke two-port engine, but in this case there is no hot bulb, the whole combustion chamber equally with the cylinder being fully water-jacketed. As before, fresh air only is compressed, and the working charge of oil is sprayed by a force pump into the combustion chamber just before the completion of the compression up-stroke. In this engine, however, the compression pressure has the high value of 400 lb. per square inch, and the high temperature thus produced suffices for the instant conversion of the oil spray into vapour, and explosion of the resulting mixture. The combustion takes place at constant volume (see fig. 6), and the maximum pressure on ignition rises immediately from 400 lb. per square inch to about 580 lb. per square inch when full power is being developed.

The cylinder head or "breech" is detachable, and carries the fuel inlet and air starting valves, and also a relief valve which opens in case of the development from any cause of abnormal pressure in the cylinder.

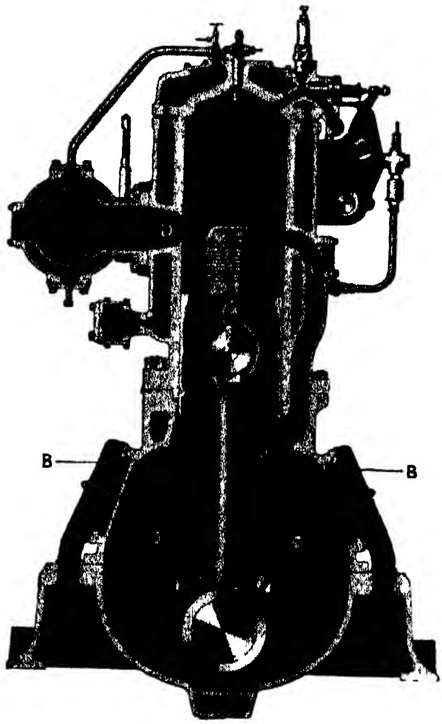

Fig. 15.—The Gardner J-type Heavy-oil Marine Engine

An external view of the six-cylinder Gardner marine semi-Diesel engine, developing 300 b.h.p. at 290 r.p.m., is given in fig. 16. The fuel-injection pumps and compressed-air starting valves —one to each cylinder—are cam-operated; the cam-shaft is driven from the crank-shaft through the enclosed vertical governor shaft fitted with helical gears; the upper helical wheel is splined on the vertical shaft and is movable through the agency of a worm gear operated by the hand-wheel shown on the left of fig. 16. To reverse, the hand-wheel is turned; the first turn puts the fuel pumps out of action and the engine slows down; the wheel-turning is continued, and towards the end of the third turn the cam-shaft has been rotated into the "reversed" position, when the fuel pumps and air-starting valves simultaneously come into action. The starting valves admit a "puff" of compressed air to the slowly *rising* pistons, bringing them momentarily to rest, and then causing their descent, i.e. reversing the direction of crank-shaft rotation; a fourth and last turn of the hand-wheel finally puts the air-starting valves out of action, and the engine at once takes up its working cycle in the reversed direction. The whole operation occupies only about four seconds. The

Fig 16 —Six cylinder two-stroke Gardner Semi-Diesel Marine Engine

starting air is stored in pressed steel " air bottles " at a pressure of 350 lb. per square inch by a two-stage water-cooled air compresser driven from a crank on the main crankshaft. The exhaust gases, at a temperature of about 400° F. at full load, are discharged into a water-cooled manifold (shown on the left in fig. 15), and pass thence into a galvanized steel expansion chamber or silencer which is either water-jacketed or fitted with water-spray injection, and are finally discharged into the atmosphere. An enclosed centrifugal governor regulates the engine speed by varying the stroke of the fuel-injection pump.

Among other well-known British builders of two-stroke oil-engines may be mentioned Allen, Babcock, Beardmore, Crossley, Marshall, Mirrlees, Petter, and Robey.

Four-stroke oil-engines are usually of the high-compression (300–400 lb. per square inch) or " cold-starting " type, with airless or solid injection of fuel, and combustion mainly at constant volume. Notable British builders of these are Blackstone, Crossley, National, Premier, Robey, Ruston, and Tangye, and the range of output is from about 6 to 300 b.h.p.

In this chapter the development of the modern internal-combustion engine, more particularly in relation to its application to marine service, has been sketched briefly from the first commercially successful simple " Otto " four-stroke cycle engine of 1876 to the large and powerful multi-cylindered two-stroke cycle marine Diesel of the present day. It has been possible to refer only to the work of a few of the more prominent, in this connection, of the great number of able engineers who have devoted themselves to this important branch of the profession. Readers who wish to become acquainted with the earlier history of the subject, and in greater detail with the very numerous more modern developments, should consult the larger treatises, as e.g. Donkin's *Gas, Oil, and Air Engines*; Clerk and Burls' *The Gas, Petrol, and Oil Engine* (2 vols.); and current engineering journals. Other more recent works by Robinson, Ricardo, Dicksee, Heldt and Judge should be consulted.

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## CHAPTER II

# Theory of Oil-engines

The foregoing brief résumé of the leading historical facts serves to bring up to date, in respect of leading principles, the development of the oil-engine, and it is desirable now to consider the theory, so far as it has been developed, based on the cycles upon which the modern types of engines operate. As in most other branches of engineering, the theory is of unquestioned importance, although with the oil-engine, more perhaps than in the case of any other prime mover, practical developments are still considerably in advance of the theories underlying the thermodynamical changes involved. Figs. 17, 18, 20, and 21 * show the principal cycles. Fig. 17 represents the Carnot ideal cycle, and is the type of diagram which would be produced in

* For these illustrations we are indebted to the *Proceedings of the North-East Coast Institution of Engineers and Shipbuilders*, and to the paper by Sir Dugald Clerk.

an ideal engine using pure air as the working medium. The four operations, (1) isothermal expansion, (2) adiabatic expansion, (3) isothermal compression, and (4) adiabatic compression, are shown. The assumption is made that no heat is passed to or from the walls surrounding the working medium, but that such passage as is necessary is controlled through the end of the cylinder (see *Applied Mechanics and Heat*, p. 357—THE GRESHAM LIBRARY OF MECHANICAL ENGINEERING).

Fig. 18 shows a standard air-cycle diagram of a constant-pressure engine without heat losses. Adiabatic compression of air is shown from atmospheric to a pressure consequent upon 12·24 compressions. Heat is added at a constant maximum pressure, and expansion carried down to the atmospheric line.

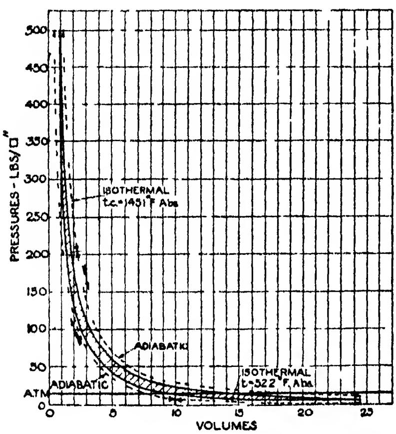


Fig. 17.—Diagram of Air Engine, Constant-temperature Type (Carnot Cycle)

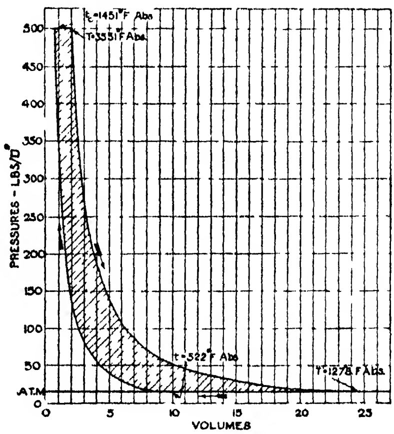


Fig. 18.—Diagram of Air Engine, Constant-pressure Type

Fig. 21 shows a standard air-cycle diagram of a constant-volume engine in which heat is added or combustion takes place at constant volume. Heat is similarly rejected at constant volume.

Fig. 20 shows a standard air-cycle diagram in which the constant pressure addition of heat is performed exactly as in fig. 18, but the expansion is only carried to the volume existing before compression commenced, and the heat is discharged at constant volume instead of at constant pressure. These, then, are the four accepted standards to which reference can be made. It will be exceedingly interesting, therefore, to compare the standards in respect of efficiencies from the thermodynamical point of view, and in regard to the amount of work which can be done per unit cylinder volume employing these various means of utilizing the working fluid, which is air. This comparison gives an index of the size of the engine to give a desired output, as well as examining their utility from practical considerations which will be apparent on inspection. The thermal efficiency of the Carnot cycle is given by $\frac{T_1 - T_2}{T_1}$, where $T_1$ is the maximum absolute temperature, and $T_2$ the minimum, of the working substance. The thermal efficiencies of the Diesel and Otto cycles are given below.

The cycle shown in fig. 21 is more efficient than that shown in fig. 20, as a cycle with the same compression ratio, but less efficient with the same maximum pressure. In comparing these cycles the following assumptions are made, which are necessary in order to bring the comparison to a practical basis. Firstly, that compression starts at atmospheric pressure; secondly, that the maximum pressure in the cylinder with these four cycles is the same, namely, 500 lb. per square inch, which is a good basis for practical comparison, as higher pressure would give loads on the parts so great as to militate against commercial success, and thirdly, that the maximum temperature does not exceed 3583° F. With these bases the air-standard efficiency is 64 per cent in the case of the Carnot cycle (fig. 17), 56 per cent with constant-pressure cycle and limited expansion (fig. 20), and 48 per cent in the case of constant-volume cycle and limited expansion (fig. 21).

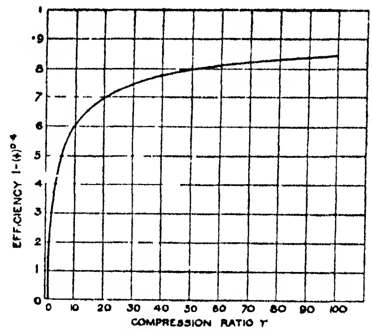

Fig. 19.—Theoretical Thermal Efficiencies for different Compression Ratios in the three Symmetrical Cycles

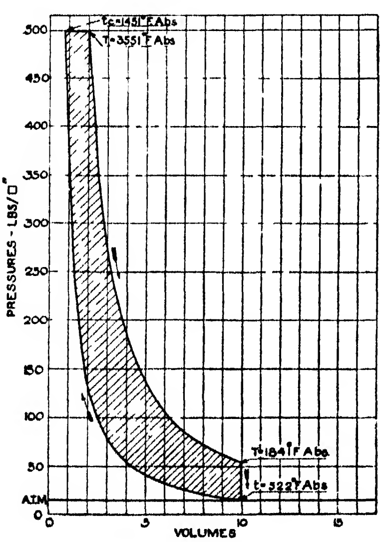

Fig. 20.—Diagram of Air Engine, Constant-pressure Type (Limited Expansion)

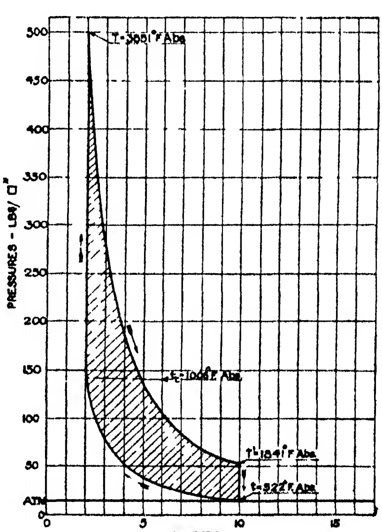

Fig. 21.—Diagram of Air Engine, Constant-volume Type

The mean effective pressure in the cylinder, which represents the amount of work which can be obtained from an engine of unit cylinder volume, is greatest in the case of fig. 20, and least in the case of Carnot cycle, where it is only 6 lb. per square inch, and is therefore quite insufficient to overcome the mechanical and friction losses required to drive an engine working on this principle. The cycle illustrated in fig. 20, which has an efficiency of 56 per cent, falls slightly short of the 64 per cent obtained with figs. 17 and 18, but, on the other hand, gives the very much higher mean effective pressure of 117 as against 56 and 6 lb. per square inch obtained by the constant-pressure cycle (fig. 18) and the Carnot cycle (fig. 17) respectively. For practical reasons, therefore, fig. 20, the constant-pressure cycle with limited expansion, is much to be preferred, and it is upon this cycle, or a slight modification of it, that many oil-engines operate. The curve given in fig. 19 shows the theoretical thermal efficiencies for different compression ratios in the three cycles, the Carnot, the Joule, and the Otto (figs. 17, 18, and 21), the efficiency of each of which equals $1 - \left(\frac{1}{r}\right)^{0 \cdot 4}$, and

TABLE I

AIR-STANDARD DIAGRAMS. FIGS. 17, 18, 20, AND 21 COMPARED

| | 17 | 18 | 20 | 21 |
|---|---|---|---|---|
| Air-standard efficiency .. .. .. .. | 0·64 | 0·64 | 0·56 | 0·48 |
| Mean pressure, pounds per square inch .. | 6 | 56 | 117 | 105 |
| Maximum pressure, pounds per square inch .. | 500 | 500 | 500 | 500 |
| Ratio, $\frac{\text{Maximum pressure}}{\text{Mean pressure}}$ .. .. .. | 83·0 | 8·7 | 4·3 | 4·8 |

shows clearly that little is to be gained by carrying the compression ratio above 20. Working on the constant-volume cycle engines (fig. 21), the value $r = 7$ is the highest in ordinary use, and with Diesel engines 12 to 14 as a compression ratio is not exceeded, primarily for practical reasons later to be explained, although this curve shows that any theoretical gain to be achieved by adopting a figure higher than 14 would be relatively slight. Comparing these cycles from the point of view of the ratio of the maximum to the mean pressure, which, after all, is a factor of prime practical importance, since the construction of an engine to carry a very high maximum pressure where the mean pressure is only a small fraction of such maximum is obviously costly and heavy, Table I above shows at a glance the leading characteristics of these principal cycles of operation. It is not difficult to find variations in practice from these standard cycles; indeed, it can be said that the tendency of the present time with oil-engines favours a cycle of operation where the fuel is injected into the cylinder at an earlier point than the top dead centre of the revolution, so that on the compression curve the engine somewhat favours the constant-volume type of cycle (fig. 21). Injection of the fuel is carried on past the top dead centre where the constant-pressure type of diagram is followed (see fig. 20).

For an analysis of the thermodynamic changes when combining these two characteristic diagrams, reference should be made to the paper "Thermodynamic Cycles of Internal Combustion Engines", by William J. Walker, Ph.D., read before the Institution of Mechanical Engineers, 17th December, 1920, where the advantages of this type of indicator diagram are very fully elucidated. For a considerable number of years past it has been known to practical engineers that such combustion as combines constant volume and constant pressure burning gives actually in practice the best results, and the treatise aforementioned gives the best published explanation of the particular thermodynamical gains. The pure Diesel indicator card will now, however, be examined in detail.

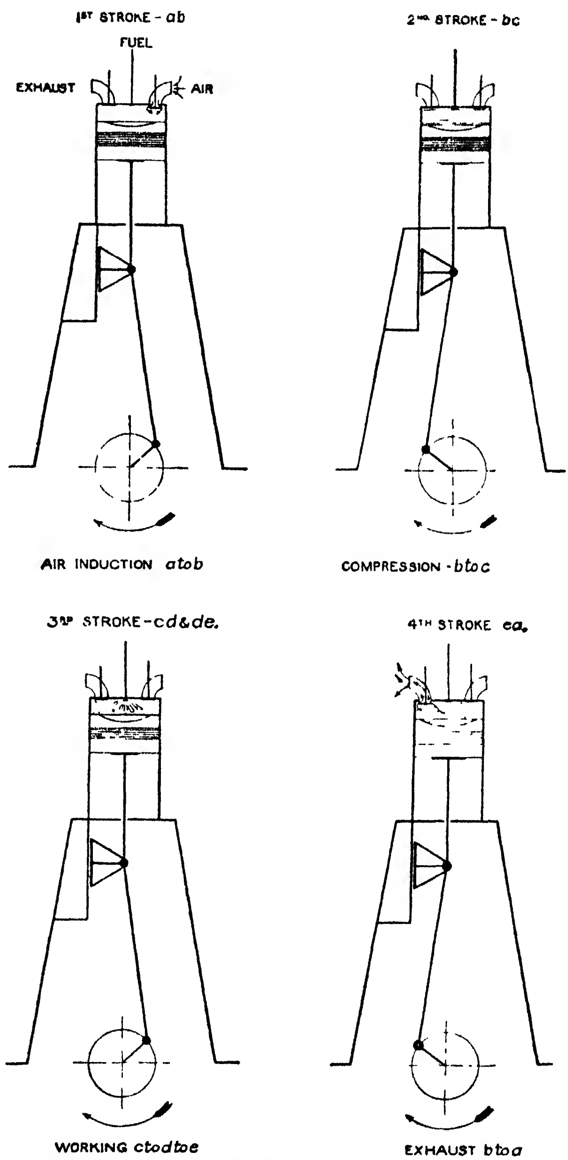


Fig. 22.—Sequence of Operations, 4-Stroke Cycle Diesel Engine

**Diesel Cycles.**—The diagrams (fig. 22) show the phases of the Diesel cycle. The engine is an ordinary single-acting reciprocator. During the first, the downward, stroke one valve in the cylinder head is opened, and the downward moving piston draws in pure air. At the bottom, the end of this stroke, this valve is closed, and the upward moving piston compresses in the cylinder the entrapped air to such a pressure and resulting temperature that the fuel on being injected into this air is ignited. Expansion follows, and the piston is driven downwards doing work. This is the one working stroke of the four which go to make up the cycle. The next, the upward, stroke serves to drive out the products of combustion through the exhaust valves in the cylinder head positively opened at about the bottom dead centre. The four strokes are as follows:

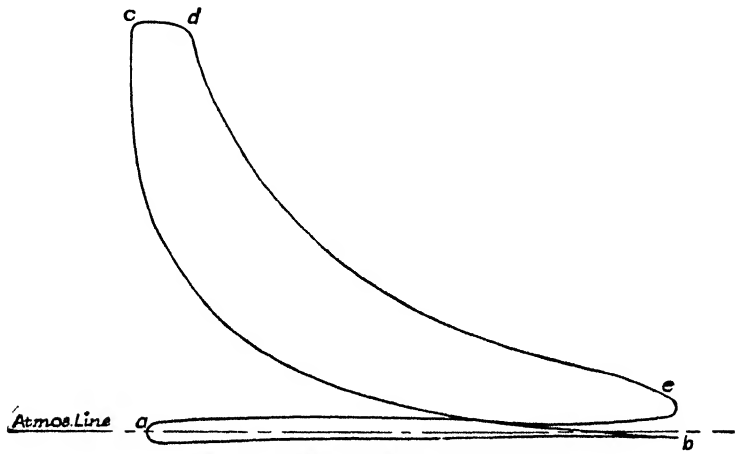


Fig. 23.—Indicator Diagram for 4-Stroke Cycle

1. Suction—filling the cylinder with air—down stroke.
2. Compression—of the air drawn in—up stroke. 1 revolution (first).
3. Working—injection of fuel, ignition, combustion and expansion power—down stroke.
4. Exhaust—emptying the cylinder of the gaseous products of combustion—upstroke. 1 revolution (second).

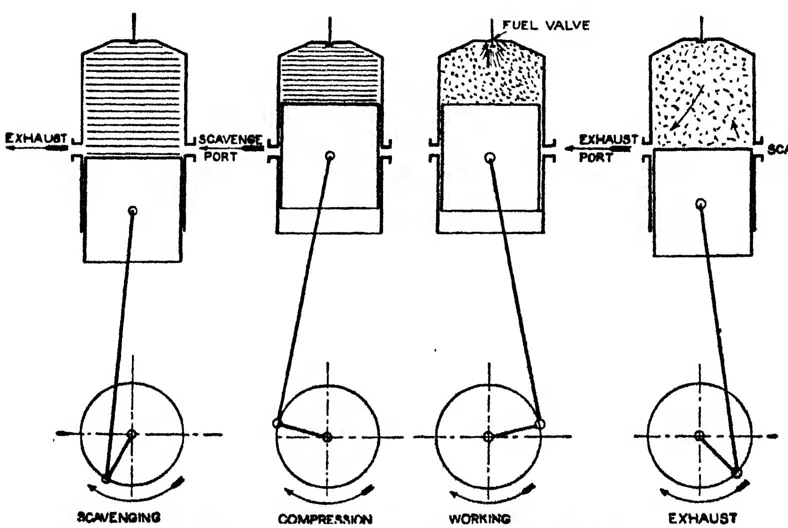


Fig. 24.—Sequence of Operations, 2-Stroke Cycle Diesel Engine

The corresponding indicator diagram, representing the pressure changes on the cylinder during this cycle, is given in fig. 23. The two-stroke cycle is a modification of the standard cycle, and one revolution or two strokes are made to serve for the same functions (figs. 24 and 25). The working or expansion stroke (3 above) is shortened, and towards the end the gases escape through ports, and immediately the cylinder pressure is reduced by the escape of the products of combustion, fresh air previously compressed in a separate pump is introduced through ports and serves to recharge the cylinder. In 120° are carried out the functions of exhausting and recharging, which require 360° with the standard four-stroke cycle. The efficiency of charging is reduced, as is, of course, the working stroke, so that with the two-stroke cycle very far short of double the power of the four-stroke cycle is achieved. If the gain is as much as 30 to 40 per cent, this is a maximum in general practice.

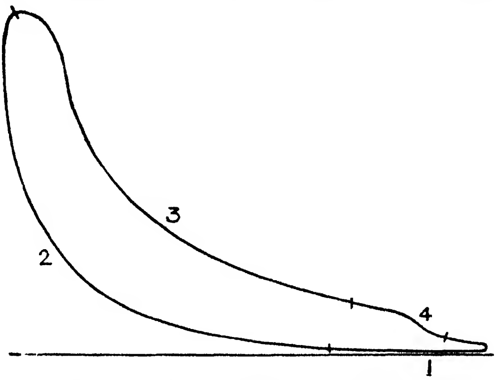

Fig. 25.—Indicator Diagram for 2-Stroke Cycle

Engines of the two-stroke cycle type, using end-to-end scavenging, have now come into use, allowing more efficient air charging to be obtained, with consequent increase in power per unit of cylinder area.

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## CHAPTER III

# Efficiencies

The claim made by the oil-engine to-day is that of being the most efficient known prime mover. It has gained its footing alike in land, marine-work, and in all the other multifarious duties to which it is now being applied on the basis of its efficient working. This, after all, must be the fundamental test which every innovation shall satisfy. Efficiencies may be regarded from many points of view, and in any case will always be considered in conjunction with the first cost.

Neglecting, however, for the moment the question of first cost, depreciation allowances, upkeep and repairs, the relative prices of different fuels, and considering only efficiency from the scientific view-point, the question arises as to how much fuel is required by an oil-engine to develop 1 b.h.p. for 1 hour. This consumption varies from 0·4 to 0·5 lb. of fuel per b.h.p.

per hour. The calorific value of different qualities of liquid fuel varies from about 15,000 to slightly over 19,000 B.Th.U. per pound, and since 1 h.p. hour equals 1,980,000 ft.-lb. the over-all thermal efficiency at 0·4 consumption of fuel in pounds per b.h.p. per hour is $\frac{1,980,000}{0{\cdot}4 \times 18,000 \times 778} = 35$ per cent. Therefore, approximately one-third of the heat supplied to the engine is converted into useful work. This, then, is the fundamental test of efficiency.

**Mechanical Efficiency.**—A further point concerns mechanical efficiency, that is, the ratio of the work given out at the fly-wheel to that done in the cylinder. Were it possible to design and construct an engine without friction, and were no other losses entailed, obviously these quantities would be equal. The mechanical efficiency, however, can never be unity, and with oil-engines varies generally from 70 to 85 per cent, and even reaches 90 per cent in particular cases. This ratio is expressed $\frac{\text{B.H.P.}}{\text{I.H.P.}}$, and is an important quantity, particularly for those responsible for construction and running, as being a measure of sound construction and good adjustment. The mechanical efficiency, of course, increases with the load upon the engine, since the difference between the indicated horse-power and the brake horse-power, assuming constant speed of running, is practically constant, and is the quantity of power necessary to drive the engine at a constant speed of revolution. Mechanical efficiency depends on speed of rotation; generally the higher the speed the lower the mechanical efficiency. For obvious reasons, the larger the engine the higher the mechanical efficiency, since there are certain parts connected with the valve gear and so forth which have a minimum size irrespective of the power of the engine, and so require a constant power to drive them. To deal with this important subject in some detail the following notes are given:

Mechanical efficiency (the ratio between brake horse-power and indicated horse-power) is affected by the number of auxiliaries driven by the main engine, such as lubricating oil, fuel, cooling water, and other pumps.

1. Except in so far as auxiliaries are concerned, the difference between the indicated and the brake horse-power can be apportioned as follows:

(*a*) Fifty per cent is due to piston and piston-ring friction.
(*b*) Twenty-eight per cent can be attributed to main cylinder pumping losses, suction, exhaust, and scavenging.
(*c*) Twenty-two per cent is allocated to valve gear and bearing friction, &c., including windage losses and other unimportant factors.

2. Piston friction depends primarily on the following factors:

(*a*) The quality of the metal of the liner, the piston, and the piston-rings.
(*b*) The quality of the lubrication. (Certain tests prove that a diminution in viscosity of oil increases the mechanical efficiency.)
(*c*) The clearance between the piston and the cylinder walls has an influence on efficiency.

(*d*) The m.e.p., the compression pressure, and the pressure between the liner and the piston, and the liner and the piston-rings, can probably have a most suitable value for the reduction of friction loss to a minimum.

(*e*) The fit and the condition of the piston-rings.

(*f*) The temperature at which the engine runs will have an effect on the lubrication and clearance; it has been substantially proved that there is a temperature of maximum mechanical efficiency.

3. The suction loss, 28 per cent of the total, is primarily a function of design of valves, valve setting, piston speed and gas speeds, and form of induction pipe.

4. The valve gear and the bearings, responsible for 22 per cent of the total loss, will depend on the design of the engine, the alignment, the efficiency of the lubrication, &c.

In addition to the foregoing, there are records of mechanical efficiency being reduced by increased weight of flywheel.

Generally mechanical efficiency is decreased by increased speed, reduced m.e.p., and by malalignment, &c.

The mechanical efficiency may be affected by the form of the combustion chamber, which may produce undue distortion of the piston under working conditions, although this is probably extremely slight; distortion of the piston being due rather to the condition of the gudgeon-pin bearing than to any other cause.

The mechanical efficiency, assuming a constant m.e.p., is practically unaffected by the size of the engine. In recent years airless injection has been widely adopted in the marine field, thereby eliminating the blast-air compressor, thus giving a corresponding increase in the mechanical efficiency of the engine.

In connection with the above, a large number of records of tests of engines have been investigated from Guldner, Supino, D. Clerk, &c.

Actual test results (Tables II to IV, p. 164) will give further information.

**Efficiency of Combustion.**—With oil-engines this subject is of great importance. It is not inseparable from the over-all thermal efficiency already described, nor is it a subject which is easily capable of detailed analysis apart from the particular deductions drawn from an investigation of the fuel consumption per b.h.p. or per i.h.p. actually developed. It can, however, be stated, as a result of practical experience, that it is possible for an air-injection Diesel engine to run exceedingly well, from a mechanical point of view, to give a clear exhaust, and yet for the efficiency of combustion to be much lower than is desirable. With all internal-combustion engines combustion is not complete at the time the fuel valve in the cylinder head closes, and there is a certain amount of combustion during the expansion stroke.

Two things are requisite for satisfactory combustion, i.e. *Atomization* and *Penetration.* Atomization is the splitting up of the fuel into very fine particles in order to ensure rapid combustion. Combustion, however, will not take place unless each particle is in contact with the oxygen necessary for its combustion, and the cloud of oil particles *must* have sufficient velocity to penetrate well into the combustion chamber. Unfortunately the finer

TABLE II

| Type and Cycle. | Beardmore-Tosi. 4-Cycle Single-acting. | Nobel. 2-Cycle Single-acting. | Still. Combined Steam and Oil. | Sulzer. 2-Cycle Single-acting. | Doxford.* 2-Cycle Opposed Piston. | N. B. Diesel 4-Cycle Single-acting. |
|---|---|---|---|---|---|---|
| Designed b.h.p. .. | 1250 | 1600 | 330 | 1250 | 2600 | 2000 |
| Number of cylinders .. | 6 | 4 | 1 | 4 | 4 | 8 |
| Bore .. .. .. | 620 mm. | 675 mm. | 22 in. | 600 mm. | 580 mm. | 26·5 in. |
| Stroke .. .. | 975 mm. | 920 mm. | 36 in. | 940 mm. | 1160 mm. | 47 in. |
| Trials. | | | | | | |
| Duration of test .. | 100 hours | 24 hours | — | 6 hours | 7 days | 7 days |
| Ind. horse-power .. | 1644 | 2022 | 392 | 1721 | 3050 | 2600 |
| Brake horse-power .. | 1267 | 1635 | 343 | 1245 | 2610 | 2043 |
| R.P.M. .. .. | 123·6 | 107 | 124·3 | 101 | 76·7 | 96 |
| Mean ind. press., lb./square inch.. .. | 98·6 | 93 | 91·25 | 102 | 105 | 103·5 |
| Mech. effic., per cent | 77 | 81 | 87·5 | 72·5 | 85·6 | 78·5 |
| Fuel cons., lb./b.h.p./hr. | 0·41 | 0·405 | 0·375 | 0·42 | 0·441 | 0·42 |
| Temp. exh. gas, deg. F. | 514° F. | 599° F. | — | 470° F. | 572° F. | 650° F |

TABLE III

| Type and Cycle. | Sulzer.* 2 Cycle S.A. | Beardmore-Tosi. 4 CycleD.A. | B. & W.* 2 Cycle, S.A. | Vickers, M.A.N.* 2 Cycle D.A. | Beardmore-Tosi. 4 Cycle S.A. Trunk Piston | Doxford.* 2 Cycle Opposed Piston |
|---|---|---|---|---|---|---|
| Designed b.h.p. .. | 2400 | 850 | 1740 | 4000 | 435 | 2900 |
| No. of cylinders .. | 6 | 3 | 8 | 6 | 6 | 4 |
| Bore .. .. .. | 600 mm., 23·62" | 510mm., 20" | 500 mm., 19·7" | 600 mm., 23·6" | 14½" | 600 mm., 23·6" |
| Stroke .. .. .. | 1040 mm., 40·95" | 620 mm., 24·4" | 900 mm., 35·4" | 900 mm., 35·4" | 16" | 1340mm.52·7" 980mm.,38·6" |
| Trials. | | | | | | |
| Duration of test .. | 24 hours | 6 hours | — | — | 24 hours | — |
| Ind. horse-power .. | 2930 | 1183 | 2150 | 4530 | 580 | 3279 |
| Brake horse-power .. | 2415 | 881 | 1740 | 4052 | 435 | 2945 |
| R.P.M. .. .. | 135·6 | 244 | 106 | 130·9 | 350 | 93·2 |
| Mean ind. press., lb./square inch .. .. | 79·5 | 86·6 | 93·2 | 77 | 88·3 | 88 |
| Mech. effic., per cent .. | 82·4 | 74·4 | 81 | 89·4 | 75 | 89·6 |
| Fuel cons., lb./b.h.p./hr. | 0·36 | 0·43 | 0·397 | 0·361 | 0·42 | 0·323 |
| Temp. exh. gas, deg. F. | — | 690° F. | 490° F. | 451° F. | 530° F. | 705° F. |

TABLE IV

| Type and Cycle. | McIntosh & Seymore. 4-Cycle S.A. | Sulzer. 2-Cycle S.A. | Doxford.* 2-Cycle Opposed Piston | Scott Still.* Combined Steam and Oil | Worthington. 2-Cycle D.A. | M.A.N. 2-Cycle D.A. |
|---|---|---|---|---|---|---|
| Designed b.h.p. .. | 2700 | 3600 | 2900 | 1250 | 2900 | 4460 |
| Number of cylinders .. | 6 | 6 | 4 | 4 | 4 | 6 |
| Bore .. .. .. | 32 in. | 30 in. | 22·83 in. | 22 in. | 28 in. | 27·5 in. |
| Stroke .. .. | 60 in. | 53 in. | 45·67 in. | 36 in. | 40 in. | 47·2 in. |
| Trials. | | | | | | |
| Duration of test .. | 30 days | — | — | — | 30 days | — |
| Ind. horse-power .. | 3334 | 4721 | 4190 | 1553 | 4038 | 5320 |
| Brake horse-power .. | 2712 | 3689 | 3630 | 1415 | 2927 | 4460 |
| R.P.M. .. .. | 95·4 | 91·93 | 94·6 | 135·7 | 95·4 | 83·5 |
| Mean ind. press., lb./square inch.. .. | 95·6 | 90·5 | 117 | — | 89·5 | 79·0 |
| Mech. effic., per cent | 81·4 | 78·3 | 86·8 | 91 | 72·5 | 84 |
| Fuel cons., lb./b.h.p./hr. | 0·43 | 0·406 | 0·405 | 0·354 | 0·473 | 0·4 |
| Temp. exh. gas, deg. F. | 718° F. | 432° F. | 551° F. | 296° F. | 548° F. | 520° F. |

Engines marked * are airless injection

the atomization is the less the penetration will be with a given injection pressure, so that the mixing must be assisted by giving the air a whirling or turbulent motion in the cylinder. By suitable arrangement of inlet valves or ports and by correct disposition of spray very successful results have been achieved in modern engines.

Other efficiencies, such as the volumetric efficiency with four-cycle engines, and the scavenging efficiency with two-cycle engines, are concerned with the proportion of fresh air and exhaust gases which remain in the cylinder on the completion of a cycle of operation. With four-cycle engines high volumetric efficiency can only be achieved by suitably proportioning the port areas, the area through the inlet and exhaust valves, to the piston speed at which the engine is run. A certain gain may accrue through suitably proportioning the length and diameter of exhaust and induction pipes to give momentum effect both on the exhaust and suction strokes. In the case of 4-cycle engines, augmented induction or so-called super-charging has been recently adopted as a means of increasing the output for a given size of cylinder. This consists in supplying the induction air to the engine under pressure, thereby ensuring that the air quantity in the cylinder is augmented and so can consume the additional fuel required for the increased power.

**Scavenging.**—With two-cycle engines efficiency of scavenging equally concerns design of ports, pipes, and passages in respect of areas, bends, and so forth, both for the scavenging air and exhaust gases, and is also concerned with the capacity of the scavenging pumps. At first the scavenging pumps were given a volume of 1·4 to 1·5 times the volume of the cylinders which they supply; subsequently, when this was found to be insufficient, the volume was increased to 1·7 to 1·8 times the volume of the cylinders.

Later improvements, however, in the means of introducing scavenging air into the cylinders have enabled the volume to be cut down to 1·3 to 1·4, so reducing the negative work which has to be done in the scavenging pumps, although the efficiency of charging in terms of the main engine-cylinder volume is considerably lower than with the four-stroke cycle. With some designs of two-cycle engines this disadvantage has been overcome by adopting "end to end" scavenging, which ensures a higher efficiency of charging and therefore allows a greater output to be obtained for a given size of cylinder.

**Comparison of the Two- and Four-stroke Cycles.**—It will not be inapt to compare the two cycles of operation. With the advent of the Diesel engine, it was earlier hoped that the true medium for the fulfilment of the hopes of the two-stroke cycle enthusiasts had arrived, since with the Diesel engine air alone was the charge, and the fuel was not introduced into the cylinder until compression was practically complete. The difficulties of pre-ignition with gas-engines and loss of charge through the exhaust ports could not arise.

For this reason the development of the two-stroke cycle Diesel engine has been energetically pursued, and there is every probability of this type of engine being the most suitable for the increasing size of units now in demand.

Designers are now giving close attention to the two-cycle engine, and it is probable that, with improvements in the scavenging arrangements, utilization of supercharging and highly efficient turbo blowers for the supply of scavenging air, this type of engine will finally replace the standard four-cycle engine.

In practice the gains of having one injection of fuel or one power stroke in every two as against every four are much more elusive than at first sight appears. The losses of

(*a*) shorter working stroke, due to exhaust ports,

(*b*) separate scavenging pumps,

(*c*) inefficiency of charging,

are considerable.

Furthermore, in the discussion of heat transfer and heat transmission on pages 170, 171, and 172, reasons are given why the full quantity of air trapped within the cylinder cannot be used, or, in other words, why the maximum quantity of fuel, if injected, would cause heat stresses in the metal which are inadvisable.

The diagram (fig. 26) shows the heat-balance sheet of Diesel and steam plants, and indicates the much higher thermal efficiency of the Diesel over the average steam.

**Heat-balance Diagram.**—Analysing the steam plant, it is seen that of the 100 per cent available heat only 11·5 per cent is actually delivered as brake horse-power, whereas the Diesel engine delivers 34 per cent. In this comparison, it must be understood that comparison is being made between Diesel and steam plants of approximately the same power. Large steam-turbine installations have, of course, a much higher efficiency than 11·5 per cent over-all. Of the various losses with the Diesel system, the engine exhaust (23 per cent) and the engine cooling water (28 per cent) are the greatest, in addition to which there is radiation (4·0 per cent), friction (8·5 per cent), and air compressors (2·5 per cent).

In connection with the "Still" engine (see figs. 54, p. 203, and 87, p. 257), reference to fig. 26 shows that part of the engine cooling water and engine exhaust may be recovered, and of this combined 51 per cent, 29 can be regarded as available for steam raising, although the actual steam so raised only increases the final 34 per cent available as brake horse-power by a slight fraction. See the results of the Still engine tests given in Table II (p. 164). Reference should also be made to these diagrams when considering the heat-transfer question, because a certain considerable proportion of the 51 per cent heat lost to the cooling water and the exhaust must pass through the metal walls surrounding the combustion space.

**Calculations of Power Output.**—Consideration may next be given to the calculations for the size of engine to fulfil a given duty. There are four fundamentals:

(*a*) The cycle of operations of the engine.

(*b*) The number of cylinders in which it is desired to develop the power.

(*c*) The mean pressure in the cylinders.

(*d*) The speed of revolution.

When answers are given to the foregoing, substitution in the formula $\frac{P \times L \times A \times N}{33{,}000}$ = horse-power gives the solution for any one unknown. P represents the mean pressure of the whole cycle in pounds per square inch, L the length of the stroke of the piston in feet, A the area of the piston in square inches, and N the number of power strokes per minute.

This formula gives the horse-power for one cylinder. For four-stroke cycle engines N = half the number of revolutions per minute, for two-cycle engines N = the revolutions per minute, and for double-acting engines the values of N are double that for single-acting of the same cycle.

To take first (*a*), the cycle of operations of the engine; this will be decided by questions of policy, and examples

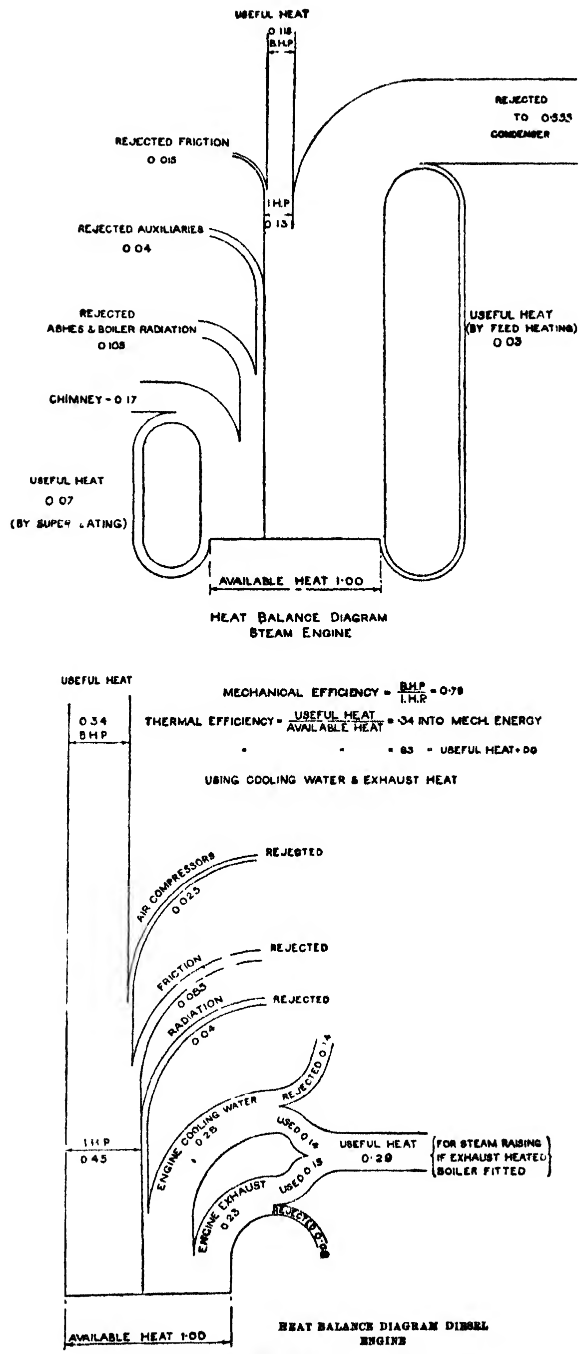


Fig. 26.—Heat-balance Diagrams

of cylinders working on the two- and four-stroke cycles will be given. In regard to (*b*), the question of the number of cylinders, the considerations are (i) ease of starting, (ii) evenness of turning moment, (iii) cost and weight.

To consider the starting problem: a marine engine must be able to start instantaneously in any crank position, i.e. it is quite inadmissible to bar or turn the engine by turning-gear into the requisite position prior to starting. Therefore with the normal system, whereby starting-air is introduced into the working cylinders during the starting period (see fig. 28), 4 cylinders are a minimum for two-stroke cycle engines, and 6 cylinders with four-stroke cycle engines. There have been cases, as with some "Tosi" and "Polar" designs, where starting is carried out on cranks separate from the main cranks, i.e. cranks serving to drive the main compressors or scavenging pumps, when a lesser number of cylinders may suffice, as, for instance, 3 two-cycle and 3 or 4 four-cycle cylinders, whilst still enabling the engine to be started in any position.

For land-work the starting question has only a secondary importance, as the engine can always be barred round by the turning-gear, either hand-operated in the case of small engines, or power-driven with larger plants.

The largest single-cylinder engine seldom exceeds 150 to 210 b.h.p. and even this is somewhat exceptional, because of weight and cost and the necessity for a large fly-wheel to give the necessary evenness of turning moment.

With all internal-combustion engines of a given type the higher the power per cylinder the greater the weight per unit power developed. The piston speed and the mean effective pressure are relatively constant, so that the power formula for approximation can be stated as horse-power = constant × area of piston. The power increases as the area or square, but the weight and cost will increase as the volume or cube. In practice this law is modified to some extent by the fact that the larger the engine the higher the piston speed generally adopted. Nevertheless, the law holds good and favours a multiplication of cylinders. On the other hand, standardization and a natural desire to limit the number of stock sizes and patterns often plays an important part in deciding the number of cylinders. For instance, if the standard sizes of a manufacturer are of 35, 50, 80, 120, and 200 b.h.p. per cylinder, and an engine of 250 b.h.p. is required, the various alternatives are:

(1) Two cylinders of 120 b.h.p., slightly increased rating or speeding up.
(2) Three cylinders of 80 b.h.p.
(3) Six cylinders of 50 b.h.p., with reduced rating or speed of revolution.

Alternatives (2) or (3) will be adopted according to whether the lower speed of the 80 b.h.p. unit or the higher speed of the 50 b.h.p. is the more suitable, whether the engine must be able to start in all positions of the crank (i.e. as in marine practice), whether an exceedingly even turning moment which would favour 6 cylinders is required. If the load is consistently heavy and approximating to full load, 6 × 50, giving 300 b.h.p., will give a margin which is never undesirable if it is possible. This example is typical.

The next item in importance is mean effective pressure. On the test-bed very high results with both two- and four-stroke cycle engines have been achieved, and 140 lb. per square inch has been recorded in a number of tests. For marine work and four-stroke cycle engines for consistently maintaining full power the standard practice is 95 lb. per square inch, in actual service, with large engines. If an engine is only required to develop full power infrequently and for short periods of time, or if the engine is a small one, full power can be said to be 100 lb. per square inch mean effective pressure. For two-cycle engines the figures on the same basis are approximately 80 and 85 lb. per square inch respectively. In the case of 4-cycle engines mean effective pressures up to 20 per cent above the figures quoted can be carried continuously when "supercharging" is adopted.

The question of mean effective pressure is not altogether independent of the consideration of the speed of revolution. A large mean effective pressure means a large quantity of fuel per cycle, and a high speed of revolution involves a large quantity of fuel per minute. Therefore the heat that must pass through the metal walls surrounding the combustion chamber is increased, with higher temperature stresses in the metal (see pp. 170, 171, and 172).

The final factor, speed of revolution, is governed by the desired piston speed foreseen for the Diesel engine and the requirements of the driven machine. The piston speed varies according to the size of the engine, from 500 to 900 ft. per minute. For special applications, such as submarines, speeds up to 1200 ft. per minute have been successfully adopted.

The actual cylinder sizes can now be considered to meet various cases, but first the ratio of stroke to diameter may be discussed. This may vary from 1 : 1 to 2 : 1, the former applying to high-speed engines and the latter to slow and high-powered engines. With certain forms of scavenging arrangements this ratio is governed by the fact that an efficient sweep out of the exhaust gases and recharging with fresh air demand a certain configuration of the cylinder with a stroke-bore ratio of not more than 1·5 to 1.

To take certain specific examples:

(1) A submarine engine, four-cycle, single-acting, of 1500 b.h.p. total. The limitations of space with this type of craft demand at least 8 cylinders. A piston speed of 1200 ft. per minute is decided upon as satisfactory with light trunk pistons. A stroke-bore ratio of 1·275 to 1 gives a reasonably good combustion chamber. The equation

$$\text{B.H.P.} = \frac{\text{PLAN}}{33{,}000} \text{ becomes } \frac{1500}{8} = \frac{100 \times A \times 1200}{33{,}000 \times 4}.$$

L and N can be combined, using the piston speed in feet per minute which equals the revolutions per minute × stroke in feet × 2; as it is a four-cycle engine, the revolutions have to be divided by 2.

$$\text{Hence } 2 \times \frac{\text{stroke} \times \text{revolution}}{2} = \frac{\text{piston speed}}{2}.$$

Therefore L × N in the formula becomes $\frac{\text{piston speed}}{4}$.

By calculation A = 206, so that the cylinder diameter $= \sqrt{\frac{206}{0 \cdot 7854}}$ = 16·3 in., say 16½ in.

The stroke becomes 16½ in. × 1·275 = 21 in.

$$\text{The speed of revolution is } \frac{1200}{\frac{21}{12} \times 2} = 340 \text{ r.p.m.}$$

Obviously other sizes can be found to satisfy the original specification, and if 340 r.p.m. is considered too high for the propeller, or there is available space for an engine with larger cylinders, 300 r.p.m. may be tried, and cylinder sizes of approximately 18¼ by 23½ in. will meet the case, the piston speed remaining the same.

Similarly with a slow-speed four-cycle engine: (2) marine engine, four-cycle, 120 r.p.m., 1200 b.h.p. Six cylinders is the minimum, and gives a suitable engine. A piston speed of about 770 ft. per minute at 120 r.p.m. will be adopted, giving a stroke 39½ in. Taking a m.e.p. of 95 lb. per square inch, the diameter becomes 24 in., and the ratio stroke to diameter of 1·57 is suitable for a slow-speed engine.

These examples will serve to show the methods adopted.

**Heat Transfer.**—When discussing the heat balance sheet of the Diesel engine (see fig. 26), it was shown that of every pound of fuel burnt 55 per cent is lost either to the exhaust or to the water-jackets surrounding the combustion chamber. Part of the heat of the exhaust is further generally given up to water-jackets in the exhaust line and in the silencers.

To split up this quantity of heat, it may very well be assumed that 10 per cent of this heat of the fuel passes through the liner of the engine. Furthermore, the ratio of maximum to mean heat-flow may be taken as 2·5 to 1. This figure may be low, as may also the 10 per cent, but it cannot be too high. If the consumption of fuel oil of the engine is 0·425 lb. per b.h.p. per hour, the heat-flow becomes $0 \cdot 425 \times 18{,}000 \times \frac{10}{100} \times 2 \cdot 5$ = 1920 B.Th.U. per brake horse-power, assuming a calorific value of the fuel of 18,000 B.Th.U. per pound.

With an average piston speed a fair allowance of liner surface exposed to the combustion gases per brake horse-power is given by 9 b.h.p. per square foot.

The temperature drop to convey this heat-flow through a wall of thickness $t$ is

$$T_1 - T_2 = \frac{Q \times t}{K},$$

where $T_1$ and $T_2$ are the temperatures in degrees Fahrenheit of the inside and outside of the wall to be considered, Q the quantity of heat in British thermal units, $t$ the thickness in feet, and K the conductivity of the material under consideration.

Substituting, $T_1 - T_2 = \frac{1920 \times 9 \times \frac{1}{12}}{26} = 55^\circ$ F.

Calculation shows that a temperature gradient of 55° F. per inch is equivalent to a stress of 2000 lb. per square inch.

As a matter of interest, the case of a boiler furnace may be considered. In this instance it is somewhat more simple to imagine the heat flowing from the combustion of the fuel to the water through the steel furnace. Assuming an evaporation from the furnace crown of 15 lb. of water per hour per square foot of heating furnace: the number of British thermal units required to raise 1 lb. of water from a feed temperature of 191° F. to 391° F. (the temperature corresponding to a pressure of 200 lb. per square inch) is $200 + 839 = 1039$. Therefore the British thermal units per square foot of furnace $= 1039 \times 15 = 15{,}600$ B.Th.U. per square foot. The temperature drop to convey this heat-flow through a furnace shell of $\frac{5}{8}$ in. thickness, using the same formula as before, is

$$T_1 - T_2 = \frac{Q \times t}{K} = \frac{15{,}600 \times \frac{0{\cdot}625}{12}}{26} = 31^\circ \text{ F.}$$

31° F. of a temperature drop in $\frac{5}{8}$ of an inch is equal approximately to 50° F. drop per inch, and a resulting stress of slightly more than 3000 lb. per square inch.

It will thus be seen that the temperature drop and the stress in the oil-engine cylinder liner and the boiler furnace are of the same order, and this will give a clear idea of the nature of the heat problems.

To examine the concrete cases of the cylinder heads of oil-engines, making exactly the same assumptions as were done with the cylinder liner and basing these calculations on the figures given in Table I:

(*a*) Four-cycle single-acting engine of 1200 b.h.p.
Fuel consumed per hour per square inch of piston area = 0·181 lb.
Temperature gradient per inch in liner, 61° F.
Stress due to temperature drop, liner 2·25 in. thick, is 4750 lb. per square inch.
Temperature gradient per inch in cylinder cover, 151° F.
Stress due to this temperature gradient cover of cast iron 2 in. thick, 10,500 lb. per square inch.

(*b*) Two-cycle single-acting engines of 1200 b.h.p.
Fuel consumed per hour per square inch of piston area, 0·315 lb.
Temperature gradient per inch in liner, 95·5° F.
Stress due to temperature gradient, liner assumed to be 1·75 in. thick, is 5800 lb. per square inch.
Temperature gradient per inch in cylinder head, 261° F.
Stress due to this temperature gradient, head of cast iron assumed to be 1·625 in. thick, 14,900 lb. per square inch.

Actually, the temperature gradient in the metal may be appreciably lower owing to the resistance of a gas film adhering to the metal, so that the actual temperature stresses will be correspondingly lower.

## CHAPTER IV

# Résumé of Power and Efficiency Formulæ

**Mean Effective Pressure.**—The mean effective pressure, $p$, in pounds per square inch, on the piston throughout its working stroke, is the average height of the indicator diagram (see fig. 23) as measured by the pressure scale of the diagram; it is readily ascertained by aid of an ordinary planimeter.

**Indicated Horse-power.**—During each *working* stroke an average of $p$ lb. per square inch acts over the piston whose diameter is $d$ in.; the total mean effective force on the piston is therefore $\frac{\pi}{4}d^2p$ lb.-wt., and this acts throughout the stroke of $s$ in.; accordingly the indicated work done per cylinder per working stroke is $\frac{\pi}{4}d^2p \times \frac{s}{12}$ ft.-lb.

If the engine be of the single-acting four-stroke type and runs at $n$ revolutions per minute, then there are $\frac{n}{2}$ working strokes per minute; thus the indicated work done per minute is $\frac{\pi}{4}d^2p \times \frac{s}{12} \times \frac{n}{2}$ ft.-lb. per cylinder.

If the engine has N cylinders, the total indicated work per minute, assuming them all equal, is evidently $\frac{\pi}{4}d^2p \times \frac{s}{12} \times \frac{n}{2} \times N$ ft.-lb. But a rate of working of 33,000 ft.-lb. per minute is, by definition, 1 h.p.

Finally then, the indicated horse-power of the engine is expressed by

$\frac{1}{33,000}$ of this expression; and this reduced by ordinary arithmetic becomes

$$\text{I.H.P.} = \frac{1}{1000000} d^2psnN \quad \ldots\ldots\ldots\ldots\ldots (1)^*$$

for a single-acting four-stroke engine with N cylinders.

**Brake Horse-power.**—Brake horse-power, or shaft horse-power, or flywheel horse-power, as it is variously termed, is the actual power available at the crank-shaft coupling, and is less than the indicated horse-power by the power expended in overcoming the frictional and other resistances of the engine itself. The ratio of the brake horse-power to the indicated horse-power is termed the mechanical efficiency of the engine, and is usually denoted by $\eta$. Thus

$$\eta = \frac{\text{B.H.P.}}{\text{I.H.P.}} \ldots\ldots\ldots\ldots\ldots\ldots\ldots (2)$$

Combining equations (1) and (2) we get

$$\text{B.H.P.} = \frac{1}{1000000} d^2\eta psnN \ldots\ldots\ldots\ldots\ldots (3)$$

for a single-acting four-stroke engine with N cylinders. The product $\eta p$ is termed the brake mean effective pressure.

**Piston Speed.**—In 1 min. each piston makes $2n$ strokes each of $s$ inches; thus the piston speed in feet per minute is $2n \times \frac{s}{12}$, i.e. $\frac{ns}{6}$. This is usually denoted by $\sigma$; so that

$$\sigma = \frac{ns}{6} \text{ ft. per minute.} \ldots\ldots\ldots\ldots\ldots\ldots (4)$$

**Two-stroke and Double-acting Engines.**—In single-acting two-stroke engines and in double-acting four-stroke engines there is one working impulse per revolution per cylinder. In double-acting two-stroke engines there are two working impulses per revolution per cylinder. On account of these differences in type it is convenient to write equations (1) and (3) in the form

$$\text{I.H.P.} = \frac{1}{\beta} d^2psnN, \ldots\ldots\ldots\ldots\ldots\ldots (1')$$

and

$$\text{B.H.P.} = \frac{1}{\beta} d^2\eta psnN, \ldots\ldots\ldots\ldots\ldots (3')$$

where $\beta$ is a constant having the following values:

| Type of Engine. | Value of $\beta$. |
|---|---|
| For single-acting four-stroke engines .. .. | 1000000 |
| For double-acting four-stroke, and single-acting two-stroke engines .. .. .. | 500000 |
| For double-acting two-stroke engines .. .. | 250000 |

* More accurately, I.H.P. $= 992\, d^2psnN \times 10^{-9}$, but since the value of $p$ is rarely obtainable within 1 per cent, the numerical factor may be rounded off.

A further point to be noted is that in double-acting Diesel engines the piston rod diminishes the effective area of the lower surface of the piston. If $\delta$ denote the piston rod diameter in inches, then for such cases in place of $d^2$ in all formulæ herein the expression $(d^2 - \frac{1}{2}\delta^2)$ must be used.

**Torque.**—If a force of P lb., acting always at right angles at the end of an arm of length $l$ ft., turns the arm through one revolution, then the work done is $P \times 2\pi l$ ft.-lb.

The product $Pl$ is termed the "torque", or "turning effort", or "turning couple", and is expressed in pound-feet; it is usually denoted by T. Thus the work done per revolution by a torque of T lb.-ft. is $2\pi T$ ft.-lb. If $n$ revolutions per minute are made, the work done per minute is $2\pi Tn$ ft.-lb., and the horse-power is accordingly $\frac{2\pi Tn}{33,000}$. But, by equation (3′), the brake horse-power is also $\frac{1}{\beta} \cdot d^2\eta psnN$. Equating these two expressions we obtain, on reduction:

$$T = \frac{5252 \cdot d^2 sN}{\beta} \times \eta p \text{ lb.-ft.} \qquad \ldots\ldots\ldots\ldots\ldots(5)$$

Obviously T is here the mean or average crank-shaft torque of the engine; the actual torque fluctuates in value periodically in a manner dependent on the nature of the cycle and the number of cylinders of the engine.

It will be noted from this result that as, for the same engine, $d$, $s$, and N are all constant, the mean torque is directly proportional to the brake mean effective pressure, $\eta p$.

Again equation (3′) may be written:

$$\eta p = \frac{\beta}{d^2 sN} \frac{\text{B.H.P.}}{n}, \qquad \ldots\ldots\ldots\ldots\ldots\ldots(6)$$

which shows that, for the same engine, the brake mean effective pressure is directly proportional to the value of the ratio $\frac{\text{B.H.P.}}{n}$.

Lastly, combining equations (5) and (6) gives the simple relation

$$T = 5252 \times \frac{\text{B.H.P.}}{n}, \qquad \ldots\ldots\ldots\ldots\ldots\ldots(7)$$

which shows that, for the same engine, the mean torque is also directly proportional to the value of the ratio $\frac{\text{B.H.P.}}{n}$.

Thus, in any engine, both the brake mean effective pressure and the mean torque are simply proportional to the value of the ratio of the brake horse-power to the revolution speed.

**Determination of Brake Horse-power.**—The brake horse-power of an engine may be determined at various revolution speeds by some form of electrical, torsion, or hydraulic dynamometer, the Froude water dynamometer being a well-known type. By plotting the powers thus obtained against the corresponding revolution speeds the ordinary power-speed chart, fig. 27, is obtained.

From this the value of the ratio $\frac{\text{B.H.P.}}{n}$ at any speed is at once found by measuring off OM and MP from their respective scales, and then taking the ratio $\frac{\text{PM}}{\text{OM}}$. This ratio obviously attains its maximum value at the point where a tangent drawn from the zero origin O meets the curve. From equations (6) and (7) it is clear that this simple procedure enables us to determine the revolution speed and brake horse-power at which the brake mean effective pressure and the mean torque attain their maximum value, and to determine also the numerical values of these two important quantities. And again, by taking a series of values of $\frac{\text{B.H.P.}}{n}$, and evaluating the corresponding $\eta p$ and T from equations (6) and (7), and plotting these against $n$, a curve showing the variation of $\eta p$ and of T with speed is found, the same curve serving for both, with the appropriate different scales for ordinate measurement.

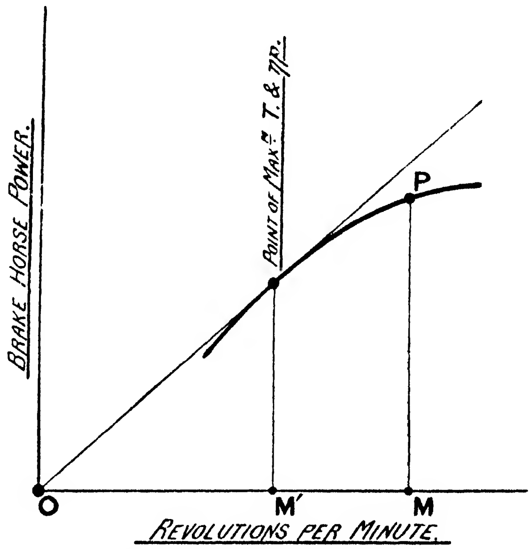


Fig. 27.—Diagram of Power-speed Curve

**Thermal Efficiencies.**—The British thermal unit (B.Th.U.) is the quantity of heat required to raise 1 lb. of water through 1° F.* The mechanical equivalent of 1 B.Th.U. is 778 ft.-lb. of work.

Hence as 1 h.p. is 33,000 ft.-lb. of work per minute, it is also expressed by $\frac{33,000}{778}$, i.e. 42·4 B.Th.U. per minute.

Suppose that by calorimeter test an oil fuel is found on combustion to evolve H B.Th.U. of heat per pound, lower value.† And suppose that, by test, an engine is found to consume $w$ lb. of fuel oil per I.H.P. per hour.

Then $w$H B.Th.U. of heat are expended in obtaining 42·4 × 60

* At, or near, its temperature of maximum density, i.e. 39° F.

† *Lower* value, as the water of combustion in an engine is always exhausted as *steam*.

= 2545 B.Th.U. of indicated mechanical work. Accordingly the absolute indicated thermal efficiency of the engine is expressed by

$$100\epsilon = \text{abs. indic. thermal efficiency} = \frac{2545}{wH} \times 100 \text{ per cent.} \quad (8)$$

For example, a modern Diesel engine uses per I.H.P. hour about 0·35 lb. of fuel oil of (lower) calorific value roundly 18,000 B.Th.U. per pound. The corresponding absolute indicated thermal efficiency from equation (8) is 40·4 per cent.

An important quantity in engine design in connection with the cooling problem is the amount of heat evolved per cubic foot of working stroke swept by the piston. With the largest gas-engines in land installation it is, for example, found necessary in order to avoid overheating to limit the amount to about 35 B.Th.U. per cubic foot of swept volume.

Suppose $h$ B.Th.U. of heat to be evolved per cubic foot of swept volume, i.e. by 1 sq. ft. of piston area describing 1 ft. of stroke, then $778h\epsilon$ ft.-lb. of indicated work are obtained. But $p$ denoting the mean effective pressure on the piston in pounds per square inch, this same amount of indicated work is also expressed by $144p \times 1$, i.e. by $144p$. Accordingly $144p = 778h\epsilon$; and thus

$$h = 0{\cdot}185 \times \frac{p}{\epsilon}. \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots (9)$$

The mean effective pressures adopted in modern large marine Diesel engines range roundly from 90 to 105 lb. per square inch in four-stroke designs, and from 85 to 100 lb. per square inch in two-stroke designs; the corresponding values of $h$ from equation (9) are 41·6 to 48·5 B.Th.U. per cubic foot of swept stroke in four-stroke engines, and from 39·3 to 46·25 in two-stroke engines. The cylinder diameters of these Diesel engines are however, in general, much less than those of large land gas-engines.

**Absolute Brake Thermal Efficiency.**—The mechanical efficiency $\eta$ has already been defined as the ratio $\frac{\text{B.H.P.}}{\text{I.H.P.}}$; hence the absolute brake thermal efficiency of an engine is $\eta\epsilon$. Or, expressed as a percentage, and by aid of equation (8),

$$\text{Percentage absolute brake thermal efficiency} = \frac{2545\eta}{wH} \times 100. \quad (10)$$

(H is the calorific value of the fuel in B.Th.U. per lb.)

The mechanical efficiency $\eta$ of large slow-running marine Diesels is usually considered to range from about 0·69 to 0·73 in two-stroke designs, and from 0·75 to 0·79 in four-stroke cases. Corresponding therefore to an absolute indicated thermal efficiency of 40·4 per cent, the absolute brake thermal efficiency may range from about 29 per cent to 32 per cent.

In many cases in practice, particularly with small-powered engines, the fuel consumption is directly ascertained per b.h.p. hour. In such cases the percentage absolute brake thermal efficiency is immediately given by the expression $\frac{2545}{w'H} \times 100$, where $w'$ is the weight of fuel consumed per *brake* horse-power per hour.

**The "Air Standard".**—Referring to the constant-volume cycle indicator diagram (fig. 21), the volume ratio of compression, $\frac{1C}{0C}$, is usually denoted by $r$, and is a quantity of fundamental importance in design. In the ordinary simple thermodynamic theory of the constant-volume cycle, air alone, of constant specific heat, is considered to be the working substance, and no exchanges of heat with cylinder walls are taken into account. With these simplifying assumptions it is easily shown * that the ideal thermal efficiency of the constant-volume combustion cycle is given by the simple equation:

$$\text{Air standard ideal efficiency} = 1 - \left(\frac{1}{r}\right)^{0.4}. \quad \ldots\ldots\ldots(11)$$

A committee of the Institute of Civil Engineers in 1905 recommended the adoption of this expression to indicate the ideal thermodynamic efficiency of the constant-volume cycle engine, and it is known as the "1905 Air Standard", and has been extensively employed. Notwithstanding the simplifying assumptions involved it has proved in practice very useful, and extended experience has shown that as $r$ is increased, the efficiency of an engine, *cæteris paribus*, also increases, and nearly in the proportion of the air standard formula. In the Diesel engine the compression ratio is high, and has usually a value about 12. For $r = 12$ the air standard value, from equation (10), is 0·63, or 63 per cent. Thus, referring to the example above, it was found that the absolute indicated thermal efficiency was 40·4 per cent; accordingly for this case the indicated thermal efficiency ratio $= \frac{40.4}{63} \times 100 = 64$ per cent. Similarly, for an absolute brake thermal efficiency of 32 per cent the brake thermal efficiency ratio would be $\frac{32}{63} \times 100 = 50$ per cent roundly.

**Ideal Efficiency of the Diesel Cycle.**—The constant-pressure cycle of the Diesel engine is thermodynamically somewhat less efficient than the constant-volume cycle. With the same simplifying assumptions as before it may be shown† that (see fig. 21) if the constant-pressure ratio $\frac{3'C'}{2'C'}$ be denoted by $\rho$, the ideal thermodynamic efficiency is given by

$$\text{Ideal Diesel air cycle efficiency} = 1 - \left(\frac{1}{r}\right)^{0.4} \times \frac{\rho^{1.4} - 1}{1.4(\rho - 1)}. \quad (12)$$

* See e.g. Clerk's *Gas, Petrol, and Oil Engines*, Vol. I, p. 83.

† See e.g. Clerk and Burls' *Gas, Petrol, and Oil Engines*, Vol. II, p. 723.

This has the same value as the air standard efficiency when $\rho = 1$, i.e. when there is no constant pressure combustion at all, in which case the power output of the engine would, of course, be zero: and it diminishes as $\rho$ increases.

In actual Diesel engines the maximum value of $\rho$ may be taken at about 1·75. For $r = 12$ and $\rho = 1·75$, equation (12) gives as the ideal Diesel cycle efficiency the value 0·575, or 57·5 per cent, which is 5·5 per cent less than the air standard for $r = 12$. Thus in strictness the absolute indicated efficiency of 40·4 per cent in the above example should be compared not with the 63 per cent of the air standard, but with the 57·5 per cent of the ideal Diesel standard, and if this be done the relative indicated efficiency has the higher value of $\frac{40·4}{57·5} \times 100 = 70$ per cent roundly.

**Volume of Combustion Chamber.**—It is worthy of note that if C be the volume of the combustion chamber in cubic inches, then as

$$r = \frac{\text{combustion-chamber volume} + \text{swept volume}}{\text{combustion-chamber volume}},$$

we have

$$r = \frac{C + \frac{\pi}{4}d^2 s}{C},$$

and this on reduction becomes

$$C = \frac{\frac{\pi}{4}d^2 s}{r - 1} \text{ c. in.}, \qquad \ldots\ldots\ldots\ldots\ldots\ldots(13)$$

from which C may be immediately determined.

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# CHAPTER V

## Motor Boats, Yachts, Trawlers, &c.

Prior to the great development of railway systems in the first half of the nineteenth century, the wide extent and importance of inland and coastal water transport attracted the attention of engineers to the problem of improving this transport by aid of some form of mechanical propulsion, and as early as 1827 trials were made on the Thames with a 36-foot paddle-boat driven by a Brown " gas-vacuum " engine. Again, a 2-h.p. Lenoir engine (1860) was used to drive a boat that plied for two years between Paris and Charenton, while in America, about 1878, some boats propelled by Brayton engines were in use for a time upon the Hudson river.

None of these early attempts proved successful either technically or commercially, but about 1887 Daimler succeeded completely in driving launches, barges, &c., on the canals of the near Continent by means of his small quick-speed motors as already described.

The United States, with its many beautiful rivers, lakes, and sheltered waters, was quickly to the fore in applying the small petrol or "gasoline" motor to boat propulsion, and the two-stroke two-port Day type of 1891 (see fig. 3) has been very largely used by the Americans on account of its simplicity, reliability, low first cost, and its early successful adoption of electrical ignition. In England its development was retarded owing to the hot-tube method of ignition which prevailed for some years, as with this, accurate timing of the firing was impracticable, and this resulted in these little engines exhibiting a disconcerting habit of sudden reversal.

The two-stroke Day petrol engine, usually of the three-port type (see fig. 4), has long been used with complete success in motor bicycles, and the confidence thus created in the type has resulted in its rapidly extended use in connection with the smallest type of pleasure motor-driven craft, viz. the outboard motor-boat.

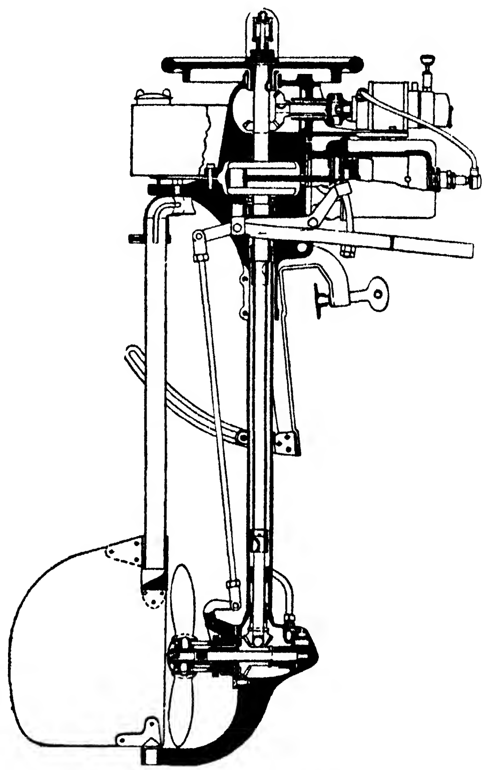


Fig. 28.—Sectional General Arrangement, with Reversing Propeller Gear

**The Outboard Motor.**—The outboard motor is a completely self-contained petrol power unit of small weight, which can readily be attached to the stern of a small boat, thus converting it at will into a motor-boat. Many designs exist, remarkable both for the care and knowledge manifested in their arrangement, and the excellence and strength of their construction.

**The "Watermota" Power Unit.**—Typical of the smallest of outboard motors is that of Messrs. Fair, of which a sectional view is given in fig. 28. The two-stroke three-port motor (see fig. 4) has a single cast-iron cylinder with sheet-copper water jacket, removable for cleaning. The bore is 2·75 in., the stroke 3 in., and normal full speed 900–1000 r.p.m., with an output of 3–4 b.h.p. The combustion chamber is fitted with a compression release tap, by which also the cylinder can be primed with petrol to facilitate starting in cold weather. All bearings are renewable throughout, and no rubber hose piping is used.

The light alloy piston carries a hollow steel case-hardened and ground gudgeon or " wrist " pin. The drop-forged crank-shaft is vertical and is fitted at the top with a cast-iron flywheel; by sharply pulling this round by hand on the rim the motor is started. The whole engine is carefully balanced, and the weight of the flywheel and crank-shaft is carried by a ball thrust bearing under the crank. A vertical tubular downward extension of the crank-shaft drives a horizontal bronze propeller shaft, carried in white-metal bearings, through enclosed steel cut and case-hardened bevel gearing, the two-bladed bronze propeller revolving at two-thirds of the engine speed. At the forward end of the horizontal propeller shaft the tiny gear-wheel type jacket water circulating pump can be seen. From the top of the cylinder jacket the heated water is passed into the upper end of the rudder tube and mingles with the exhaust gases.

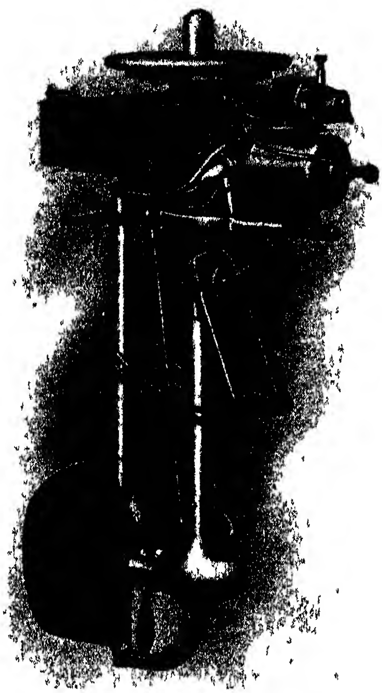


Fig 29.—3-4-h.p Reversible-bladed Propeller Watermota

The waterproof H.T. magneto for ignition, bevel-driven from the crank-shaft, is seen above the cylinder towards the right, while the 1½-gall. fuel tank is situated partly beneath the flywheel on the left. The petrol, mixed with lubricating oil (" Petroil "), is gravity-fed to a float-feed carburettor fitted with both gas throttle and air controls. The exhaust pipe is water-cooled, and the exhaust gases, mixed with the jacket discharge water, pass down the rudder tube, and are expelled silently below water. The entire mechanism is completely encased, the propeller gear-box being provided with a removable water-tight cover. The whole power unit is mounted on a substantial bearing permitting it to be tilted for negotiating shallows and before beaching, and it is secured to the transom of the boat by bronze brackets and stout clamping screws; the brackets have slotted ends, enabling the angle of the motor to be varied to suit that of the transom carrying it.

The illustration shows the reversible-bladed propeller design easily operated by aid of the hand lever shown just below the engine cylinder. By aid of this lever the engine may be started with the propeller in a neutral position; and possessed of three means of controlling speed, viz. by throttle, magneto, and propeller, the boat may be caused to run at any desired rate from zero to full speed in either direction without shock to the engine or mechanism.

An external view of this compact little power unit appears in fig. 29; the several parts will be readily recognized from the description given above. It is used for the propulsion of small craft varying from 12-ft. dinghies to whale-boats of 25 ft. in length.

**The 4–11 h.p. "Speed" "Watermota".**—For speed boats, hydroplanes, or "skimmers" a larger single-cylindered outboard unit is built, also of the two-stroke three-port Day type, with 3-in. bore and 3-in. stroke, developing 4 to 11 h.p. at from 1000 to 4000 r.p.m. On account of the high revolution speed the piston is of aluminium alloy, and the flywheel a rust-proof steel stamping, machined all over. The propeller is non-reversible, and the propeller thrust is taken by a ball bearing. This power unit fitted to a 13-ft. dinghy—the whole weighing but 160 lb.—is capable of a speed of about 23 m. per hour with one person on board, though two or three passengers can be accommodated comfortably. The dinghy, complete, is sold for less than £100; the outboard "speed" motor-boat is thus well within the means of the average owner of a small motor-car.

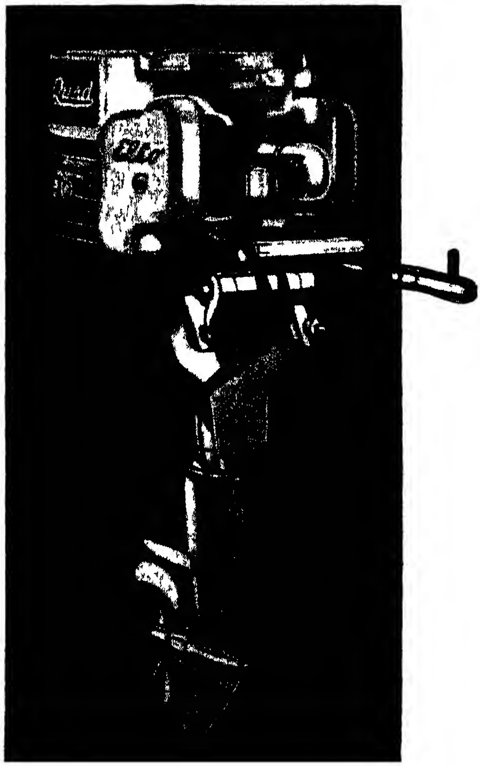


Fig 30—"Elto Quad" Four-cylinder Outboard Motor

**The "Elto Hi-speed Quad".**—Typical of the larger racing type of outboard motor is the Elto Hi-speed Quad, weighing complete about 110 lb., which has a horizontal four-cylinder two-stroke three-port motor of 2·75 in. bore and 2·094 in. stroke running at from 800 to 5500 r.p.m., and developing a maximum of 35 h.p. An external view appears in fig. 30; it will be seen that the arrangement of parts is generally similar to that of the smaller unit previously described.

This little very fast-running engine is fitted with "Ivnite" pistons and connecting-rods, and a very robust three-bearing crank-shaft. The cylinders have high compression with hot spot, water-jacketed heads, and water-jacketed sparking plugs. The ignition is of the Atwater-Kent type with dry battery, and cylinder heads, coils, wiring, and sparking plugs are completely protected from the weather by enclosing aluminium alloy casings. Opposed cylinders fire alternately in pairs. Lubrication is on the "Petroil" system, about 1 pt. of lubricating oil being well mixed with each gallon of petrol prior to its introduction into the fuel tank. The jacket water circulation is maintained by a centrifugal pump carried on the main drive shaft. The propeller has fixed blades, and runs at two-thirds of the engine revolution speed. Starting is effected by means of a disappearing handle on the top of the flywheel. The ignition being retarded and carburettor flooded, a quick flick of the flywheel against the compression starts the engine. To reverse, the ignition is momentarily cut off and the engine slows down; the ignition is next fully advanced, resulting in rising pistons receiving a downward working impulse which brings them momentarily to rest, and immediately thereafter causes them to descend, thus reversing the direction of the crank-shaft rotation The ignition is then immediately altered towards the "ahead retarded" position, which is, of course, towards the astern "advanced", and the engine speeds up in the reverse direction. Speed boats fitted with "Elto Quads" attained speeds of roundly 40 m. per hour in American competitions during 1928.

**Other Outboard Motors.**—Among other examples of this very interesting and increasingly popular type of small marine two-stroke power unit may be mentioned the Johnson outboard motors, built with one, two or four cylinders, with output ranging from 3 to 32 h.p.; and the "Dunelt", whose single-cylindered engine of 2·62-in. bore and 2·81-in. stroke develops 82·5 b.h.p. at 3400 r.p.m., and weighs, complete, but 62 lb. in the Sports type, and 10·6 b.h.p. at 3800 r.p.m. in the Racing type. A view of a 14-ft. two-seater single-stepped hydroplane for fast cruising with the racing type motor is shown in fig. 31.

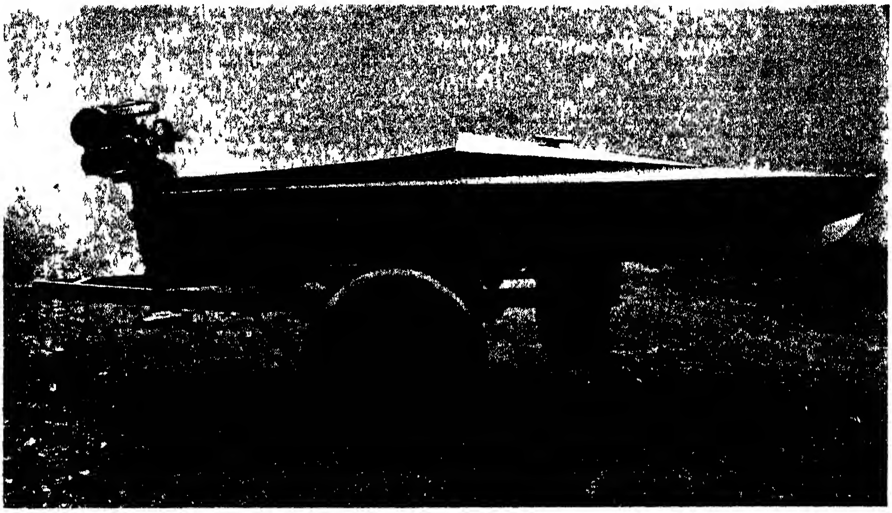
Fig. 31.—Typical 14-ft. Single-stepped 2-Seater Hydroplane

Owing to their lightness and small size, outboard motor-boats are readily transported by road, mounted on specially built trailers which are towed behind an ordinary motor car, as shown in the illustration, fig. 32.

**Inboard Motors.**—In this class are included vessels varying from small open boats to large motor-yachts fitted with engines developing several hundreds of horse-power. The small open boat with cased-in motor fitted amidships is illustrated in fig. 33, which shows the 1929 pattern of 18 ft. × 4½ ft. "Runabout" by Brooke of Lowestoft. This is capable of a speed of 16 m. per hour, and seats four persons, two in each cockpit.

The motor is a totally enclosed robust four-cylinder monobloc four-stroke side-valve petrol-engine of normal type with detachable cylinder heads, having a bore of 2·625 in. and stroke of 4·0 in., and is rated as of 10 h.p. The complete engine weighs about 398 lb., and runs at up to 1500 r.p.m.

An external view of the engine is given in fig. 34, which shows the raised marine-type starting handle by which the crank-shaft is turned by



From *The Motor boat* by permission

Fig 32 —Transporting Outboard Motor-boat by Road

hand by means of an encased chain drive and free wheel; the flywheel, on the left of fig. 34, is also completely encased. Ignition is by H. T. Bosch magneto, and the carburettor is the standard Zenith two-jet type. When desired, an electric self-starter is fitted, as in a car, with dynamo, starting motor, battery, cut-out, switchboard, &c.

Integral with the engine is the reversing gear, also completely enclosed, and operated by the single upright lever shown on the right; this gear,

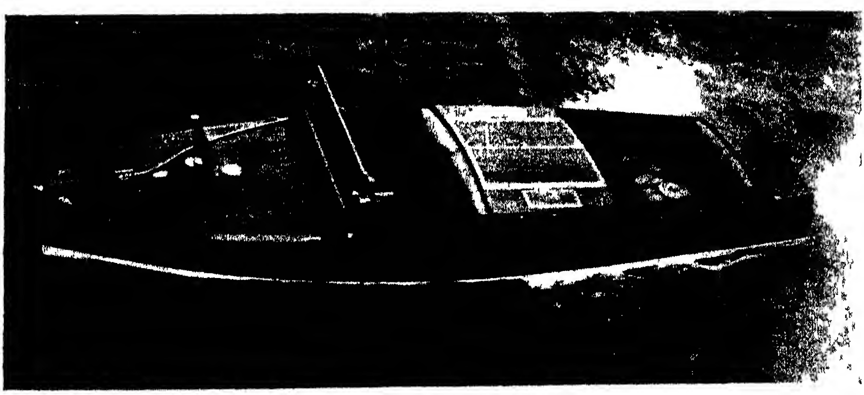


Fig 33 —The Brooke 18-ft. Runabout

of the double clutch and bevels type (see fig. 38), provides a neutral, or free engine, position, and when in reverse the propeller runs at ahead speed. All controls are brought to the port side of the forward cockpit (see fig. 33), and the boat is equipped with a steering wheel and instrument board on motor-car lines, with revolution indicator, clock, oil and petrol gauges, electric light switches, electric horn, starter switch, and throttle-control lever. The fuel tank is housed below the fore deck, and has a capacity of 6 gall., corresponding to about 100 m. running at full speed on petrol. As fuel paraffin may be used if desired, but with this, as explained in Chapter I, there is a loss of power output by the engine of some 15 to 20 per cent. The propeller used in this type of boat is either a three-bladed

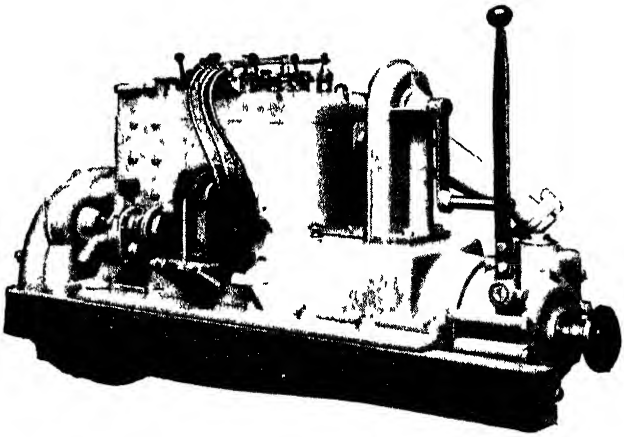


Fig 34—10-h p Marine Motor

13-in. or a two-bladed 16-in. diameter type. This 10-h.p. engine is used for the propulsion of craft varying from the 18-ft. runabout here described, to the 30-ft. twin-screw Brooke "sea-cruiser", and in 30-ft. 10-m.p.h. North-east Coast fishing "cobbles"; and, as an auxiliary, in a 5-m.p.h. 32-ton yawl and a 70-ft. "monkey" barge.

The Brooke marine motors of 1929 included thirteen designs, varying from a single-cylindered 3·375 in. × 3·25 in., 3-h.p. engine, weighing 332 lb., complete with flywheel and reversing gear, to a six-cylindered overhead valve monobloc 4·25 in. × 4·75 in. engine developing 100 b.h.p. at 1850 r.p.m. with petrol as fuel, and weighing 1100 lb. complete with bevel and cone-clutch type reversing gear. With two exceptions, all the Brooke engines can be arranged to run on paraffin as fuel; and with one exception, all can be fitted, when required, with electric self-starting equipment.

A longitudinal line section of the 100-h.p. 24-ft. Brooke "runabout", showing the general arrangement of the whole design, appears in fig. 35. This type is capable of a speed of 30 m.p.h.; its length is 24 ft., breadth 6 ft., and draft 2 ft.; it weighs, complete, about 29 cwt. Easy accommodation is provided for two persons forward, and three in each of the after cock-pits. The inclined position of the motor in the hull, which is frequently necessitated in design, has to be taken into account in the arrangement of the lubricating system of the engine.

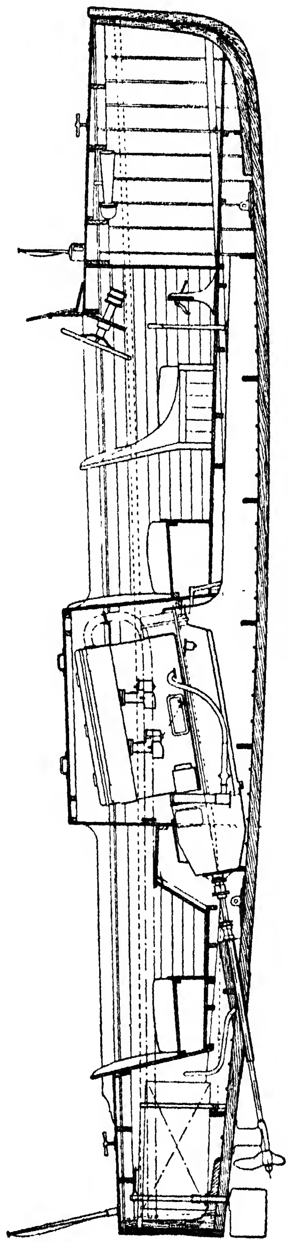


Fig. 35.—24-ft. Brooke Runabout; general lay-out

**The Gleniffer Paraffin Motor.**—Gleniffer Motors, Ltd., have for years specialized in the production of marine motors operated by paraffin, kerosene, or lamp-oil, as it is variously termed, though in all their engines petrol may be used if desired. The Gleniffer engines are well-designed robust four-stroke cycle single-acting governed engines with enclosed crank-chambers and H.T. magneto ignition. The centrifugal governor is mounted in the cam-shaft driving wheel, and is thus completely enclosed; it controls the engine throttle, the hand control merely regulating the speed at which the governor functions. This prevents the engine from racing when declutched, and from stopping when the clutch is engaged. The paraffin vaporizer is shown in part section in fig. 36. The engine *may* be started by a preliminary heating of the vaporizer by a blow-lamp for a few minutes, but is preferably started on petrol supplied from a small auxiliary tank through the two-way fuel supply cock to a normal float-feed carburettor, and is sprayed from a jet into the choke-tube as indicated; when the engine is well warmed up the fuel supply cock is turned over to the paraffin supply, and the engine then continues running on the heavier fuel. It will be noted that on issuing from the jet the paraffin spray passes up a "vaporizing tube" which is heated by an exhaust gas jacket; the air which is drawn by the piston suction past the spraying jet is also heated by this exhaust gas jacket as indicated.

To the rich mixture from the vaporizer tube on its way to the engine the correct amount of fresh cool air is added through a readily adjustable air valve of spring-controlled automatic type, damped by an oil dash-pot, and the cooled correct mixture then passes on to the inlet valve of the engine. The main air inlet is controlled by a throttle valve governor-operated as already described. When the engine is running light, or slowly, the main throttle is nearly closed, causing a partial vacuum in the vaporizer pipe which facilitates the vaporization of the paraffin

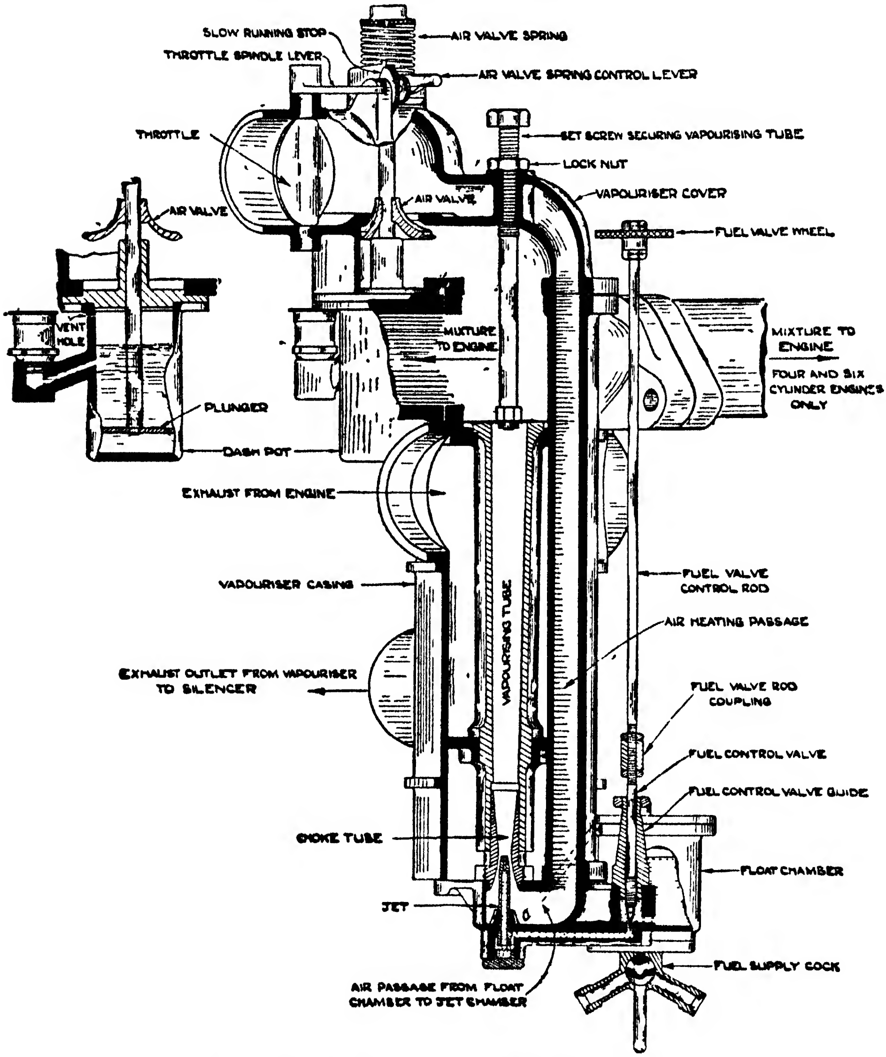



Fig. 36.—Gleniffer Vaporizer partly cut away to show Inside Passages and Dash Pot

spray, and so permits of long periods of slow running without risk of the vaporizer becoming over-cooled.

The cylinder jacket water is passed into a silencer wherein it mingles with, and cools, the exhaust gases, the mixture thence passing out through a copper exhaust pipe.

An external view of the K4 four-cylindered Gleniffer paraffin marine

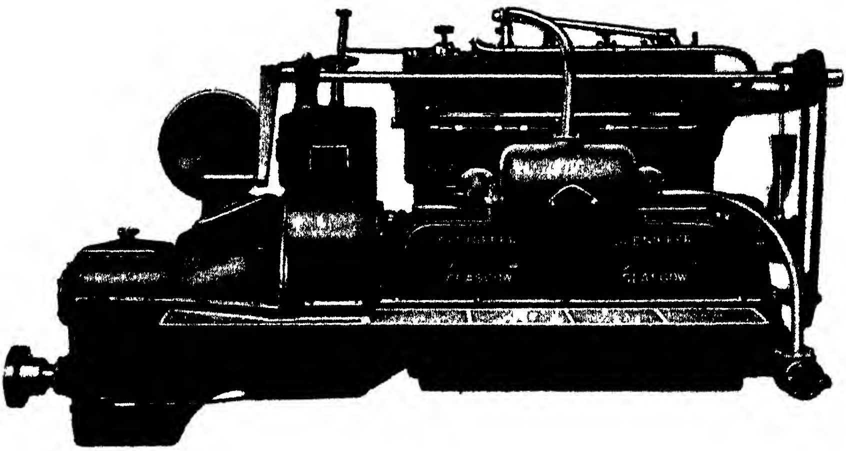


Fig. 37.—K 4 Gleniffer Engine, with Reverse Gear and Reducing Gear, Starboard Side

engine of 6-in. bore and 8-in. stroke is given in fig. 37. This 55–70 h.p. engine * has a normal full speed of 600–750 r.p.m., and a consumption at full output of about 0·8 lb. of paraffin per b.h.p. hour. Hand starting is by the hand-wheel and chain as indicated, but electric starting (and lighting of the vessel)—as in motor-car engines—can be fitted if required. The H.T. magneto is fitted with an impulse starter which obviates any need of violent "cranking" of the starting handle.

*The Reversing Gear.*—The reversing gear is completely encased and is shown, with its geared hand-operated wheel, at the left-hand end of fig. 37. The reversing gear runs in oil, and provides: (1) an ahead propeller drive with no moving gear-wheels; (2) a free engine; and (3) a reverse propeller drive at ahead revolution speed. The principle of its action will be understood from the diagram, fig. 38. The after end of the crank-shaft A terminates in a bevel wheel; an equal bevel wheel is carried by the forward end of the driven shaft B which, in the absence of a reducing gear, is the propeller shaft. These two bevels mesh with the "planet" bevels CC whose spindles are fixed in a casing DD, which, when not otherwise constrained, is capable of turning freely on the axis AB. The casing DD also carries

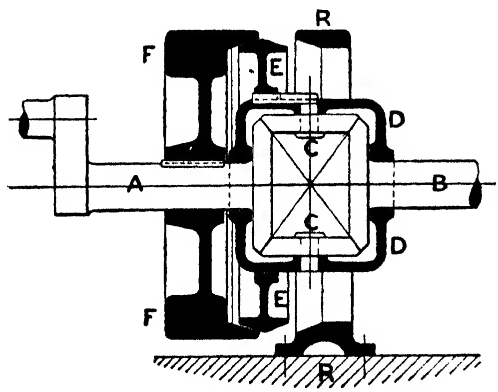



Fig. 38

* 55 h.p. on paraffin, 70 h.p. on petrol.

the double-coned member EE which rotates with it, but is capable of sliding upon the key shown, so that it may engage either with the coned surface of the engine flywheel FF, or with that of a ring RR *fixed* in the boat, or may occupy an intermediate position in which it is not in engagement with either FF or RR.

By means of a hand-operated gear EE may be brought into pressure engagement with FF; EE, DD, and CC all then revolve with the crank-shaft as one piece; the planet wheels are unable to revolve on their spindles, and accordingly the driven shaft B is compelled to revolve at the same speed, and in the same direction, as that of the crank-shaft; this is the direct drive forward.

Next suppose EE to be free of either FF or RR, and that the engine is

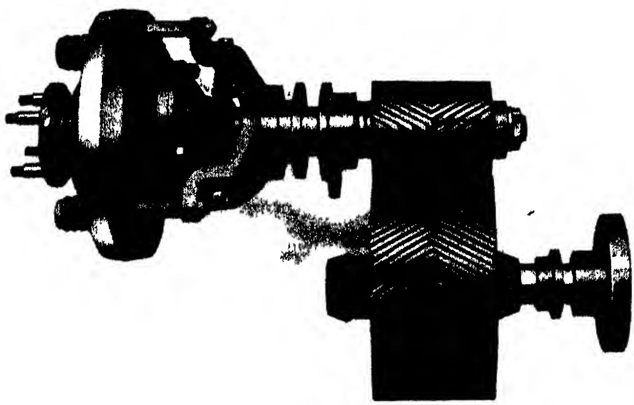


Fig. 39 —Revolving Parts of Reverse Gear and Reducing Gear Wheels

running. The shaft B is now at rest; the planet wheels CC run round B's bevel as a rack, being driven by A's bevel; and their spindles, together with the casing DD, and the coned member EE, revolve idly in the same direction as the crank-shaft A, but at half its speed; this is the free engine position.

Lastly, let EE be brought into pressure engagement with RR; then EE and DD are fixed; the planet wheels CC, driven by A's bevel, revolve on their fixed spindles, and drive B's bevel in the opposite direction to that of A, but at the same speed; this is the reverse position.

The key on which EE slides on DD is actually not axial, but inclined, i.e. is a short portion of a helix. The effect of this is that both when in ahead and astern gear the inclination of the key causes an axial pressure forcing the coned surfaces into closer contact. Thus no clutch spring is necessary, obtuse-angled cones may be used, and all wear is automatically taken up. Actually, also, the clutch member EE is in two parts, the outer coned part being connected with the disc and boss through helical springs, to prevent too fierce a clutch engagement (see fig. 39).

The reversing gear is operated by a hand-wheel and gear pinion meshing with a geared sector, as shown on the left in fig. 37. About one revolution of the hand-wheel suffices to pass from the ahead, through neutral, to the astern position; and when desired the gear can be operated from the deck or elsewhere in the boat.

*Reducing Gear.*—In many boats, and particularly in slowly moving vessels as auxiliaries, tugs, barges, &c., a gain in efficiency of both engine and propeller may be obtained by arranging for the propeller to run more slowly than the engine; in rough weather also a reducing gear is advantageous in enabling the boat to drive into a head sea with less loss of speed than when the drive is direct.

The reducing gear used by the Gleniffer Company is shown in fig. 39; it consists of a pair of very wide and strong double-helical gears giving a silent transmission of about 98 per cent efficiency, working in an oil bath, and completely encased in an extension of the crank-case as shown in fig. 37. The pinion is carried on an axially floating sleeve by the shaft B (fig. 38), to permit of true meshing, while the wheel is mounted on a short rigid shaft below borne on coned roller bearings which also take the propeller thrust. The slow-speed shaft being 8–9 in. below the crank-shaft allows of a large-diameter well-submerged propeller. The reduction in this case is nearly two to one, the actual ratio of propeller to engine speed being 18 to 35.* Gleniffer marine motors are built in sizes ranging from 7–8 h.p., 3 in. × 4½ in. two-cylindered, to 80–105 h.p., 6-in. bore × 8-in. stroke six-cylindered engines; the lower powers quoted being the normal full output when paraffin is used as fuel.

**Thornycroft Paraffin Marine Motors.**—Messrs. Thornycroft have for many years specialized, *inter alia*, in the production of paraffin marine motors and build a wide range of designs. An early installation, about 1912, was a set of eight-cylindered 12-in. bore × 8-in. stroke × 550 r.p.m. four-stroke enclosed engines for submarines for the Italian Government.

**The RD4 Thornycroft Engine.**—A typical design is that of the RD4 engine, a sectional elevation of which appears in fig. 40. This is a governed four-cylindered monobloc side-valve four-stroke enclosed motor of 4·375-in. bore and 5·5-in. stroke, running normally at up to 1100 r.p.m., with an output at this speed of 35 b.h.p., using paraffin of ·78 to ·825 specific gravity and flash-point 70° F. to 160° F. (close test) as fuel. The consumption at full power is about 0·8 lb. of paraffin per b.h.p. hour.

The enclosed centrifugal spring-loaded governor, running at about half crank-shaft speed, will be noted on the right of fig. 40. This governor acts directly on the throttle control valve. On the left side is shown the reversing gear, completely enclosed, with a large inspection door in the top of the casing. The reversing gear is of the epicyclic type, with expanding ahead clutch and contracting reverse clutch; a free engine position is provided between ahead and reverse.

* As 18 and 35 are prime to one another, each tooth of each wheel meshes in working with each of the other.

The cylinder head is detachable, and the cylinders themselves are readily removable. The three-bearing crank-shaft is carried in the lower half of the crank-case, thus facilitating overhaul of the engine in the boat if necessary.

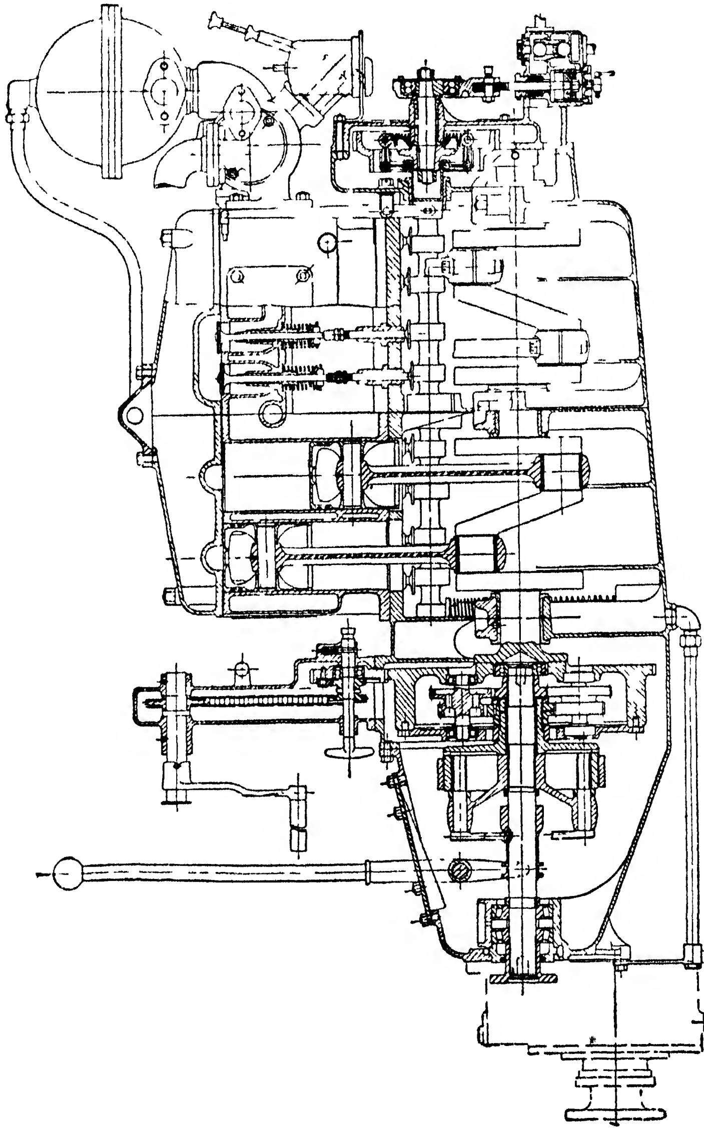


Fig. 40.—Sectional Arrangement of "RD4" Engine and Reverse Gear

The engine is hand started on petrol, the H.T. magneto being fitted with an impulse starter; when well warmed up the paraffin fuel supply is turned on. Starting *can* be effected on paraffin by heating the vaporizer by blow-lamp, but this is rarely resorted to. Electric starting (and lighting) is fitted when required.

*The Paraffin Carburettor and Vaporizer.*—The paraffin carburettor comprises a float-feed and an exhaust-heated vaporizer; an external view is given in fig. 41. The liquid fuel is supplied to the float chamber A

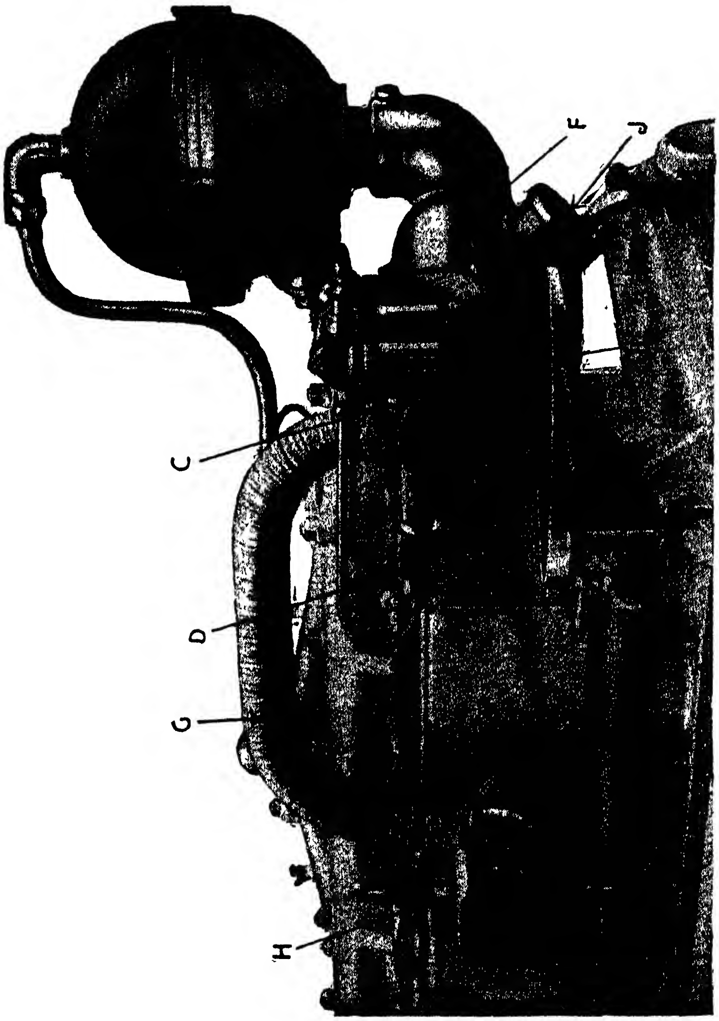



Fig. 41.—The Thornycroft Paraffin Carburettor and Vaporizer

through the two-way cock D, the engine being started on petrol and turned over to paraffin when warmed up. The piston suction induces a current of air through the gauze strainer E; this air next traverses a horizontal choke tube B in which is situated a vertical spraying jet controlled by a needle valve C, from which it takes up a charge of fuel spray; the rich mixture thus formed passes through the exhaust-heated vaporizer F, which is fitted with a simple deflector by which the supply of exhaust gas to the vaporizer may be varied to the extent of shutting it off completely, and passing it direct from the exhaust manifold to the silencer.

From the vaporizer F the rich mixture is drawn through the pipe G and passes the adjustable spring-controlled automatic air valve H which admits fresh air, cooling it, and forming a correct mixture for combustion. This corrected mixture finally passes to the engine through a throttle valve controlled jointly by a hand lever and by the governor, by which the amount of mixture, and therefore the engine speed, is determined. It may here be remarked that it is well to cause the engine both to start and *stop* on petrol; and when stopping, to turn off the petrol and allow the engine to run on until the float chamber is emptied.

The jacket-water circulation is maintained by a double-acting plunger pump with ball valves, driven by an eccentric, through gearing, from the crank-shaft at about half engine speed; this pump has a capacity of about 300 gall. per hour at normal full speed. The hot water from the jackets passes into the silencer, and is discharged overside, mixed with the cooled exhaust gases.

*Lubrication.*—Lubrication is forced, an eccentric-driven spring-loaded plunger pump drawing oil from the engine sump through a brass strainer and delivering it to the main bearings, and to the big ends through ducts drilled in the crank webs; the exude from these bearings lubricates the pistons, gudgeon-pins, reversing-gear, &c. A plunger-type oil indicator fitted near the throttle control shows, by the protrusion of the plunger, that the oil pump is correctly working. The sump requires 2–3 gall. of oil, dependent upon the angle at which the engine is set in the boat. The oil consumption at full power is about 1 pt. in 3 hr.

**A 140-h.p. Petrol Marine Engine.**—For use in fast hydroplanes and vessels of the "express cruiser" type, Messrs. Thornycroft build a six-cylindered governed engine of 4·75-in. bore and 6·5-in. stroke giving 140 b.h.p. at 1800 r.p.m., with a consumption not exceeding 0·65 lb. of petrol per b.h.p. hour at full power. An external view of the starboard side of this engine is given in fig. 42; this shows the enclosed reversing gear, H.T. magneto with impulse starter, 12-volt dynamo, and starting motor.

The engine is of the normal enclosed four-stroke type with two blocks each of three cylinders, the inlet valves being located in the detachable heads and placed immediately above the exhaust valves which are seated in the cylinder blocks. Large doors are provided for cleaning the cylinder jackets. The jacket-water sea inlet is fitted with a stop-cock, and also with a strainer which can be withdrawn and cleaned without stopping the engine. The exhaust manifold is also water-jacketed, and a silencer with water injection is fitted at the forward end; cocks and plugs are provided for draining all jackets.

The pistons are of aluminium alloy, and the connecting-rods and crank-shaft of nickel chrome steel; the crank-case is of aluminium.

There are two carburettors of Zenith type, simultaneously controlled by one throttle lever; petrol feed is by gravity or, when this is impracticable, by pressure supplied by a small mechanically-operated air-pump. A bilge pump is fitted at the forward end of the engine, directly driven through a disengaging clutch.

A multi-disc clutch of bronze and steel plates alternately and a reversing gear of normal epicyclic type are at the after end, and totally enclosed in an oil-tight casing; the propeller thrust is taken by a double roller-coned bearing at the after end of this casing; the control lever can be fitted on either side of the casing, thus facilitating operation in the case of twin-screw installations.

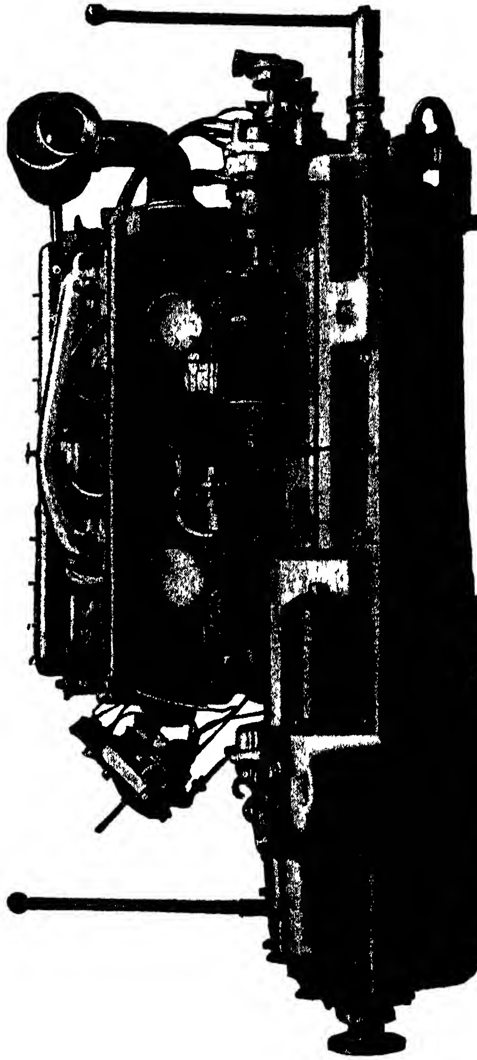


Fig. 42.—6-cyl 140-h.p. Thornycroft Petrol Marine Engine

The tail shaft is of naval brass, 1·75 in. in diameter, and the stern-tube is of solid-drawn brass with gun-metal ends having at its inner end a gland and stuffing-box with large greaser; the tube is fitted with renewable white-metal bearings, 7 in. in length, of a suitable alloy for under-water service; the standard length of stern-tube is 4 ft. The three-bladed propeller ordinarily used with this engine varies from 17·5 in. in diameter in the case of high-speed craft, to 22 in. diameter in " cruisers "; for twin-screw installations engines can be supplied to rotate in opposite directions.

The net weight of the engine, reverse gear, stern gear, and including standard tanks and piping, is 21 cwt.

A line view of a typical single-screw installation appears in fig. 43, while fig. 44 shows an external view of the 55-ft. twin-screw cruiser *Cuba*, 280 h.p. and 14·75 knots.

**The Petter Super-Scavenge Oil-engine.**—A compression-ignition

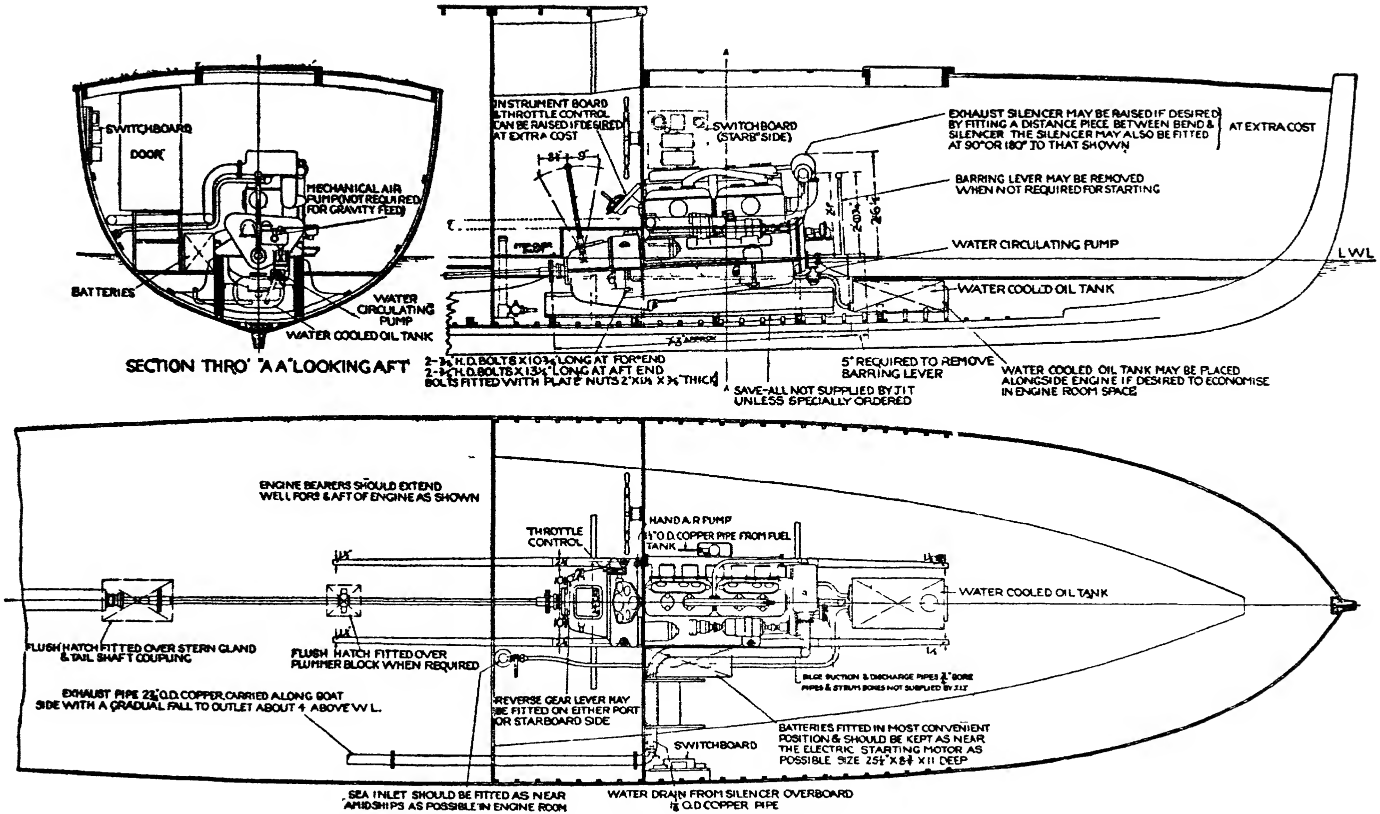



Fig. 43.—R.I D. 100. Typical Installation Arrangement of Engine and Reverse Gear

engine of unique design with an excellent performance has been developed by Petters, Ltd., of Yeovil. It operates on the two-stroke cycle, and its characteristic features are exceptional compactness, very favourable power/weight ratio, capacity for high overload for long periods and a low piston speed of 1082 ft. per min. It embodies a form of super-scavenge of which the details are a very efficient blower acting in unison with twin overhead exhaust valves in the cylinders, arranged to effect complete scavenging with clear exhaust.

Figs. 45 and 46 are sectional views of the engine, which is made in two to six cylinder units, developing a b.h.p. of 62½ per cylinder at 500 r.p.m.

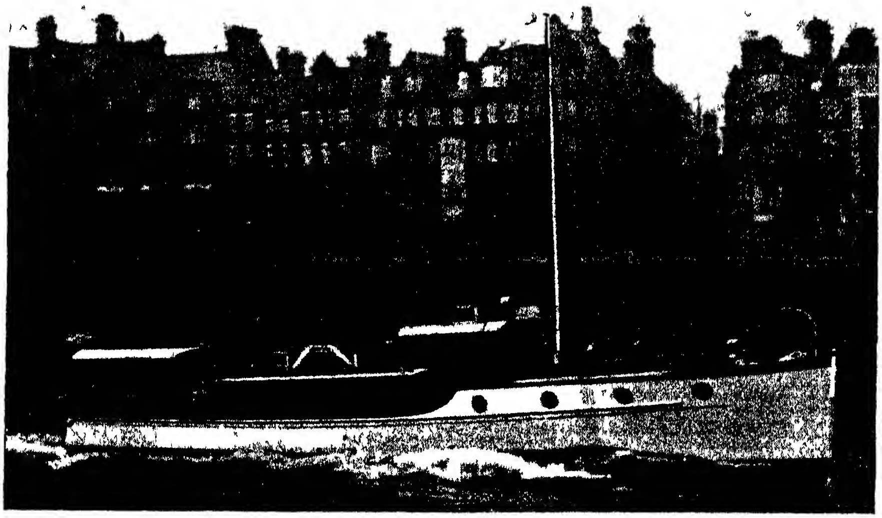


FIG 44.—*Cuba* 55-ft Thornycroft Twin-screw Cruiser Speed 14¾ knots

The most important development in this engine is the introduction of uni-directional scavenging. The arrangement of the air ports controlled by the piston secures the necessary controlled swirl and turbulence, while the exhaust is taken through valves located in the cylinder head. At the same time, the blower pressure has been reduced from 2½ to 1½ lb./sq. in. owing to the improved scavenging, thus reducing the size of the blower and the power required to drive it.

One result of this arrangement is that it is possible to rate the engine higher than most other two-stroke engines, while the exhaust even on overload is invisible, a feature which will commend itself to all users of oil engines.

To obtain the improved performance, together with a satisfactory fuel consumption and brake m.e.p., the following factors were taken into consideration, all of which were included in the new design:

(1) The provision of a pre-advance opening of the exhaust valves in

order to discharge the bulk of the exhaust gases before the air ports are opened. The exhaust valves are closed shortly before the air ports, so that the cylinder is completely filled with cool air only.

(2) The admission of air by more or less tangentially disposed ports at the opposite end of the cylinder, to secure complete scavenging and provide necessary swirl. Excess air is delivered in this way and assists in keeping the piston and exhaust valves cool.

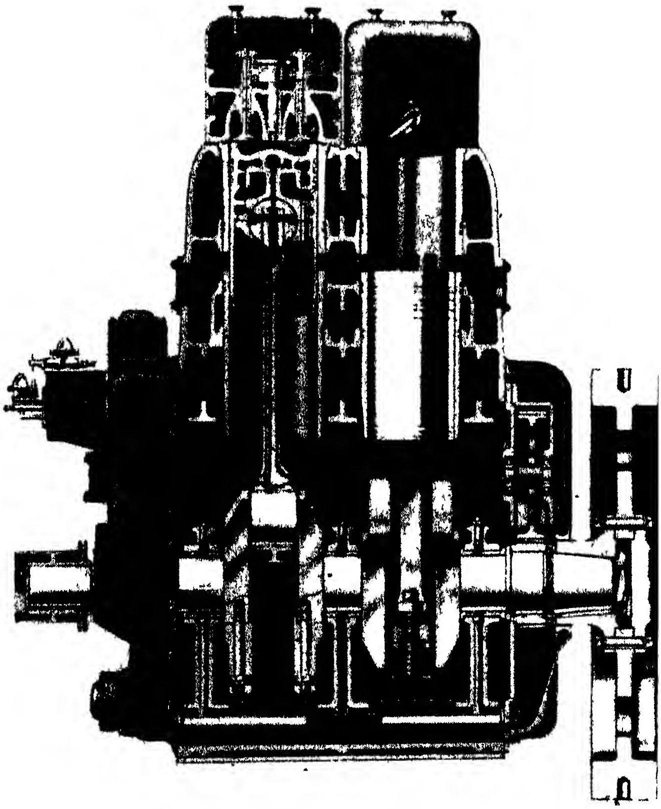


Fig. 45.—Petter Super-scavenge Oil-Engine. Sectional illustration

(3) Admission of the atomized fuel in such a manner that it is completely burnt, advantage being taken of the complete scavenging and swirl obtained as above.

(4) By disposing the exhaust valves in the cylinder head and the air ports at the bottom of the piston travel, freedom from piston and cylinder distortion is ensured. At the same time the temperature conditions throughout the engine are more equable. This results in a more reliable engine, which is capable of withstanding high overloads continually without distress.

Other features which are embodied in the engine are:

The underside of the cylinder head is flat, the combustion chamber being formed by a depression in the piston crown. A trouble-free and accessible cylinder head is thus obtained. Special deflector pipes are fitted to direct cooling water on to the spaces around the fuel injector and exhaust valve ports.

C.A.V. Bosch injector equipment is standard.

The lubricating oil pumps are driven by a bush-roller chain from the forward end of the crankshaft.

Forced lubrication is employed with a dry sump. The oil collected by the scavenge pump is delivered to an external storage and settling tank through an oil cooler. A pressure oil pump delivers oil to all bearings and to the spherical small-end bearings. After circulation through the piston for cooling purposes the hot oil returns to the sump. A streamline filter is incorporated and by means of a tapping taken off the lubrication system a proportion of the oil is continually reconditioned. Clean lubricant is delivered to the cylinder liners by a mechanical lubricator.

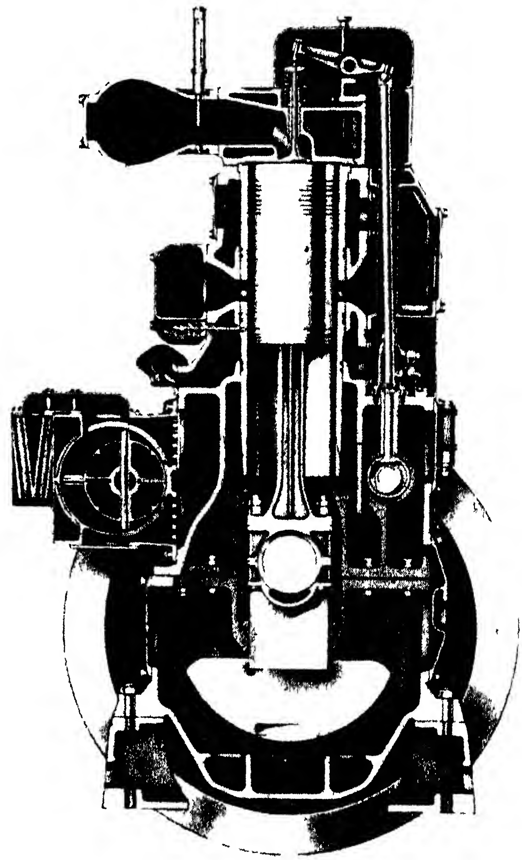


Fig. 46.—Petter Super-scavenge Oil-Engine. Sectional illustration

The guaranteed fuel consumption is less than 0·39 lb. per b.h.p. hour, the fuel consumption curve on the standard 500 r.p.m. showing a flat loop from no load to a brake m.e.p. of 90 lb. sq. in. Between brake m.e.p.s of 45 and 86 lb. sq. in. the fuel consumption is below 0·4 lb. per b.h.p. hour, the lowest consumption of 0·39 lb. being at 67 lb. sq. in. The exhaust temperatures are very moderate under these conditions, being below 750° F. at full load.

A standard engine with no special tuning was tested by Prof. C. J. Hawkes in June, 1938. A fuel consumption of 0·381 lb. per b.h.p. hour was recorded on a four hours' duration test at full load. This was followed by one hour at 10 per cent and two hours at 20 per cent overload, the fuel consumptions being 0·383 and 0·386 lb. respectively. After a test at three-quarter, half, quarter and no load, the engine was run at 34 per cent overload for over 10 minutes, the exhaust being only slightly hazy. During all of the other tests the exhaust was quite colourless. This was confirmed by the author's own observations of similar tests on an engine of the same type at Messrs. Petters' works.

The engine is now in production in 2, 3, 4, 5 and 6 cylinder units of 62½ b.h.p. per cylinder. The reduction in the power weight ratio commends it as a propulsion unit to owners and builders of small craft of all types, the light weight factor being of special importance where vessels of shallow

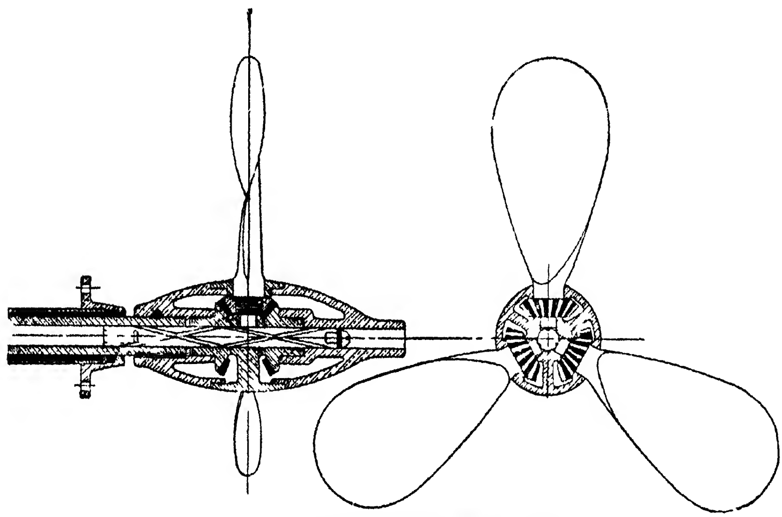


Fig 47—The Duerr Reversible-bladed Propeller

draught are concerned. Manœuvring is effected by compressed air, all starting and speeding being controlled by a single handwheel, reversing by a separate interlock lever. The air compressor and interconnected bilge and circulating water pumps are built into the forward end of the engine and are gear driven at low speed.

*Reversing.*—Where reverse gear is fitted it is of the completely enclosed double-cone clutch and bevel type substantially as already described (see fig. 38); this is usually fitted to the 5- and 8-h.p. engines. Larger engines when so fitted have reversing gear including a multi-disc clutch of alternate bronze and steel discs, the reverse gear being of epicyclic type, with brake band contracting on a drum. Alternatively, a Duerr reversible-bladed propeller may be used, as illustrated in fig. 47; in this the blades are operated by bevel gear wheels in the propeller boss. Crown wheels on an internal shaft are operated by means of three keys fitted in helical keyways which are cut, right- and left-handed respectively, in the shaft. Thus an axial movement of the internal shaft rotates the pair of crown wheels in opposite directions, and these then turn the blades through their attached bevels.

Each blade is well supported in two bearings, and the whole device is water-tight and grease-retaining. Obviously all blade positions from full ahead, through neutral, to full astern, are obtainable, and this is of great value in line fishing when, for long periods, an exceedingly slow movement of the vessel must be maintained.

**General.**—Among other established builders of the smaller types of marine internal-combustion engines typified by the cases above described in the present chapter are: Ailsa Craig (Chiswick), Amanco (London), Atlantic Works (Scotland), Chrysler (London), Dorman (Stafford), Widdop (Keighley), Woodward (Hull), and Morris Engines (Coventry).

With marine internal-combustion engines, especially of the smaller classes, it is to be borne in mind that a breakdown, which in a motor-car might be a trifling matter, may easily become a grave occurrence at sea or in a crowded anchorage in a tideway. Thus the marine engine must be of robust design, with all parts well protected against the action of the sea and moist air, to ensure maximum reliability. Moreover, careful consideration must be given to the problem of rendering parts easy of accessibility for examination and adjustment. Large crank-case doors should, for example, be provided where they can not only be easily removed, but also when removed enable the big ends to be comfortably adjusted or renewed. The question of twin-screw versus single-screw installations has been fully discussed by B. Joy,* who points out that the great advantage of the twin-screw is the increased safety, as in the event of the breakdown of one engine the other is capable of carrying on and keeping the boat under way during the repair. Again, with two screws a boat can be turned in its own length, while if the steering gear should fail the boat can still be steered by manœuvring the two engines. With twin screws each engine is smaller and lighter, and the fore-and-aft length of the engine-room is consequently reduced; further, in boats where accommodation is provided aft of the engines, a twin-screw outfit gives additional head room aft, since the propeller shafts can be run under bunks or seats, while the floor can be lowered so far as to leave only just enough room for bilges or ballasting. With two engines also there is a clear passage between them at maximum head-room, whereas with a single engine the passage-way must be at the side, which is often very irksome. Again, owing to their smaller cylinders, twin-screw engines are, in general, more easily started up—often an important matter; while when repairs become necessary the smaller and less heavy parts of a twin-screw set are an advantage in handling and lifting out.

In shallow-draught boats twin screws enable the propellers to be reduced in diameter on account of the smaller power of each; and again, in a heavy sea twin propellers have better immersion not only on account of their smaller diameter but also because they are situated nearer the

* See *Proc. Inst. Auto. Engrs.*, 1926, Vol. XX.

midship section of the boat, and are thus less liable to race; and even if racing should occur it is less objectionable with a small than with a large propeller.

The ideal arrangement is a twin-screw installation with the engines completely " handed ", i.e. with each engine a mirror image of the other. Thus both will run inwards or both outwards, and all valves, carburettors, magnetos, &c., will in each case be both on the outboard or both on the inboard side.

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# CHAPTER VI

## Types of Oil-engines

Oil-engines can be classified according to the cycle upon which they work, whether the two- or the four-stroke, and as to whether they are used for land duties or for marine work, whether vertical or horizontal, of the trunk piston or of the crosshead type, whether single- or double-acting.

To deal first with the differentiation of vertical and horizontal oil-engines. The first Diesel engine built was a vertical engine, and for some ten years thereafter development somewhat slavishly copied this earlier pattern, so that it was not until 1910 or thereabouts that the question of building horizontal Diesel engines was seriously considered and their construction undertaken.

**Horizontal Engines.**—Gas-engines, as is well known, had been developed primarily as horizontal machines, and there is no fundamental reason underlying the preference for this system with gas-engines as compared with Diesel engines. The advantages of the horizontal system of construction are: (1) less head-room is required, (2) the pistons are more easily withdrawn for examination, and (3) the construction may be somewhat less costly. As against these factors it must be stated that larger foundations are called for to support the horizontal cylinders, and the wear on the liners in most cases will be less regular, due to the operation on the liner of the effect of gravity as well as that of the connecting-rod side pressure.

Horizontal Diesel engines of the double-acting four-cycle type, similar to the large gas-engines constructed by a number of firms, have been built up to relatively large powers, the largest set giving in 4 cylinders (arranged in two sets of 2 tandem cylinders, each double-acting) operating one generator, some 2000 to 3000 b.h.p. total. Many small horizontal Diesel engines with airless injection have been built in recent years for various land duties, almost all, with a very few exceptions, working on the single-acting, four-stroke cycle principle, and having trunk pistons. All the double-acting engines of course had crossheads.

For land work the four-stroke cycle principle predominates at the present time, but the two-stroke cycle engine has made some headway, and will no doubt be more generally used in the future.

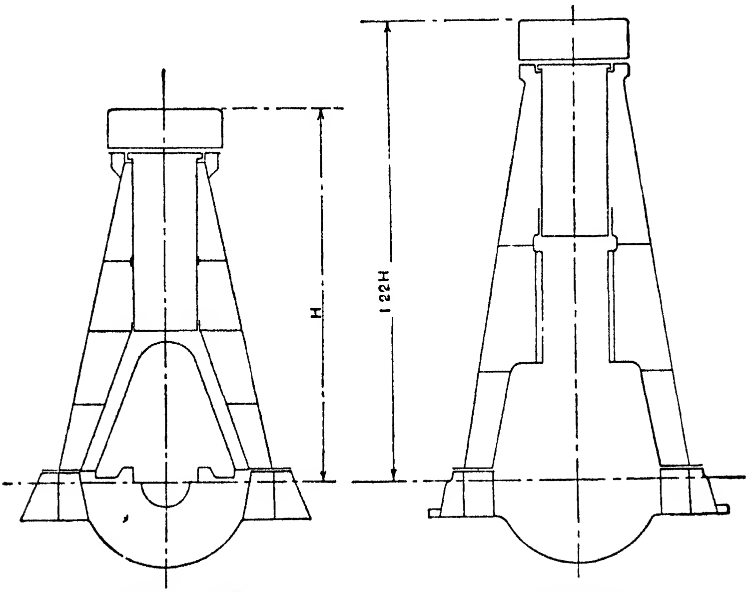
**Trunk Pistons.**—The majority of land engines are of the trunk-piston type. This system has given wonderfully satisfactory results, although the double duties demanded of the piston—acting as a gas-tight plug to confine the gases to their proper sphere of operation, and as a crosshead to take any side thrust due to the obliquity of the connecting-rod—are rather more severe than is desirable for continuous operation without undue wear and the necessity of overhaul. The tendency, however, with land engines at the present time is slightly marked towards providing a crosshead to take the side thrust due to the obliquity of the connecting-rod for engines of 100 or more b.h.p. per cylinder, although trunk-piston engines have been built and have operated reasonably satisfactorily up to the relatively high figure of some 400 b.h.p. per cylinder.

In the marine field during the past few years the trunk-piston type of engine has been fitted in cross-channel and moderate-sized passenger vessels, where it is giving satisfaction The engines vary in output from 500 to 6000 i.h.p. developed in six to twelve cylinders running at from 150 to 350 revolutions per minute. See Tables II to IV (p. 164).

This type of engine working in conjunction with gearing is also in use for ship propulsion. This arrangement allows two engines of comparatively high speed, say from 250 to 400 revolutions per minute, to be coupled through reduction gearing to a single propeller shaft, and the benefits of having comparatively light-weight engine parts to handle when overhauling assist in reducing upkeep expenses; further, one or both engines can be used as required. In some cases hydraulic couplings are interposed between the engines and gears, insuring a very flexible engine drive.

**Crosshead Marine Engines.**—The diagram (fig. 48) indicates the extra height required by the interposition of a crosshead between the crank and the piston to take the side thrust of the connecting-rod. For marine work, except in special cases such as high-speed machinery for cross-channel vessels, the crosshead is becoming a general provision, and is to be strongly preferred, as simplifying the duties called for from the piston, facilitating overhaul, and achieving much better conditions, lower temperatures and pressures for the gudgeon-pin or top-end bearings. At the present time it is in the marine field, however, that the Diesel engine is finding its fullest application, and higher powers are now being called for by shipowners, who have gradually come to appreciate the economies possible and to realize that reliability with this type of prime mover can undoubtedly be achieved. The advantages claimed are gradually assuming, as time goes on, greater and greater proportions in view of the present cost of labour and the gradual lessening of the percentage difference between the price of coal and oil fuel at the principal ports of the world. In Chapter VIII is indicated the nature of the savings which are effected. As already stated, the majority of marine Diesel engines are of the slow-speed crosshead type—slow speed, in order to obtain a reasonable efficiency of propeller with a relatively slow ship speed.

There is, however, as mentioned in the section under trunk pistons, a tendency to adopt gearing between the engine and propeller, allowing comparatively high-speed engines with their lighter working parts to be used in conjunction with the most efficient propeller. The crosshead type of engine can also be used in such an arrangement, so obtaining similar benefits.




Trunk Piston Type Crosshead Type

Fig. 48.—Diagram showing Comparison of Heights of Engines

The seven diagrams (figs. 49–55) indicate generally the various types of oil-engines now being applied for marine propulsion. Whilst all but one of these refer to the two-stroke cycle principle, by far the largest number of motor-ships and the largest horse-power in commission at sea depend upon the four-stroke single-acting engine for their propelling machinery.

These engines can use all grades of fuel oil suitable for heavy-oil engines, and will give a fuel consumption at full power rating of about 0·38 lb. per b.h.p. per hour.

They are being extensively fitted for electric generator driving on board ship, and will no doubt ultimately find favour for propulsion purposes when used in conjunction with gearing between the engine and the propeller.

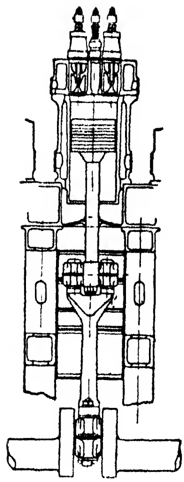


Fig 49

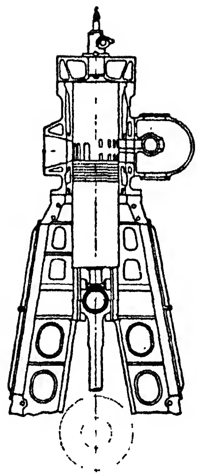


Fig. 50

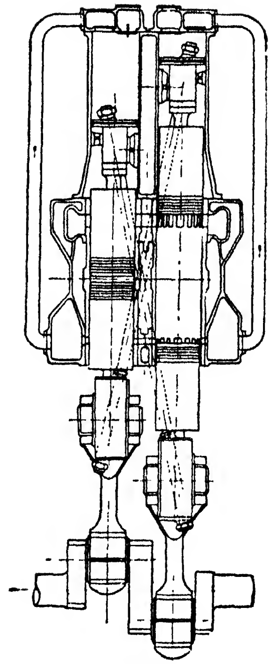


Fig. 52

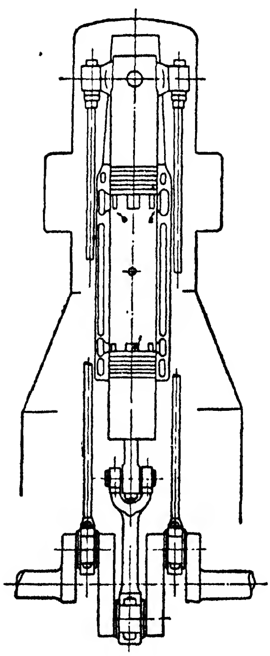


Fig. 51

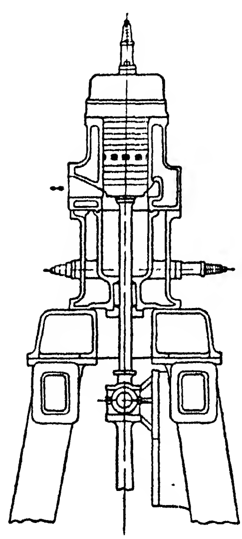


Fig. 53

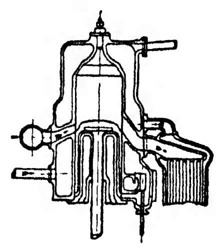


Fig. 54

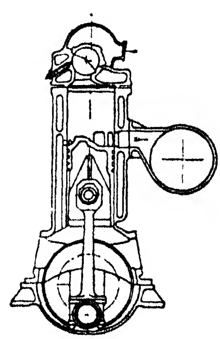


Fig. 55

**Diagram illustrating various types of Oil-engines**

**Fig. 49.—4-Stroke Cycle, Burmeister & Wain Type. Fig. 50.—2-Stroke Cycle, Sulzer Type. Fig. 51.—2-Stroke Cycle, Doxford Opposed-piston Type. Fig. 52.—2-Stroke Cycle, Fullagar Opposed-piston Type. Fig. 53.—2-Stroke Cycle, M.A.N. Double-acting Type. Fig. 54.—Combined Steam and Oil, Still System. Fig. 55.—2-Stroke Cycle, Semi-Diesel Type.**

# CHAPTER VII

## The Units of the Diesel Engine

The diagram (fig. 56) shows the various units that compose in principle the complete standard Diesel engine plant. This engine works on the four-stroke cycle principle, and the various valves in the cylinder head will clearly be seen, the fuel, starting-air, inlet, and exhaust.

**Fuel Pump.**—From the fuel tank fuel is led down the suction pipe to the suction valve of the fuel pump, which may be driven off any convenient part of the engine mechanism, generally either from the camshaft operating the valves or from the vertical shaft driving the camshaft. This pump draws in the fuel through the suction valve, which is of the automatic spring-loaded " Poppet " type, and discharges through the automatic spring-loaded discharge valve up to the fuel valve in the cylinder head. The fuel valve in the cylinder head is also in communication with the fuel-injection air-bottle, which is kept charged from the compressor driven generally from the main engine. The pressure at which the fuel-injection air-bottle is maintained must, of course, be higher than the compression pressure in the engine cylinder, otherwise no fuel and injection air would enter the engine cylinder until the pressure had dropped to lower than that of the bottle. The actual margin of excess pressure required depends upon the load at which the engine is running and upon the type of fuel oil which is being used. The heavier and more viscous the fuel oil, or the higher the horse-power required, the greater the difference in pressure required between that in the bottle and that in the engine cylinder. This in no case, however, exceeds 600 lb. per square inch, and with the usual type of Diesel fuel oil 300 to 400 lb. is sufficient; that is, with a compression pressure of 450 lb. per square inch in the main cylinder the injection-air pressure is maintained at 850 to 900 lb. per square inch.

The fuel pump, therefore, it will be seen, must deliver the fuel oil against the pressure of air in the fuel-valve in the cylinder-head fuel-valve casing and in the bottle.

**Regulation of Engine.**—Regulation of the power and speed of the engine is carried out by adjusting the fuel pump so that the less the power required from the engine, the smaller the quantity of fuel oil which is pumped to the cylinder-head fuel valve and is injected into the cylinder, and conversely. This is effected on most Diesel engines by tripping open the suction valve of the pump, and through it by-passing part of the delivery charge from the pump back to the fuel tank. With engines which are controlled by a governor suitable mechanism intervenes between this suction valve and the governor, so that the quantity of fuel is regulated by the governor and its mechanism according to the load to give the desired constant speed of the engine. The fuel, therefore, is delivered up to the fuel valve, and on this valve opening at the correct time and for the requisite period, the injection air blows the fuel into the cylinder, atomizing it, where it burns, heating the air within the cylinder, which, on expanding, drives down the piston and does the work of the cycle.

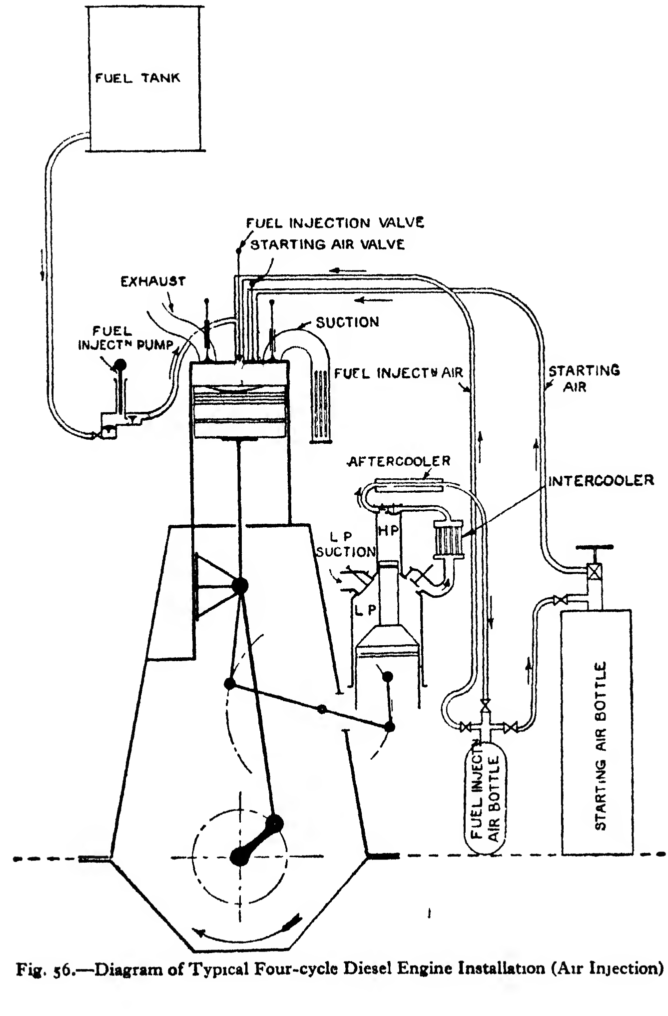



Fig. 56.—Diagram of Typical Four-cycle Diesel Engine Installation (Air Injection)

**Airless Injection.**—A method of fuel injection, which has almost entirely displaced air-injection within recent years, is known as the solid- or airless-injection method. In this case no compressor or fuel-injection air-bottle is required; the fuel is delivered by the pump into the fuel-valve casing at a very high pressure, is injected into the cylinder through minute holes, and by its own pressure is sufficiently atomized to a finely divided state to ensure ignition and combustion. The fuel pressures with airless injection vary, according to the details of the system, from 4000 to

10,000 lb. per square inch at full power. The higher pressure is used with the larger engines in order to secure sufficient penetration.

As already indicated in the sections dealing with the four-stroke cycle engine, two-stroke cycle engine and high-speed Diesel engine, airless injection is now common practice, thus simplifying the engine by the elimination of the blast-air compressor with its attendant fittings.

**Atomizing of the Fuel.**—A large number of atomizing devices have at various times been suggested both for the solid- and the air-injection engines, but it is doubtful whether these play any vitally important part. The one fundamental is, that to avoid carbonization at the point of the fuel valve,

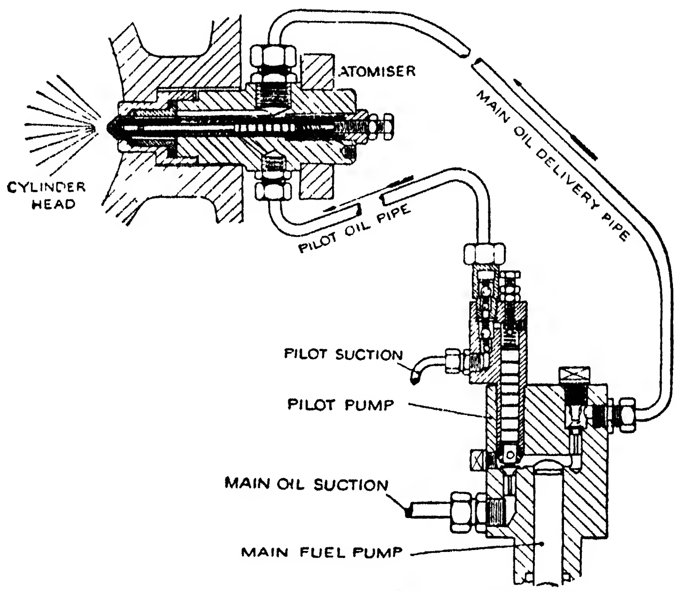



Fig. 57.—Main Pilot and Fuel Pumps and Atomizer for Tar Oil (Ruston & Hornby High-compression Oil-engine)

the flame plate must be kept cool, and the tendency is evidenced at the present time to eliminate a large number of the devices, so to make the interior of the fuel valve very much more simple, and to rely on the drop in pressure between the contents of the fuel-valve casing and the cylinder, as well as the form of the flame plate or nozzle, to give the required degree of atomizing and distribution throughout the combustion chamber. The cycle, in respect of the inlet and exhaust valve, has been fully described.

Fig. 57 illustrates one form of fuel injection arrangement for a horizontal compression-ignition engine. The fuel pump is the injector and no air-compressor is required. The pump not only measures out the quantity of fuel required but injects it through the nozzle into the spherical combustion chamber at the requisite high pressure to secure atomization and at the correct point in the cycle. The needle closing the orifice in the spraying nozzle is lifted off its seat by the oil pressure acting against the

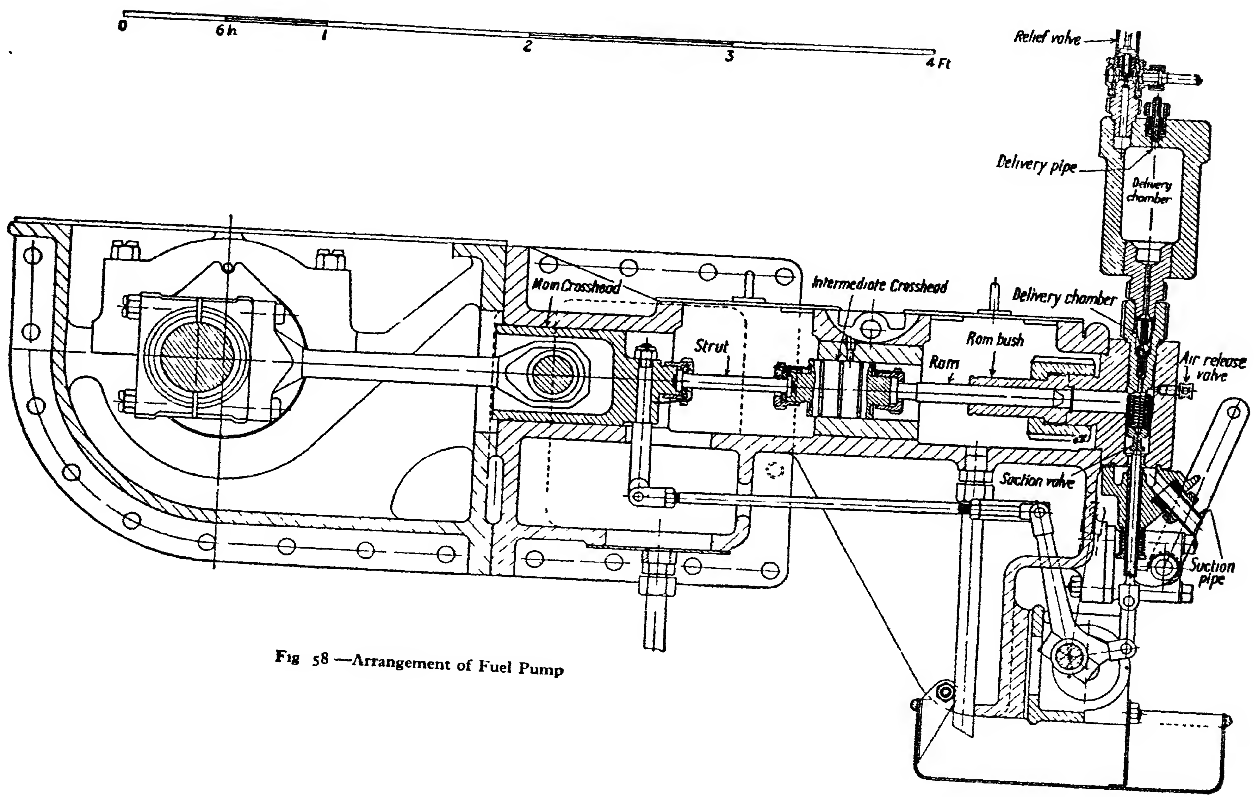



Fig 58 —Arrangement of Fuel Pump

pressure of the spring shown in the diagram. Governing is by a " spill valve ", which allows more or less of the oil pumped to return to the suction side of the pump, depending upon the load.

Fig. 58 shows the arrangement of fuel pump for the improved Doxford engine. Each cylinder is provided with an individual fuel pump, the complete battery being mounted over the main thrust block. The motion of the connecting rod shown in fig. 58 is derived from the crank-shaft through

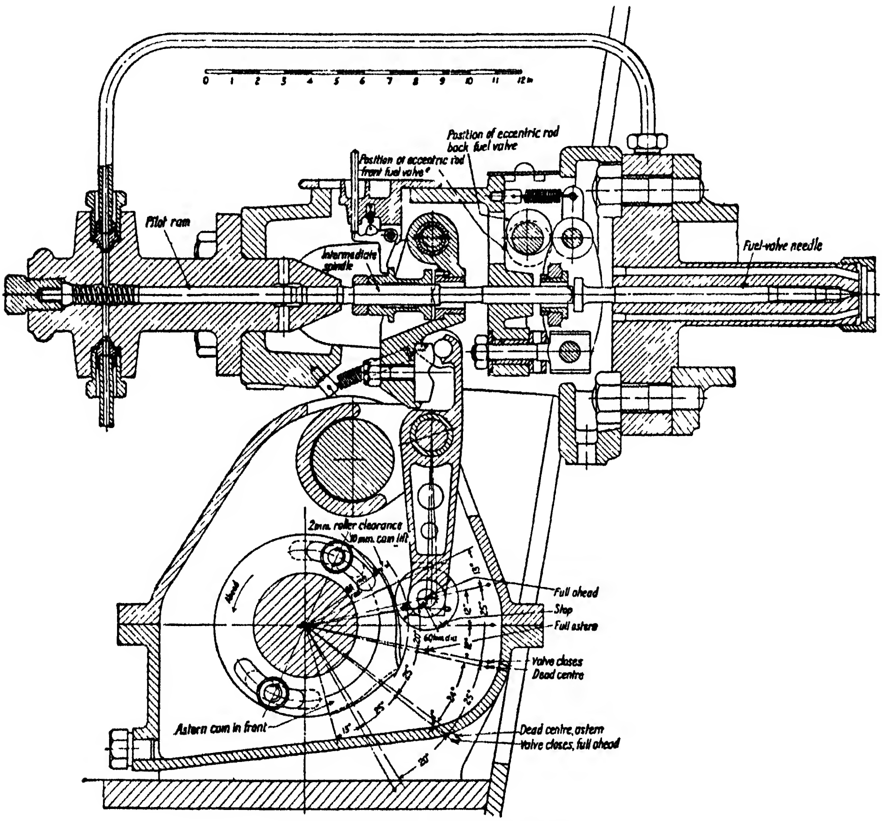



Fig. 59.—Arrangement of Fuel Valve

a chain drive. The pump suction valve is raised off its seat at each suction stroke by links and levers actuated by the pump crosshead. Governing is effected by the control lever seen on the extreme right of fig. 58.

Fig. 59 shows the fuel valve for the Doxford engine and the means of actuation.

Fig. 60 is a section of the fuel valve for the B. & W. type of engine manufactured by Messrs. Harland and Wolff, Ltd. The valve opens at a pressure of 4000 to 5000 lb. sq. in., and the oil speed through the nozzles is about 650 ft. per sec. There are 3 nozzle holes each 0·037 in. bore (cylinder top) and 0·0292 in. bore (cylinder bottom). The nozzle length is about 3 diameters. The nozzle is kept cool by circulating the incoming fuel oil around it, fins being provided.

**Starting.**—The starting air valves operated from the cam-shaft are so arranged that they can be put into gear while the fuel valves and fuel pumps remain cut out, thus allowing the engine to revolve by air pressure. After a few revolutions the air-starting valves are put out of action on certain cylinders, while at the same time the fuel valves and pumps supplying these cylinders are " cut in ", causing them to start operating on fuel. The remaining cylinders are then also put on fuel and the engine brought up to speed. These operations are usually arranged to take place automatically in sequence by moving a lever first to the starting position and then to the fuel position. For more detailed information regarding the starting up of engines, see section dealing with the operation of oil-engines (p. 303).

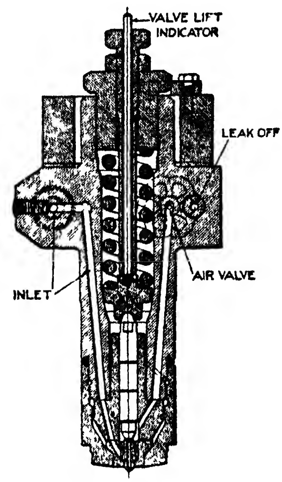


Fig 60.—Fuel Valve

# CHAPTER VIII

## Diesel-engined Ocean-going Vessels

The most striking event in the history of marine engineering during the first portion of this twentieth century has been the rapidity with which the Diesel engine has supplanted the reciprocating steam-engine as the means of propulsion of large ocean-going ships. The success attained and the confidence thereby created with the Diesel engine in land installations soon prompted engineers to study the problem of its application to marine propulsion, and as early as 1903 Messrs. Nobel's 1100-ton $7\frac{1}{2}$-knot boat *Vandal*, fitted with three reversible propellers each driven by a small Diesel engine, plied regularly on the Caspian Sea.

In 1909 the 60-ft. launch *Dreadnought* was fitted with a four-cylinder two-stroke single-acting Sulzer Diesel engine of 7·1 in. bore and 9·85 in. stroke, giving 85 b.h.p. at 345 r.p.m., and ran regularly on the Thames for some years. This engine used air-blast injection at 850 lb. per square inch pressure.

In 1910 the small sea-going single-crew vessel *Vulcanus* was fitted with a six-cylindered Diesel engine giving 390 b.h.p. at 140 r.p.m. But the first Diesel engines to be entrusted with the responsibility of driving a large ocean-going ship were those of the East Asiatic Company's *Selandia* (Copenhagen to Bangkok), which made its maiden trip early in 1912. This notable vessel was 370 ft. between perpendiculars, 53 ft. beam, and had a moulded depth of 30 ft. Passenger accommodation was provided amidships, and a deadweight cargo of 7400 tons was carried. There were twin screws each driven at 140 r.p.m. by an eight-cylindered single-acting four-stroke Burmeister and Wain Diesel engine of 20·875-in. bore and 28·75-in. stroke, of crosshead design, but with enclosed crank-chambers. Indicator diagrams at full power gave a mean effective pressure of 91 lb. per square inch at 129 r.p.m., corresponding to an aggregate indicated horse-power of 2336 for the sixteen cylinders (see equation (1) of Chapter IV). The exhaust was discharged into the mizzenmast, so that the vessel had no funnel.

Progress at once became accelerated, and even during 1912 fully a dozen large vessels were Diesel-engined, the largest being a twin-screw tanker of 10,800 tons gross displacement fitted with two six-cylindered single-acting two-stroke Krupp Diesel engines of 22·45-in. bore and 39·4-in. stroke, aggregating 3500 i.h.p.

In 1936 motor-ships were in service requiring a total power per vessel up to 24,000 b.h.p. The power developed per cylinder in the engines of these installations was approximately three times the total power of the *Vulcanus*. Cylinder diameters had increased from 15 in. to over 33 in., and the power developed per cylinder from 60 b.h.p. to over 1100 b.h.p.

These figures show the enormous advance made by the oil-engine during the years 1910 to 1936.

That the oil-engine has proved to be as reliable as the steam-engine for ship propulsion is clearly proved by the number of large passenger vessels now being provided with this type of engine.

In July, 1937, the total number of motor ships in service was 6763 (Motor Ship Reference Book for 1938), the gross tonnage being 13,748,713. In September, 1937, the tonnage of ships under construction was 1,236,348 (steam) and 1,649,722 (motor), the motor ship tonnage being 57·5 per cent of the total. On January 1st, 1938, the number of motor ships on order was 400, with a gross tonnage of 2,750,000. Over 20 per cent of the world's tonnage is now driven by Diesel engines (1938), the proportion showing a progressive increase.

**Oil Fuel.**—The Diesel engine is undoubtedly the marine machinery of the moment for the mercantile marine, and an analysis of the conditions to-day reveals the extreme potency of its claims. On board ship oil has proved the fuel *par excellence*, and so great are the savings, even with steam machinery, that where the price of oil and coal are as 3 to 1, the advantage still lies with the liquid fuel, and the larger the installation the greater the saving.

How much more insistent, then, is the appeal of the Diesel system, which only requires some one-half of the fuel of the best competing alternative, and generally for small- and medium-powered ships only requires in actual service one-third the liquid, or approximately one-fourth to one-fifth the amount of coal.

The significance of this low fuel consumption can well be brought out when it is stated that a motor-vessel of 14,000 tons d.w. can be propelled fully laden at 11 knots for the expenditure for all purposes of less than 13 tons of oil fuel per day, whereas a steamship burning coal under the same conditions would not consume much less than 50 tons of coal per day. This motor-ship will transport more than 3000 tons of cargo for one nautical mile at the expenditure for fuel of less than one shilling. The savings also in regard to personnel are not inconsiderable.

**Running Costs.**—When dealing with machinery costs per ton mile sailed, assuming a ratio of price of coal to price of oil 1 to 3, and allowing for a slightly higher price per ton of oil fuel in the case of Diesel machinery, excluding questions of first cost, but including wages and upkeep, the ratio for average-powered merchantmen can be stated as follows:

| | |
|---|---|
| Diesel engines, electric auxiliaries .. .. .. .. | 1·0 |
| Triple-expansion steam-engines, coal-fired boilers .. | 1·8 |
| Triple-expansion steam-engines, oil-fired boilers, 50° F. of superheat .. .. .. .. .. | 2·1 |
| Turbines and double-reduction gearing, oil-fired boilers, 200° F. of superheat .. .. .. .. | 1·5 |

The savings in port with motor-ships, especially when equipped with Diesel-electric auxiliaries, are of equal importance with those referred to at sea. Fortunately, concurrently with the introduction of the motor-ship, the electric hydraulic steering gear made its appearance, and has proved at sea to be reliable and much more economical than the usual steam gear. With oil-engines for the main propelling units, the auxiliaries required at sea can very conveniently be electrically driven, so that, except for steam heating of the cabins, no steam is required.

In recent years it has become common practice to fit steam boilers heated by the exhaust gases discharged from the engines, thus allowing a supply of steam and hot water sufficient for all ship heating and other purposes to be obtained without the expenditure of additional fuel. In port, when no exhaust gases from the main engines are available, the boiler can be fired by oil fuel.

**Oil Supplies.**—The question of the world's available supply of oil fuel naturally arises, and whereas many misleading statements are made, it is certain that wherever the propelling plant travels with the load it moves, as it does in all transport work, liquid fuel has very commanding advantages, and for marine work, where large quantities of fuel have to be carried, particularly so. For fixed stations on land, where the weight and space occupied by coal and ash conveying and handling plant are of little consequence, the whole balance is different. Authorities differ as to the amount of oil supply available. As sources become exhausted new fields open up. Conservation is undoubtedly called for, and the economical use, so far as possible, in the motor-ship is a means to that end. Oil fuel only represents in weight some 10 per cent of the world's total fuel supply. On a basis of calorific value, this 10 per cent is increased to over 13 per cent, and to a much higher fraction on the basis of power produced, if used in internal-combustion engines.

The major portion of the world's coal serves for process-work in the metal and chemical industries, so that the relative figure of importance of oil fuel for power production is much higher than the 13 per cent would suggest.

As an indication of the increase in oil production during the ten years between 1919 and 1929, the rise in tonnage of oil-carrying vessels in use might be cited. The figures are taken from Lloyd's Register book and show the oil tanker tonnage in the register at July, 1919, as 2,999,113 gross tons, and at July, 1929, as 7,070,015 gross tons, while during the period from 1929 to 1933 the tanker tonnage built amounted to 1,747,686 gross tons.

At the beginning of 1937 there were about 100 tankers on order with a dead weight capacity of practically 1,250,000 tons and for 1938 the orders in hand were of practically 2,000,000 tons dead-weight capacity, represented by about 150 ships (Motor Ship Reference Book for 1938).

All the oil fuel coming on the market at the present time is not suitable for combustion within the cylinders of oil-engines, so that the higher grades of oil should be conserved for that specific purpose. In time the Diesel oil-engine will be able to consume a wider range of qualities of fuel, without the cost of upkeep being an excessive part of the total running expenditure.

**First Costs.**—The motor-ship has made the progress indicated in spite of the cost of the machinery being very substantially greater than that of steam plants, because of the necessarily massive nature of its structure and main parts, and of the delicacy of the large number of important lesser organs. This fact has to be faced, although in the great majority of trade routes it can clearly be shown that the saving in operating costs is so great that the extra initial capital expenditure is rapidly written off, and in making a balance sheet to compare with a steamer, the fact that either the hull of the motor-ship is smaller and cheaper, or that the same hull fitted with oil-engines will carry additional cargo per annum, owing principally to the saving of bunkers, must be credited to the motor-ship.

For estimating purposes the motor-ship for the same duty will be 10 to 15 per cent more expensive than the steamer.

What of the limitations of the motor-ship? In the years 1917–1938 the power per cylinder of reliable and well-tried-out types of engines increased from 100 b.h.p. to over 1400 b.h.p. With 10 cylinders per engine this gives a maximum power per engine of 14,000 b.h.p. or a total power of 28,000 b.h.p. for a twin-screw vessel.

With the adoption of gearing between the engine and the propeller, two or more engines of relatively high speed of rotation may be coupled to each slow-running propeller shaft. With this arrangement powers for the largest types of vessels are available, while at the same time the weight of the engine parts permits of easy handling during overhaul.

**Maximum Power.**—Whilst the maximum power obtainable has gradually increased, there has been no radical change in thermodynamical theory or in practical application, but, as regards materials used, it should be noted that recently a fabricated steel construction has found favour, and is to a certain extent displacing cast iron for framing and bedplates. While the single-acting four-stroke cycle forced lubricated engine is still predominant, the double-acting four-stroke cycle engine has made some progress in recent years for high-power machinery, and a number of these are now in service.

Opinions are, however, still divided as to the future of this type, but further experience and experiment may justify the enterprise of those manufacturers who have done so much to develop it as a practical proposition for large powers. Difficulties with valve gear and with heat transmission appear to be the main obstacles to development.

Above 400 b.h.p. per cylinder the single-acting four-cycle engine becomes extremely massive, and the various parts heavy and difficult to handle during overhaul, which is not surprising when it is considered that the structure and main parts of the engine are only stressed comparable with their scantlings for one-eighth of their running time, and of the load they are designed to carry only one-seventh is really useful. The ratio of maximum to mean pressure in the cylinder is about 7 to 1. Designers have been led by these considerations to adopt the two-stroke cycle principle, the opposed-piston engine, and the double-acting cylinder. The use of reduction gearing between the engine and the propeller shaft permits large outputs per shaft to be obtained without unduly large engine parts having to be handled during overhaul, but has made little progress in recent years.

**The Future.**—The motor-ship has proved to be an economical type of cargo carrier, and it is possible that if the present rate of progress is continued the Diesel engine will ultimately replace the steam-engine as the propelling machinery for cargo-carrying vessels. The apathy with which we were previously charged has disappeared, and under the stimulating conditions of keen competition we may be fully hopeful of maintaining, with fleets of maximum utility and operating economically, our place in the cargo-carrying trade of the world.

**Principal Marine Engines.**—Tables V and VI on pp. 214, 215 give the leading particulars of motor-ships, representing some of the earliest and the best known and of the latest types at sea. Full particulars are given relative to the propelling machinery, and all types are represented.

Later instances are quoted farther on in the text.

It will be noticed that from 1927 double-acting engines of the four-stroke and two-stroke cycle have entered the field and so have made possible the large-powered units required for the larger type of high-speed vessels.

## TABLE V

| Date. | Name of Vessel | Tonnage. | Dimensions in Feet. | | | Makers of Machinery. | Type of Engine. |
|---|---|---|---|---|---|---|---|
| | | Tons. | Length. | Beam. | Depth. | | |
| 1910 | *Vulcanus* | 1,900 dis. | 208 | 37¾ | 13 | Werkspoor | Werkspoor |
| 1912 | *Selandia* | 10,000 dis. | 386 | 53 | 30 | Burmeister & Wain | Burmeister & Wain |
| 1912 | *Monte Penedo* | 6,500 dis. | 350 | 50 | 27 | Sulzer | Sulzer |
| 1913 | *Siam* | 13,200 dis. | 410 | 55 | 30½ | Burmeister & Wain | Burmeister & Wain |
| 1913 | *Hagen & Loki* | 7,900 d.w.c. | 400 | 53 | 32½ | Krupp | Krupp |
| 1914 | *Fionia* | 7,000 d.w.c | 395 | 53 | 30 | Burmeister & Wain | Burmeister & Wain |
| 1916 | *Trefoil* | 4,510 dis. | 280 | 39 | 23¼ | Vickers | Vickers |
| 1916 | *Hamlet* | 10,050 dis. | 360 | 56 | 29¾ | Atlas Diesel, Sweden | Polar |
| 1918 | *Ansaldo San Giorgio I* | 8,100 d.w.c. | 378 | 50⅜ | 30½ | Ansaldo San Giorgio | F.I.A.T. (now Ansaldo San Giorgio) |
| 1919 | *Glenapp* | 19,000 dis. | 450 | 55⅝ | 36½ | Harland & Wolff | Burmeister & Wain |
| 1920 | *Glenogle* | 19,000 dis. | 502 | 62 | 27½ | Harland & Wolff | Burmeister & Wain |
| 1920 | *Fritz* | 4,000 d.w.c. | 331 | 44⅜ | — | Blohm & Voss | M.A.N. |
| 1920 | *Maumee* | 15,000 dis. | 455 | 56 | 26 | New York Naval Yard | M.A.N. |
| 1920 | *Cubore* | 17,000 dis. | 450 | 57 | 37 | Bethlehem Steel Co. | Bethlehem West |
| 1920 | *Zoppot* | 22,000 dis. | 525¾ | 66½ | 33½ | Krupp | Krupp |
| 1920 | *Salerno* | 6,500 d.w.c. | 375 | 51 | 34 | Werkspoor | Werkspoor |
| 1920 | *Narragansett* | 14,000 dis. | 425 | 56⅜ | 33 | Vickers | Vickers |
| 1920 | *Fullagar* | 500 d.w.c. | 150 | 23¾ | 11¼ | Cammell Laird | Fullagar |
| 1920 | *Ansaldo San Giorgio III* | 8,100 d.w.c. | 394 o.a. | 51¾ | 31 | Ansaldo San Giorgio | Ansaldo |
| 1921 | *Domala* | 10,500 d.w.c. | 450 | 58 | 35½ | N. B. Diesel Co. | N. B. Diesel |
| 1921 | *Sardinia* | 3,500 d.w.c. | 302 | 42 | 20½ | Werkspoor | Werkspoor |
| 1921 | *Pinzon* | 2,000 d.w.c. | 240 | 38 | 18 | Beardmore & Co. | Beardmore-Tosi |
| 1921 | *Dalgoma* | 11,000 d.w.c. | 450 | 54½ | 25½ | Stephen & Sons | Sulzer |
| 1921 | *Scottish Borderer* | 15,000 dis. | 425 | 56⅜ | 33½ | Denny Bros. | Sulzer |
| 1921 | *Malia* | 6,000 d.w.c. | 350 | 49¾ | 37 | Cammell Laird | Fullagar |
| 1921 | *Yngaren* | 12,750 dis. | 420 | 54 | 37 | Doxford | Doxford |
| 1921 | *Suphenco* | 5,350 d.w.c. | 324 | 46 | 28½ | James Craig Engine Works, U.S.A. | Craig |

## TABLE VI

| Date. | Name of Vessel. | Tonnage. | Dimensions in Feet. | | | Type. | Single or Twin |
|---|---|---|---|---|---|---|---|
| | | Tons. | Length. | Beam. | Depth. | | |
| 1922 | *Adriana* | 6,800 d.w.c. | 340$\frac{5}{16}$ | 51½ | 31½ | F.I.A.T. | Twin |
| 1923 | *Scottish Borderer II* | 10,000 d.w.c. | 426$\frac{5}{16}$ | 56¾ | 33½ | Sulzer | Twin |
| 1924 | *Moveria* | 7,500 d.w.c. | 385½ | 51½ | 28½ | Vickers | Single |
| 1924 | *Henry Ford II* | 8,877 gross | 586 | 62 | 32 | Sun-Doxford | Single |
| 1925 | *Bintang* | 9,700 d.w.c. | 420 | 54$\frac{1}{16}$ | 33½ | Sulzer | Single |
| 1925 | *Asturias* | 22,500 gross | 630 | 78 | 45 | B. & W. | Twin |
| 1925 | *Aorangi* | 17,491 gross | 580$\frac{1}{16}$ | 72½ | 46½ | Sulzer | Quadruple |
| 1926 | *Tampa* | 9,120 d.w.c. | 402 | 54 | 33$\frac{7}{16}$ | Worthington | Single |
| 1926 | *Ramses* | 11,500 d.w.c. | 482 | 62$\frac{5}{16}$ | 32 | M.A.N. | Single |
| 1927 | *Christiaan Huygens* | 15,636 gross | 550 | 68$\frac{5}{16}$ | 39$\frac{7}{16}$ | Sulzer | Twin |
| 1927 | *Augustus* | 33,000 gross | 666$\frac{5}{16}$ | 82½ | 47½ | M.A.N. | Quadruple |
| 1928 | *Highland Monarch* | 14,137 gross | 520 | 69 | 35¾ | B. & W. | Twin |
| 1929 | *Sud Expreso* | 8,263 d.w.c. | 460 | 60½ | 40 | Deutsche-Werke | Twin |
| 1929 | *Ulster Monarch* | 3,760 gross | 345 | 46 | 19 | B. & W. | Twin |
| 1929 | *Rangitiki* | 16,750 gross | 530 | 70½ | 43½ | Sulzer | Twin |
| 1930 | *Amerika* | 12,000 d.w.c. | 465 | 62 | 40 | B. & W. | Single |
| 1930 | *Kato Agoeng* | 9,640 d.w.c. | 448½ | 60½ | 33½ | M.A.N. | Single Geared |
| 1931 | *Reina del Pacifico* | 17,300 gross | 550 | 76 | 44 | B. & W. | Quadruple |
| 1932 | *Prince Baudouin* | 2,760 displ. | 355 | 46 | 24¾ | Sulzer | Twin |
| 1933 | *Malaita* | 3,500 d.w.c. | 312 | 47 | 23½ | B. & W. | Single |
| 1933 | *British Coast* | 1,400 d.w.c. | 230. | 35 | 21½ | Atlas Polar | Twin |
| 1933 | *Devon City* | 9,400 d.w.c. | 425 | 56½ | 25½ | Doxford | Single |
| 1934 | *Nagara Maru* | 9,650 d.w.c. | 450 | 62 | 34¾ | M.A.N. | Single |
| 1935 | *Port Wyndham* | 11,400 d.w.c. | 492 | 65 | 29·9 | Brown Doxford | Twin |
| 1935 | *Imperial Star* | 10,670 gross | 516·75 | 70 | 43·3 | Harland B. & W. | Twin |
| 1936 | *Stirling Castle* | 25,564 gross | 680 | 82 | 48 | Harland B. & W. | Twin |

TABLE V (*continued*)

| Single or Twin. | Cycle.* | I.H.P. per Engine. | B.H.P. per Engine. | Number of Cylinders per Engine. | B.H.P. per Cylinder. | Diameter of Cylinders. In. | Stroke. In. | Ratio Stroke/Bore. | R.P.M. | Piston Speed, Feet per Minute. | M.E.P. on B.H.P. Basis. | M.I.P. on I.H.P. Basis. |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Single | 4 S.A. | 490 | 390 | 6 | 65 | 15¾ | 23⅝ | 1·5 | 140 | 550 | 79 | 99 |
| Twin | 4 S.A. | 1250 | 1050 | 8 | 130 | 20½ | 28⅜ | 1·38 | 140 | 670 | 75 | 89·8 |
| Twin | 2 S.A. | 1200 | 850 | 4 | 212 | 18¼ | 26⅜ | 1·44 | 160 | 715 | 73 | 103 |
| Twin | 4 S.A. | 1580 | 1350 | 8 | 169 | 23⅝ | 31⅝ | 1·34 | 125 | 660 | 77 | 90 |
| Twin | 2 S.A. | 1600 | 1200 | 6 | 200 | 18⅞ | 31⅞ | 1·69 | 135 | 720 | 66 | 88 |
| Twin | 4 S.A. | 2140 | 1600 | 6 | 266 | 29⅛ | 43¼ | 1·49 | 100 | 720 | 74 | 99 |
| Twin | 4 S.A. | 1000 | 750 | 8 | 94 | 17 | 27 | 1·59 | 150 | 675 | 81 | 108 |
| Twin | 2 S.A. | 2300 | 1650 | 6 | 275 | 24 | 36 | 1·5 | 120 | 720 | 67 | 93·5 |
| Twin | 2 S.A. | 1450 | 1100 | 4 | 275 | 24¾ | 35½ | 1·43 | 100 | 590 | 64 | 84·5 |
| Twin | 4 S.A. | 3200 | 2625 | 8 | 328 | 30 | 43¼ | 1·44 | 115 | 830 | 74 | 90·5 |
| Twin | 4 S.A. | 3200 | 2625 | 8 | 328 | 29⅛ | 45¼ | 1·55 | 115 | 865 | 75 | 91·5 |
| Twin | 2 D.A. | 1250 | 850 | 3 | 283 | 18⅞ | 28 | 1·48 | 110 | 515 | 65 | 95·5 |
| Twin | 2 S.A. | 3200 | 2500 | 6 | 416 | 25¼ | 37⅜ | 1·48 | 130 | 810 | 69 | 88 |
| Single | 2 S.A. | 3900 | 2700 | 6 | 450 | 25⅝ | 48 | 1·87 | 100 | 800 | 73 | 105 |
| Twin | 2 S.A. | 2000 | 1400 | 6 | 233 | 22¾ | 39⅜ | 1·73 | 106 | 695 | 55 | 79 |
| Twin | 4 S.A. | 1400 | 1050 | 6 | 175 | 22 | 39⅛ | 1·78 | 125 | 820 | 74 | 98·5 |
| Twin | 4 S.A. | 1620 | 1250 | 6 | 208 | 24½ | 39 | 1·59 | 118 | 767 | 76 | 99 |
| Single | 2 O.P. | 660 | 500 | 4 | 125 | 14 | 20 × 2 | 1·43 | 110 | 367 | 73·5 | 92 |
| Twin | 2 S.A. | 1600 | 1200 | 4 | 300 | 24$\frac{13}{16}$ | 35$\frac{7}{16}$ | 1·43 | 110 | 650 | 63 | 84 |
| Twin | 4 S.A. | 2330 | 2000 | 8 | 250 | 26½ | 47 | 1·77 | 96 | 750 | 79·5 | 92 |
| Single | 4 S.A. | 2140 | 1600 | 6 | 266 | 26⅜ | 47¼ | 1·79 | 100 | 788 | 82 | 110 |
| Single | 4 S.A. | 1600 | 1200 | 6 | 200 | 24⅝ | 38⅝ | 1·57 | 120 | 767 | 74 | 98 |
| Twin | 2 S.A. | 2200 | 1600 | 4 | 400 | 26¾ | 43$\frac{5}{16}$ | 1·62 | 85 | 610 | 76 | 105 |
| Twin | 2 S.A. | 1700 | 1250 | 4 | 312 | 23¾ | 37 | 1·56 | 100 | 615 | 76 | 103 |
| Twin | 2 O.P. | 1580 | 1200 | 4 | 300 | 18½ | 25 | 1·35 | 115 | 480 | 77 | 101 |
| Single | 2 O.P. | 3000 | 2700 | 4 | 675 | 22$\frac{13}{16}$ | 45⅝ | 2·0 | 77 | 585 | 93 | 103 |
| Single | 4 S.A. | 2240 | 1850 | 6 | 308 | 30 | 48 | 1·6 | 105 | 840 | 68·5 | 82 |

TABLE VI (*continued*)

| Cycle.* | I.H.P. per Engine. | B.H.P. per Engine. | Number of Cylrs. per Engine. | B.H.P. per Cylinder. | Diameter of Cylinders. In. | Stroke. In. | Ratio Stroke/Bore. | R.P.M. | Piston Speed, Ft./Min. | M.E.P. on B.H.P. Basis. | M.I.P. on I.H.P. Basis. |
|---|---|---|---|---|---|---|---|---|---|---|---|
| 2 | — | 1000 | 4 | 250 | 21·25 | 29·94 | 1·41 | 150 | 748 | 62·2 | — |
| 2 | — | 1250 | 4 | 312·5 | 23·63 | 41·63 | 1·76 | 100 | 694 | 67·8 | — |
| 4 | 3,500 | 2550 | 8 | 318·8 | 30·0 | 45·0 | 1·5 | 110 | 825 | 72·3 | 99·0 |
| 2 O.P. | — | 3000 | 4 | 750 | 23·62 | 45·5 | 1·92 | 85 | 645 | 87·4 | — |
| 2 | — | 3600 | 6 | 600 | 29·92 | 52·76 | 1·76 | 90 | 791 | 71·4 | — |
| 4 D.A. | 10,000 | 7500 | 8 | 937·5 | 33·07 | 59·05 | 1·79 | 115 | 1130 | 67·0 | 89·3 |
| 2 S.A. | — | 3250 | 6 | 541·7 | 27·50 | 39·0 | 1·42 | 127 | 825 | 72·8 | — |
| 2 D.A. | 3,980 | 2900 | 4 | 725 | 28·0 | 40·0 | 1·43 | 95 | 634 | 64·6 | 88·5 |
| 2 D.A. | — | 4400 | 6 | 733·3 | 27·50 | 47·50 | 1·73 | 84 | 665 | 64·5 | — |
| 2 S.A. | 7,440 | 5800 | 10 | 580 | 26·75 | 47·25 | 1·76 | 115 | 905 | 75·3 | 96·4 |
| 2 D.A. | 7,800 | 6250 | 6 | 1041·7 | 27·50 | 47·20 | 1·72 | 120 | 944 | 64·6 | 80·75 |
| 4 D.A. | 5,330 | 4000 | 8 | 500 | 26·77 | 63·0 | 2·35 | 100 | 1050 | 58·9 | 78·5 |
| 2 D.A. | — | 4200 | 6 | 700 | 25·59 | 41·34 | 1·61 | 115 | 792 | 59·7 | — |
| 4 S.A. Trunk | — | 8000 | 10 | 800 | 24·8 | 38·5 | 1·55 | 160 | 1027 | 79·9 | — |
| 2 S.A. | 6,700 | 5000 | 5 | 1000 | 35·43 | 62·99 | 1·78 | 90 | 945 | 70·8 | 95·0 |
| 2 D.A. | — | 6000 | 6 | 1000 | 24·4 | 55·12 | 2·26 | 100 | 918 | 83·5 | — |
| 2 D.A. | — | 2750 | 5 | 550 | 20·47 | 27·56 | 1·34 | 215 | 988 | 60·0 | — |
| 4 S.A. Trunk | 6,550 | 5500 | 12 | 458 | 24·8 | 47·24 | 1·9 | 145 | 658 | 109·4 | 132 |
| 2 S.A. | — | 8500 | 12 | 708 | 22·84 | 33·07 | 1·45 | 268 | 1480 | 77·0 | — |
| 2 S.A. | — | 1740 | 8 | 218 | 19·68 | 35·43 | 1·8 | 110 | 650 | 72·7 | — |
| 2 S.A. | — | 625 | 5 | 125 | 13·88 | 22·44 | 1·67 | 220 | 825 | 71·0 | — |
| 2 O.P. | 3,300 | 2900 | 4 | 725 | 23·62 | Comb. 91·34 | — | 92 | Mean 700 | 78·0 | 88·6 |
| 2 D.A. | — | 7900 | 7 | 1130 | 27·5 | 47·75 | 1·72 | 116 | 920 | 69 | — |
| 2 O.P. | — | 4700 | 4 | 1175 | 27·5 | Comb. 88·5 | — | 110 | Mean 810 | 68 | — |
| 4 S.A. | — | 6000 | 10 | 600 | 29 | 59 | 2·03 | 112 | 1100 | 108 | — |
| 2 D.A. | — | 12000 | 10 | 1200 | 23·5 | 59 | 2·5 | 102 | 1000 | 91 | — |

* S.A. = Single Acting. D.A. = Double Acting. O.P. = Opposed Piston.

**Marine Auxiliary Machinery.**—The auxiliaries to the main propelling Diesel engine may now be discussed in detail.

(*a*) A water pump is required to supply the cooling water for the various jacketed parts of the engine.
(*b*) If fresh water is used to cool the pistons, an additional pump has to be provided for this purpose.
(*c*) A lubricating oil pump is required for the supply of oil under pressure for the forced lubrication system of the engine and for cooling the pistons when oil is used for this purpose.
(*d*) An air compressor is required to supply starting air and in the air injection engine the blast air for fuel injection. (This machine will be described later.)

These machines may be driven by the main engine, but this practice is gradually becoming obsolete and separate units driven by electric motors or oil-engines are now general.

In addition to the above machinery, there are the usual ship's auxiliaries, viz. the electric generators for supplying electric current for lighting, and power for driving engine-room pumps, winches, steering-gear, &c.

Three methods are available by which the auxiliary machinery in a vessel may be driven:

1. By steam supplied by a boiler.
2. By compressed air supplied from the ship's compressed air plant.
3. By electric current supplied by dynamos.

No. 1 has had its strong adherents, and had the advantage of utilizing simple machinery with a low first cost. In many modern cases boilers generating steam from the heat of the waste gases are fitted, the steam from these being for heating purposes and for driving certain of the auxiliaries. When the main engines are not running these boilers are fired direct by oil fuel.

No. 2 has only been tried in a few cases, and generally without much success, partly due, it may be, to the fact that suitable machinery for using compressed air has not been seriously developed. Generally it has been found that this method of drive is expensive, and to obtain reasonable efficiency arrangements must be made for preheating the air.

No. 3 is that now generally used, the generators being driven by Diesel engines, usually of the trunk piston type and ranging in output from 50 to 500 kw. per set. An engine of this type is shown in fig. 61.

For the average cargo-carrying vessel, the normal arrangement is to have three generators of equal power, so proportioned that one is capable of driving all the auxiliary machinery required at sea. Two engines together serve to take the load when the vessel is manœuvring in and out of port and when working cargo in port.

This arrangement gives the third generator as a standby.

In the larger passenger vessels where the electric load, due to lighting,

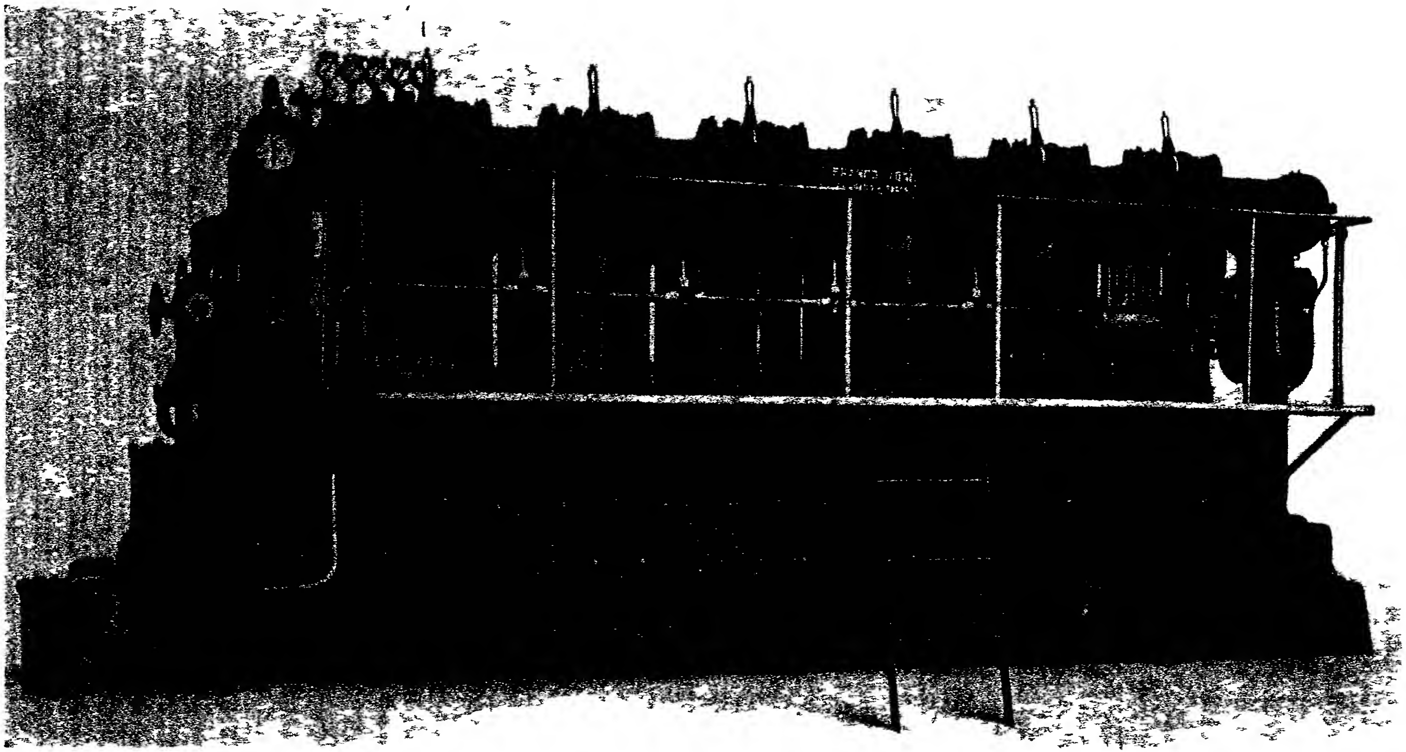


Fig 61 —Franco Tosi Diesel Engine

hotel services, and engine room, is heavy, as many as five generators may be fitted, but it should always be arranged that ample standby palnt is available.

There has been a tendency in recent years to utilize the so-called high-speed engine (see High-speed Diesel Engine) for electric-generator driving on board ship, thus obtaining a considerable reduction in weight and space occupied. These engines have been found to give very satisfactory service. As already mentioned, the pumps required by the main engine, such as the circulating water, piston cooling, lubricating oil, may be driven from the main engine itself. This, in measure, follows the corresponding earlier steam practice, but just as with steam machinery, so with modern Diesel installations it has been found advantageous, especially with larger powers, to drive these pumps separately by electric motors, in all cases installing such reserves that the failure of one pump will not put the main machinery out of action.

The advantages and disadvantages of these two systems of separate or engine-driven auxiliaries need not here be detailed, but the ease of regulation, the possibility of ensuring, for instance, that the water or oil services are working efficiently before putting the main engines into operation, the facility for reducing the quantity of oil or water according to the temperature of the sea and the power being developed by the main engines, and so forth, all lead to the adoption of the somewhat more expensive, more complicated, and more elaborate installation of separate auxiliaries as the better solution.

**Compressors.**—For the injection of the fuel into the main cylinders of a Diesel engine two alternative systems are in vogue; the first utilizes a high pressure of fuel, and the energy of this fuel expanding from 4000 to 10,000 lb. per square inch down to the compression pressure in the cylinder of 400 to 500 lb. per square inch, for atomizing the fuel and spreading it throughout the combustion chamber to ensure ready ignition and good combustion. The second system utilizes the energy of a jet of air expanding from 800 to 1000 lb. per square inch down to the compression pressure to force into the cylinder the quantity of fuel for combustion, there to atomize it and to spread it throughout the combustion chamber.

Up to 1930 the general system for both land and marine service was the latter method of air injection, as described. Since 1930, however, the solid or airless injection system has rapidly come forward, and is now practically standard practice, allowing the engine to be simplified by the elimination of the blast air compressor.

When air injection is fitted the modern compressor gives no less degree of reliability than the main engines themselves, and from some points of view air injection had certain advantages, as, for instance, in the engine so fitted being more able to use oil fuels of varying grades satisfactorily without special adjustments. Air injection also has a cooling action on the combustion chamber walls, and serves to clean the seats of the valves in the cylinder head, ensuring good combustion even when considerable derangement of the fuel feeding mechanism has taken place. With airless injection the fuel feeding mechanism must be accurately timed and kept in first-class order if good combustion is to be ensured.

The compressor for supplying injection air is now almost universally driven from the main engine itself. In earlier vessels this machine was frequently separately driven, but it is now preferred, in distinction to the system advocated for general auxiliaries, to drive it from the main engine itself, because of the low speed of revolution of the main engine. The reliability of compressors is much improved by slow-speed running. Less attention is required when the compressor runs at the same speed as the

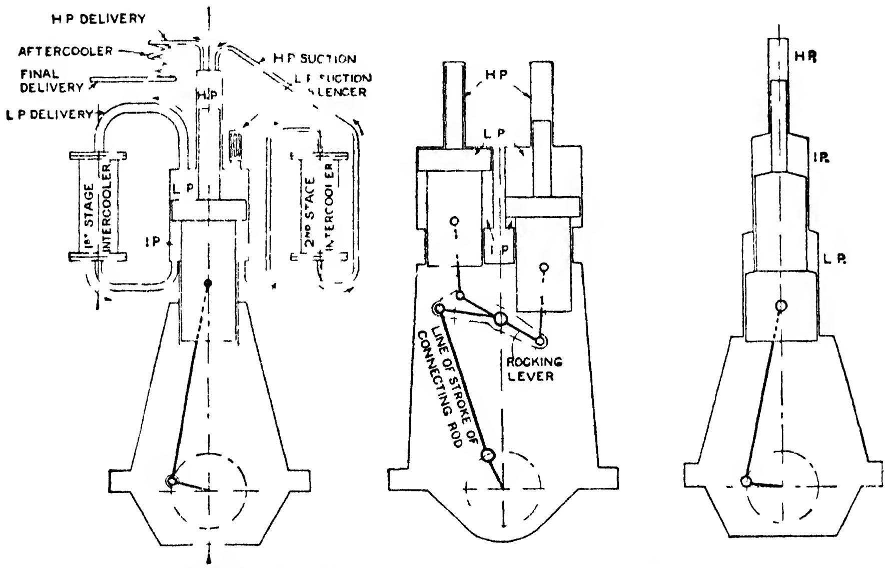



Fig. 62 —Diagrammatic Arrangement of Three Types of Vertical Three-stage Air Compressors for Fuel Injection with Marine Diesel Engines

main engine, since it automatically delivers to the main engine exactly that quantity of air required for the injection of fuel demanded by the load at which the main engine is running. The air, however, for injection must be compressed from atmospheric to from 800 to 1000 lb. per square inch, and to do so without undue rises in temperature in the compressor cylinders, which would make for carbonizing of the lubricating oil, deterioration of the springs, &c., necessitates that this process shall be carried out in not less than three stages (except in compressors of the smallest size).

With earlier Diesel engines these compressors were often of the two-stage variety, and in such cases, with stages ill-proportioned and with the compressions greatly differing from those of the design where normally equal compressions in each stage are aimed at, the conditions were such as to give rise to extreme unreliability, and were the original cause of the introduction of the alternative system of airless injection. Various arrangements of compressor cylinders are shown in fig. 62.

An auxiliary compressor must always be installed to serve as a stand-by in the event of a breakdown to the main engine compressor, and to charge the reservoirs with compressed air, which is always used for starting the main engine, except in the case of the Still engine, where steam is used. However, it has now become standard practice to dispense with the stand-by compressor and provide for the necessary stand-by supply of air by enlarging the air compressors on the Diesel generator engines. By arranging these compressors with variable output, surplus air over the requirements of the generator engines can be obtained as desired.

Among the notable Diesel-engined vessels of 1927 were:

(1) The *Saturnia*, 23,900 tons gross, 19 knots, fitted with two eight-cylindered double-acting four-stroke supercharged Diesel engines of 33-in. bore and 59-in. stroke, aggregating 20,000 h.p.

(2) The *Augustus*, 32,650 tons gross, 19 knots, fitted with four double-acting two-stroke M.A.N. Diesel engines aggregating normally 25,000 b.h.p. Further particulars of this important case are given later in this chapter, but to indicate the regularity of running of these engines some particulars of voyages made by the *Augustus* are given on p. 221.

Both of these ships are still in service (1938).

Ordinary merchant ships are mainly included in the 3000–5000-ton cargo-carrying class, and when steam-driven, with coal as fuel, the average consumption is about 1·8 lb. of coal per b.h.p. hour. The modern marine Diesel gives a b.h.p. hour for about 0·41 lb. of oil. Thus the weight of Diesel fuel is less than one-quarter that required by an equal coal-burning steam-engined vessel.

A further advantage of the Diesel is that its efficiency increases somewhat from full to half-load, whereas with the steam-engine the reverse is the case. There are no stand-by losses with the Diesel engine when in port. And, again, a ship's auxiliary engines often aggregate 20–25 per cent of the power of the main engines and, when steam-operated, are usually notoriously wasteful; much economy results here from the employment of Diesels or semi-Diesels as auxiliaries, particularly when one largish engine of this type is employed to drive a generator supplying current to electric motors driving all the auxiliary machinery.

Practically the same power is needed whether a ship be Diesel- or steam-driven: thus relative fuel costs can quickly be roughly estimated. It is clear that so long as Diesel oil does not exceed in cost about four times that of the average of the ship's coal, the motor-ship in this respect shows economy. In considering the average cost of the coal it is to be remembered that most steamers have to take coal not only when in British ports, but also at foreign ports at greatly increased prices, whereas the much greater radius of action of the motor-driven vessel renders foreign buying less often necessary, and also enables purchases to be made at ports where prices are most favourable.

Since for a given voyage with Diesel engines only about one-fourth

Voyage Particulars of the Motor Liner "Augustus" *

| Trip. | Date. | Distance, sea miles. | Average r.p.m. of Main Engines. | Average Total Load of Four Main Engines, B.H.P. |
|---|---|---|---|---|
| Genoa to Buenos Aires .. | 11.11.27<br>26.11.27 | 6283 | 119·5 | 25,120 |
| Buenos Aires to Genoa .. | 3.12.27<br>19.12.27 | 6217 | 110·0 | 19,050 |
| Genoa to Buenos Aires .. | 12.1.28<br>27.1.28 | 6170 | 119·45 | 23,710 |
| Buenos Aires to Genoa .. | 1.2.28<br>16.2.28 | 6200 | 121·17 | 25,665 |
| Genoa to Buenos Aires .. | 24.5.28<br>8.6.28 | 6182 | 116·5 | 24,005 |
| Buenos Aires to Genoa .. | 14.6.28<br>30.6.28 | 6135 | 117·1 | 24,200 |
| Genoa to Buenos Aires .. | 12.7.28<br>26.7.28 | 6169 | 116·9 | 24,530 |
| Buenos Aires to Genoa .. | 31.7.28<br>16.8.28 | 6108 | 117·73 | 25,430 |
| Genoa/Naples to New York | 28.8.28<br>7.9.28 | 4499 | 117·8 | 25,320 |
| New York to Naples/Genoa | 15.9.28<br>25.9.28 | 4523 | 117·0 | 25,000 |
| Genoa/Naples to New York | 2.10.28<br>12.10.28 | 4623 | 117·14 | 25,150 |
| New York to Naples/Genoa | 20.10.28<br>30.10.28 | 4535 | 117·2 | 25,180 |

of the weight of coal is required, very valuable bunker space is saved, and becomes available as revenue-earning cargo room in a merchant ship; there is the further point that Diesel oil can be stored where coal can not, as e.g. in the double bottom tanks. In warships, the effective range of action can be quadrupled if all bunker space is oil-filled, and this is a consideration of the greatest importance.

Two additional points of interest may be mentioned: (1) of two similar vessels, one Diesel and one coal-using steam-engined, the Diesel usually shows a somewhat higher average speed in service due to the maintenance of a more constant engine speed in all weathers, and also to no loss of speed being incurred from the necessity of making up fires periodically; and (2) the fuel consumption of the Diesel engine, so far from increasing with length of service, not uncommonly is found to *decrease*, even after upwards of twelve years running. With coal-fired steamers, however, fuel consumption shows a marked increase with long service.

* Reproduced by courtesy of *The Marine Engineer.*

As regards personnel also the Diesel-engined ship shows well. Thus a 2000-h.p. 6000-ton steamer requires a total, including stokers, of some fifteen men, whereas if Diesel-engined seven only suffice.

**Weight, Space, and Cost.**—The average weight of the slow-speed marine Diesel engine may be taken as ranging from about 350 lb. per b.h.p. in the single-acting four-stroke type, to roundly 200 lb. per b.h.p. in the case of double-acting two-stroke installations. With supercharged engines these figures are reduced by about 10 per cent. With smaller quicker-running designs weight is of course much reduced; thus, supercharged four-stroke single-acting trunk piston engines running at 350 r.p.m. weigh only some 90 lb. per b.h.p., while in the Ricardo-Brotherhood Diesel, running at 900 r.p.m., the weight per b.h.p. has been reduced to 47 lb. only (see fig. 10, p. 146).

The space occupied by a normal slow-running marine Diesel installation, including all accessory apparatus, is much less than that required by the corresponding complete steam plant, while with the quicker-running type of Diesel a still greater saving of space can be effected.

The cost of a marine Diesel equipment considerably exceeds that of either the corresponding reciprocating steam-engine plant, or geared steam-turbine installation. The Diesel engine and its accessories call for great refinements in manufacture owing to the high pressures employed; when, however, calculation of cost is based on the cost per ton of weight carried, the Diesel-engined vessel can, in general, be shown to be the more economical. Marine Diesel engines are either of the single- or double-acting four-stroke, or of the single-acting two-stroke type, with or without supercharging equipment.

The four-stroke types probably still show slightly higher economy in fuel than the two-stroke, but the latter possess the important advantages of reduced size and weight, simpler control, and lower cost.

Up to about 1920 the single-acting four-stroke engine held pride of place in marine service; the early two-stroke marine Diesel, particularly when of the valve-scavenging type, had given much trouble, and delayed its progress; a more extended experience showed that with port-scavenging an entirely satisfactory two-stroke marine engine could be made, and larger engines of this type became common about 1924, when shipping companies perceived that the Diesel engine possessed even greater advantages in the case of fast cargo liners than in slower-running vessels. The competition of the single-acting two-stroke marine engine stimulated four-stroke builders to produce reliable double-acting engines, and this again was countered by the appearance of satisfactory double-acting two-stroke designs, the final step in the direction of obtaining maximum power output per cylinder being made by the provision of supercharging equipment.

A notable feature of recent years is the increased speed of many of the newer motor-engined ships; the old 10-knot or 12-knot steamer is being challenged by the "cargo liner"; since 1929 many large vessels of this type have been constructed, designed for a normal service speed of 14 to

15 knots. For this service the choice usually lies between the single-acting two-stroke or the double-acting two-stroke type.

The type of engine selected is largely governed by the space available when the best arrangement of the cargo holds and passenger accommodation has been decided upon.* In passenger ships space above the waterline is especially valuable and thus the whole of the above-water decks should, if possible, be available for accommodation, while length of engine-room is of minor importance. To avoid breaking deck, therefore, a long low type of engine is desirable, and this is best obtained with the single-acting two-stroke type; if double-acting engines of normal piston speed are adopted in such cases, the whole engine-room length of at least one deck, and perhaps even of two if the engine is of the four-stroke type, is lost. The loss of this extremely valuable passenger accommodation may easily exceed any saving claimed from the use of the double-acting engine.

With cargo vessels the problem is simplified, as all space is of about equal value, so that the only really essential condition binding the engine designer is that the volume occupied by the propelling machinery and its accessories should be a minimum. For such cases the four-stroke double-acting engine requires most space, while of two-stroke designs, the double-acting enjoys a small advantage in both space occupied and weight.

Marine engines were for long equipped with high-pressure air-blast injection, air pressures of as much as 1000 lb. per square inch being employed, involving the provision of high-pressure air-compressing pumps and stout steel "air-bottles". Though the air blast exercised a cooling effect on the cylinder contents, the fuel was very effectively sprayed and complete combustion ensured, resulting in maximum economy of fuel used. By 1929, however, a marked tendency became noticeable towards the adoption of the simpler "airless" method of injection involving only the provision of a very small fuel-injection force pump actuated by a quick-acting cam, by which the fuel is forced through the spraying jet under a pressure of several thousands of pounds per square inch, and this simpler injection method is now almost universal. The first to achieve success in marine service with airless injection were Messrs. Doxford; and it is worthy of note that most of the former adherents of air-blast injection now fit airless injection to all types.

**Types of Marine Oil-Engines.**—A considerable amount of experimental work has been done in past years with different types of engines, involving large expenditure of capital. Naturally disappointments and set-backs have been experienced, and shipbuilders, engine-builders and owners alike are to be congratulated on their courage in trying out new types of engine. The aims of designers of new types are broadly reduction of weight and space occupied per horse-power, reliability in operation, freedom from vibration and minimum costs of upkeep and overhaul. The possibilities of important savings in fuel and operating costs as compared with steam plants and of increase of available cargo space have stimulated research

* See Calderwood, *Proc. Inst. Marine Engineers*, 1929.

and experiment, and the large and increasing number of motor-ships in operation, with horse-powers of from 1200 to 23,000, are evidence of the success attained.

Engine types may be classified as follows:

1. *Single-acting four-stroke cycle* (with or without supercharge).—Burmeister and Wain, Vickers, Werkspoor, M.A.N., Krupp, North-Eastern.

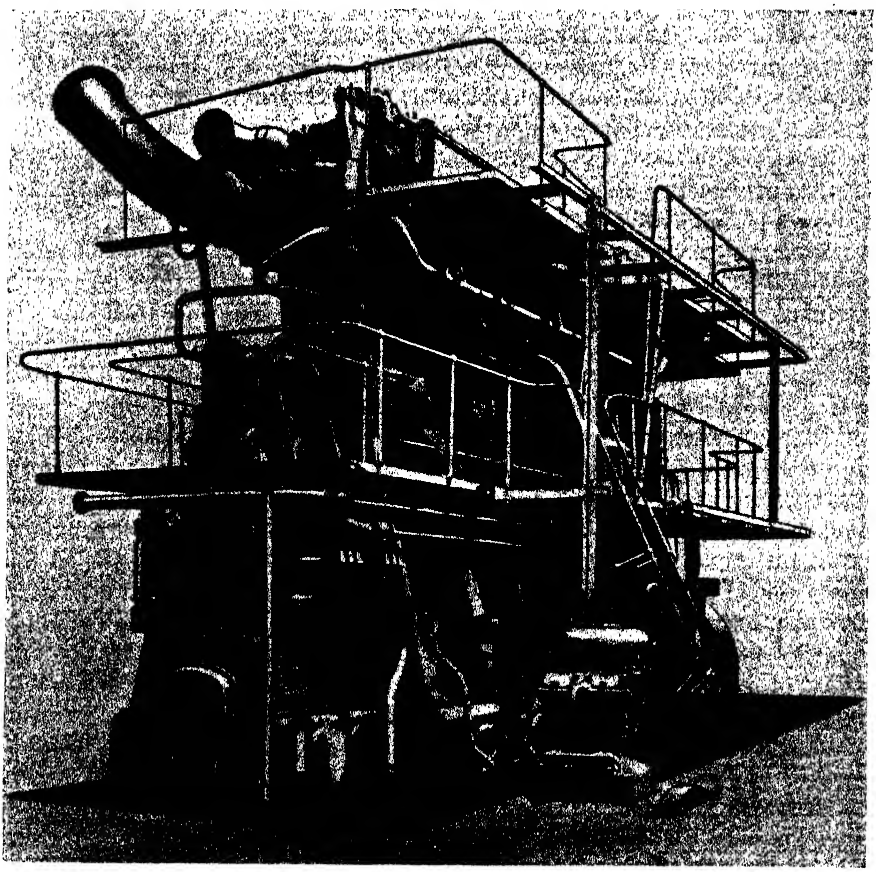


Fig. 63.—Beardmore-Tosi Marine-type Diesel Engine, 1250 b.h.p., 6 Cylinders, $24\frac{3}{8}$ in. diameter by $38\frac{3}{8}$ in. Stroke, 120 r.p.m.

2. *Single-acting two-stroke cycle.*—Doxford, Fullagar, Sulzer, Fiat, Polar, Krupp, M.A.N., B. & W., Scott, Stork Hesselman.

3. *Double-acting four-stroke cycle.*—No engines of this class have been built for some years.

4. *Double-acting two-stroke cycle.*—B. & W., M.A.N., Richardsons Westgarth, Stork Hesselman, Sulzer, Fiat.

Messrs Harland and Wolff, Ltd., Belfast, manufacture all types of engines of Burmeister and Wain design. (See C. C. Pounder, *Institute of Marine Engineers*, January, 1939.)

**Standard Four-cycle Engine.**—Fig. 49 indicates the standard unit of such an engine. In general, the frame is of cast iron, of ample scantlings to give rigidity. In some cases the piston load is taken off the cast iron by means of having steel bolts either from a cylinder head or the cylinder body to the bottom of the bedplate to carry all the tension stresses. The bottom end of the cylinder is isolated from the crank-chamber by a diaphragm provided with a gland for the piston-rod. This provision makes for easy access to the liner on the bottom side of the piston. The cylinder head, universally of cast iron, carries the valves required by the phases of the cycle; these are the inlet, the exhaust, the fuel, the starting-air, and relief valves. Fig. 63 represents such an engine.

Since 1930 the tendency has been towards the use of a fabricated construction for the engine framing, steel plates and rolled sections being built together by electric welding to form a rigid framework carrying the tension stresses due to the gas pressure in the cylinders. This construction eliminates the use of costly patterns for the cast-iron work and also gives a lighter engine.

During the same period progress has been made in the use of supercharging or augmented induction, and the following are some of the systems by which the air supply is provided:

1. By direct engine-driven compressor.
2. By direct engine-driven compressor, the underside of the working piston being used as the compressor (Werkspoor system).
3. By separate-driven compressor.
4. By exhaust gas driven turbo-compressor (Buchi system).

All these systems allow the output of the engine to be increased up to as high as 50 per cent over the unsupercharged engine of the same dimensions, without any undue increase in working cylinder temperatures and stresses.

In (4) the compressor is driven by the exhaust gases discharged from the engine, thus utilizing otherwise lost energy. The use of supercharging on the four-stroke cycle engine has to a great extent assisted in still keeping this type a competitor of the two-stroke cycle engine.

**Single-acting Four-stroke Marine Engines.**—As a typical example of a simple case of a modern moderate-powered Diesel-engined vessel, the Danish single-screw passenger and cargo ship *Tietgen* has been selected. This vessel is 270 ft. × 31 ft. × 15·3 ft. loaded draught, 2685 tons displacement, with a cargo capacity of 51,500 c. ft. and accommodation for 575 passengers. It is driven by an eight-cylindered four-stroke single-acting trunk-piston Diesel engine by Burmeister & Wain, and is equipped with airless injection; an external view of the engine is given in fig. 64.

The cylinders are 21·65 in. diameter, with a stroke of 39·37 in., and the engine has a normal full-power output of 2200 i.h.p. at 140 r.p.m., the corresponding brake horse-power being estimated as 1850, corresponding to a mechanical efficiency $\eta$ (see equation (2) of Chapter V) of roundly 84 per cent. An electric-driven centrifugal supercharger is fitted by aid of which the output can be increased to 2450 i.h.p. at 145 r.p.m. The inlet and exhaust valves are actuated by cam-shaft-operated levers on rocking shafts borne on the cylinder heads. A separate fuel-injection pump is fitted to each cylinder, and the fuel-injection valves are of the automatic pressure-opened spring-closing type.

The general lay-out of the installation is shown in fig. 65. As auxiliaries, three three-cylinder four-stroke single-acting trunk-piston airless-injection Burmeister & Wain Diesel engines are direct-coupled to three 110-volt 100-Kw. dynamos; each engine develops 150 b.h.p. at 270 r.p.m.

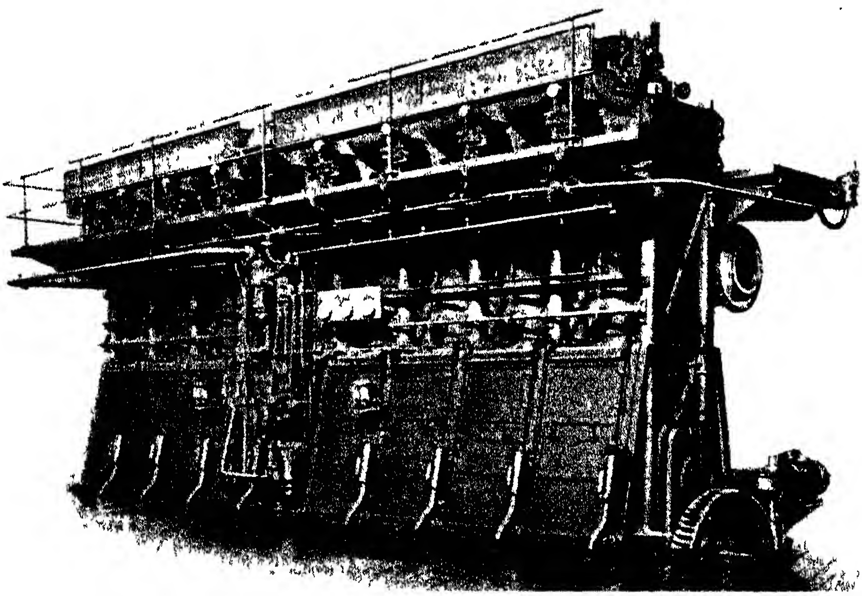


Fig 64 —Four-stroke Trunk-piston Engine of the *Tietgen* (Messrs Burmeister & Wain)

The main cooling water, forced lubrication, ballast, bilge, and sanitary pumps are all electrically driven. The fuel oil supply is also by an electrically-driven pump of gear-wheel type which draws oil from the fuel tanks to the settling tanks and daily-supply tank.

The cylinder covers are bolted together, forming a rigid top to the engine, and long through bolts are carried from the covers through the columns to the main bearings, thus transmitting, in tension, the forces acting on the cylinder covers direct to the main bearings, and relieving from stress the cast-iron cylinder jackets and A-framing of the engine. The cylinders are fitted with liners of special cast-iron mixture, machined all over, and formed with a heavy flange at the top by which they are attached to the under side of the cylinder covers by numerous steel studs and nuts. Although this ship is no longer in service, the particulars of the general lay-out are still of topical interest.

Fig. 66 shows a single-acting four-stroke cycle supercharged engine

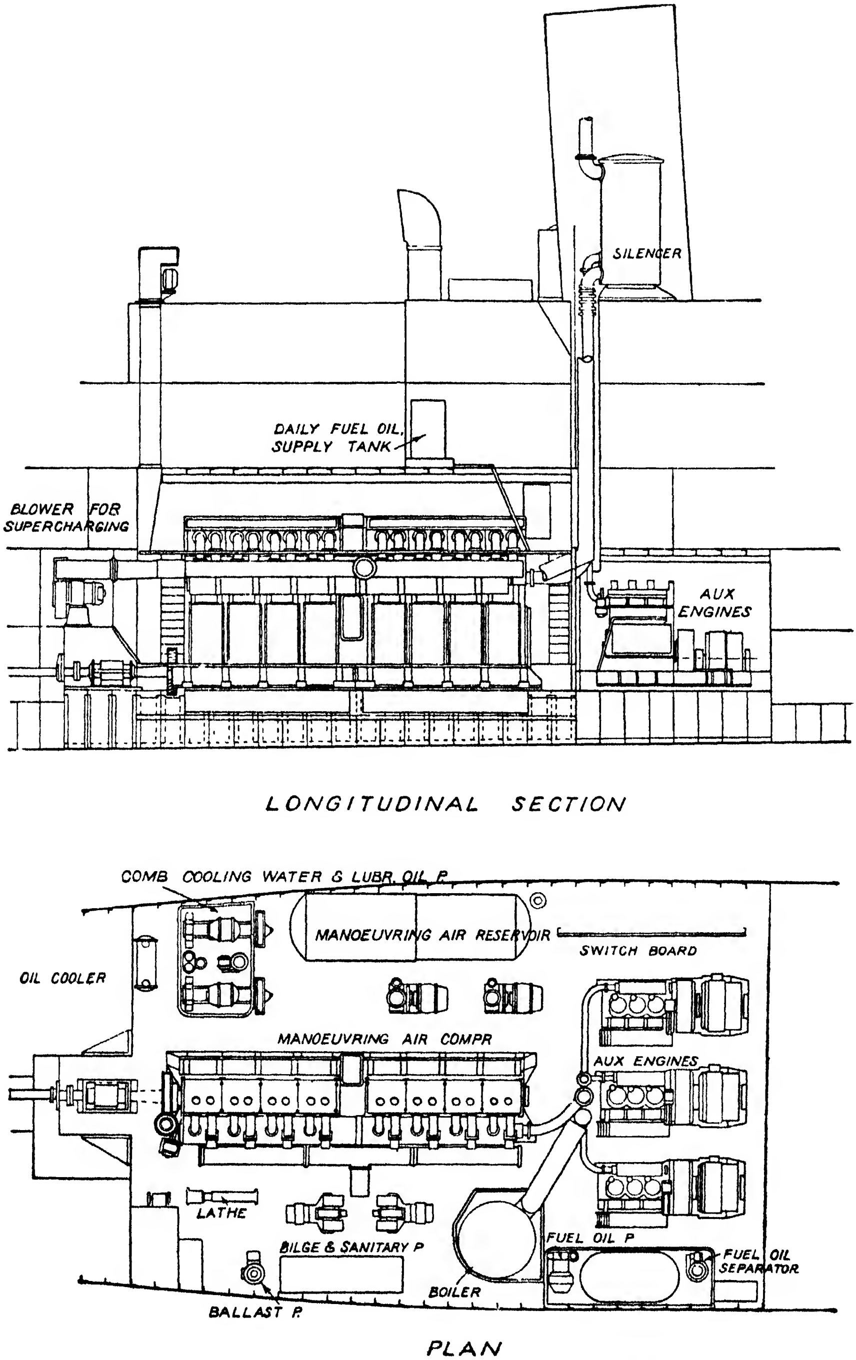



Fig. 65.—Engine-room Arrangement of the C.F. *Tietgen*

constructed by Messrs. Harland and Wolff, Ltd., in 6 and 8 cylinders, with under-piston supercharging. The scheme consists in enclosing the space between cylinder end and crank-case top, thus utilizing the undersides of the pistons as air pumps. The bore is 29·13 in. and the stroke is 59 in.

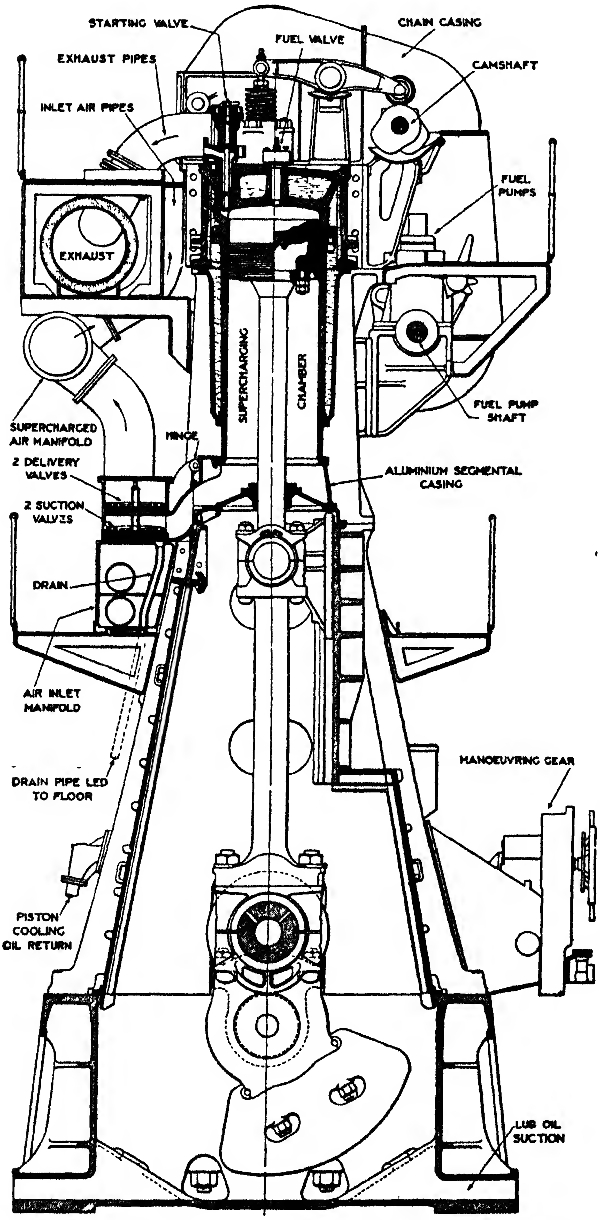
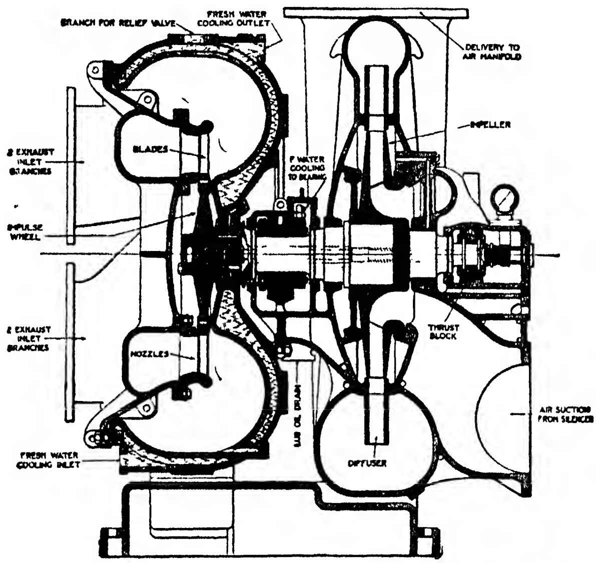



Fig. 66.—Section of Single-acting Four-stroke Supercharged Engine

Supercharging increases the cylinder pressure, and therefore the engine power, without increasing the temperature. A safe figure for continuous supercharge is 130 lb. sq. in. m.e.p. in comparison with 90 lb. sq. in. for an unsupercharged engine. The arrangements are clearly shown in the diagram. The pressure of the supercharged air is from $3\frac{1}{2}$ to $4\frac{3}{4}$ lb. sq. in. (Fig. 67 and the above particulars are reproduced from the paper by C. C. Pounder referred to on p. 224.)

An alternative method of supercharging is the Buchi system, in which use is made of the energy of the exhaust gases. Fig. 67 shows a section through this type of supercharger in which an exhaust gas turbine (on the left) drives a centrifugal blower (on the right). The turbo-blower illustrated is for a 10-cylinder engine manufactured by Messrs. Harland and Wolff, developing 6000 s.h.p. at 115 r.p.m., with a mean indicated pressure of 117 lb. sq. in. The exhaust gas quantity is 1410 lb. per min. at 17·5 lb. sq. in. absolute at turbine inlet and 14·8 lb. sq. in. absolute at turbine outlet. The air quantity is 19,000 c. ft. per min. at 18·5 lb. sq. in. absolute. The gas temperature at turbine inlet is 840° F. and the blower rotates at 4600 r.p.m.

Exhaust blowers adapt themselves automatically to changes of load. Since the weight of air increases with the load, the fuel consumption and exhaust temperature during overload conditions do not rise to the same extent as in unsupercharged engines. Since the maximum pressures are




*Inst. of Marine Engineering*

Fig. 67.—Exhaust Turbo-blower

higher the parts subjected to load must be proportionately stronger, but in spite of this the saving in weight per h.p. (C. C. Pounder, *I. Mar. E.*) is about 25 per cent.

A section through the Buchi Brown-Boveri exhaust-gas turbo-charger is given in fig. 68, the exhaust-gas turbine being shown on the right. The two air impellers, on the left, are mounted on the same spindle as the turbine rotor, the outer impeller taking in atmospheric air and discharging this under slight pressure to the second impeller, which further " boosts " the pressure and delivers into the trunk supplying the main engine cylinders.

In a paper presented to the World Power Conference on Fuel in Sept., 1928, after stating that it had recently been found that the exhaust gases can be made to drive a turbine with excellent results, M. Buchi proceeded:

" The explanation of the 50 per cent increase in output, made possible by supercharging at only 3·55 lb. per square inch (gauge), is as follows: The conditions of intake and exhaust are quite different in an exhaust-turbine supercharged engine from those in an ordinary engine, although this is not shown on the indicator diagram, because the scale of the spring is not suitable for the purpose. The timing of the inlet and exhaust valves is so chosen that the efficiency of admission and exhaust are as good as

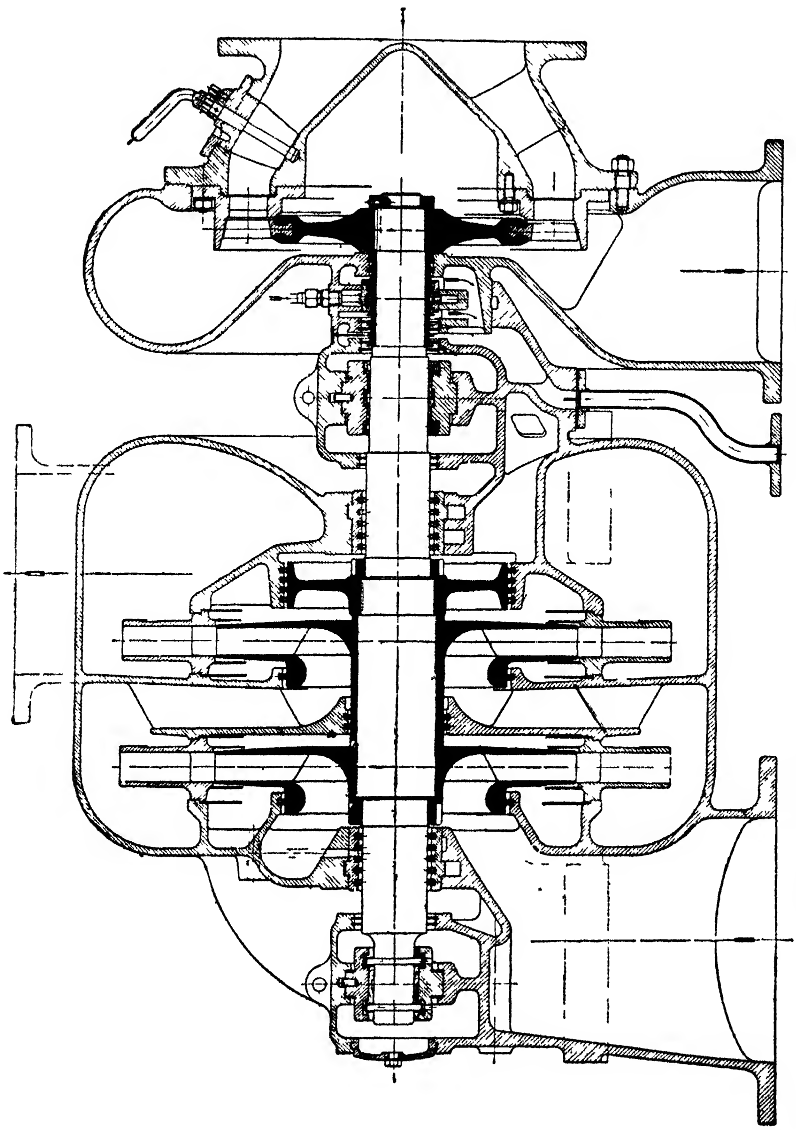


Fig 68 —Sectional View of the Exhaust-gas Turbo charger (as fitted in the *Raby Castle*)

possible; this is done by controlling the pressure between the engine and the turbine. As the lowest pressure is at the end of the exhaust stroke, it is obvious that the weight of the residual exhaust gases is then also at a minimum, which makes it possible to scavenge the combustion cylinders with charging air, the pressure of which is, at that moment, the pressure in front of the turbine. On scavenging, relatively cold compressed air enters through the inlet valves, sweeps the combustion space, and leaves through the exhaust valves, thereby clearing the cylinder and cooling the cylinder walls, piston head, and valves. It is owing to this cooling effect that it is possible to get such a low pressure at the beginning of the compression stroke. The scavenging is, therefore, of the greatest importance in determining the mean temperature of the cycle, which, in a Diesel

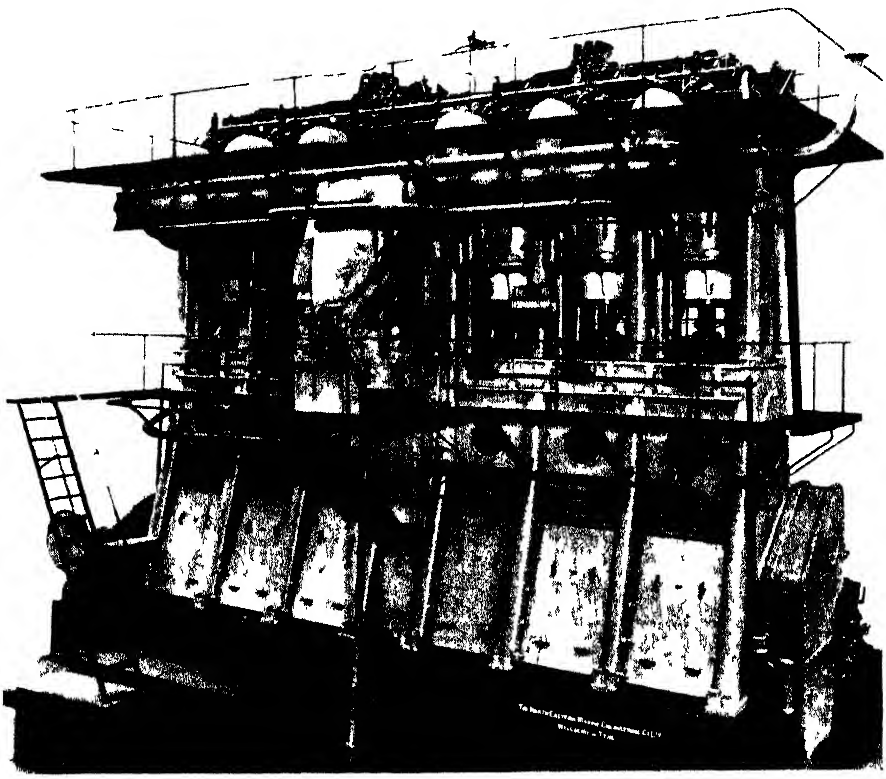


Fig 69 —North-Eastern 2600 b h p Buchi Pressure-charged Four-stroke Engine for M S *Hylton*

engine fitted with exhaust-turbine supercharging, is relatively low, and the increase in output and overload capacity. It has already been stated that the temperature at the beginning of the compression stroke in a supercharged engine is low; it therefore follows that, to obtain the same maximum pressure, a lower compression ratio must be used, with the result that the increase in temperature on compression is correspondingly lower. As the air fuel ratio is still the same, it is obvious that the increase in temperature or combustion must be the same, and that with the same excess of air the maximum temperature produced decreases with the compression ratio. The temperature reached during expansion and exhaust are the same for both types of engine, but at the commencement of the scavenging the temperature in the supercharged engine is lower than that generally found in Diesel engine practice. From these considerations it can be seen that the mean temperature of the exhaust-driven supercharged Diesel cycle must be somewhat *lower* than that of the ordinary four-cylinder engine."

Fig. 69 shows a 2600 b.h.p. Buchi pressure-charged four-stroke engine for the motor-ship *Hylton*, built by the North-Eastern Marine Engineering Co., Ltd. Built as a self-contained unit, driving its own cooling water, forced lubrication and bilge pumps, the main engine has 6 cylinders, diameter 620 mm. (24·4 in.) and stroke 1300 mm. (51·2 in.). As the engine

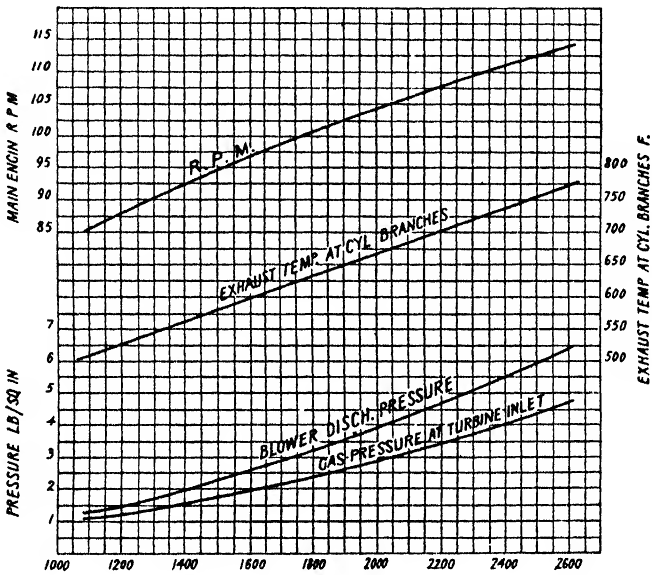



Fig. 70.—Curves obtained from Shop Tests

rating is 2150 b.h.p. at 106·5 r.p.m., the brake m.e.p. is 112 lb. sq. in. The maximum output is 2600 b.h.p. at 114 r.p.m. with a brake m.e.p. of 128 lb. sq. in.

In the *Hylton* the engine has the turbo-supercharger attached, and no means are provided for cutting it out of service. This is a result of considerable experience with these machines, which have long periods of trouble-free operation to their credit.

Curves obtained from shop tests of the engine are shown in fig. 70, and fig. 71 shows the variation of fuel consumption and mechanical efficiency with load. Curve A is for the engine supplying all water and oil required (including the dynamometer) and curve B is for an engine with independent auxiliaries.

**The Werkspoor Engine.**—In this well-known single-acting four-stroke marine Diesel of the Nederlandsche Fabriek, Amsterdam, the cylinder covers and liners are in earlier types cast in one, instead of, as in the usual practice, fitting the cylinder with a special liner; with this construction it is considered that better cooling is attainable, as the circulating water is then fully effective right up to the top of the liner, which is the point of highest temperature; this is difficult of attainment when a separate liner, with flange, is fitted. Fuel injection is by air blast, the fuel oil being supplied by a pump to a chamber located above the fuel valve, with the full pressure of the blast air acting upon its surface. Thence, through distributing boxes and piping, it is delivered to the several fuel inlet valves,

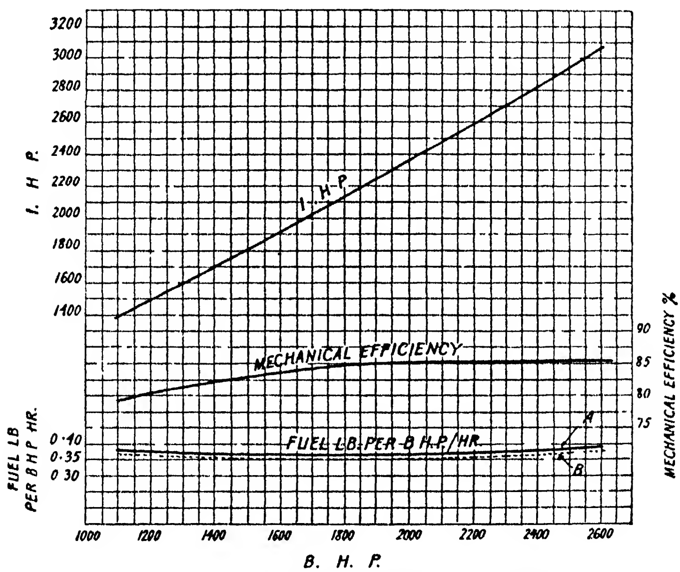



Fig. 71.—Fuel Consumption and Mechanical Efficiency

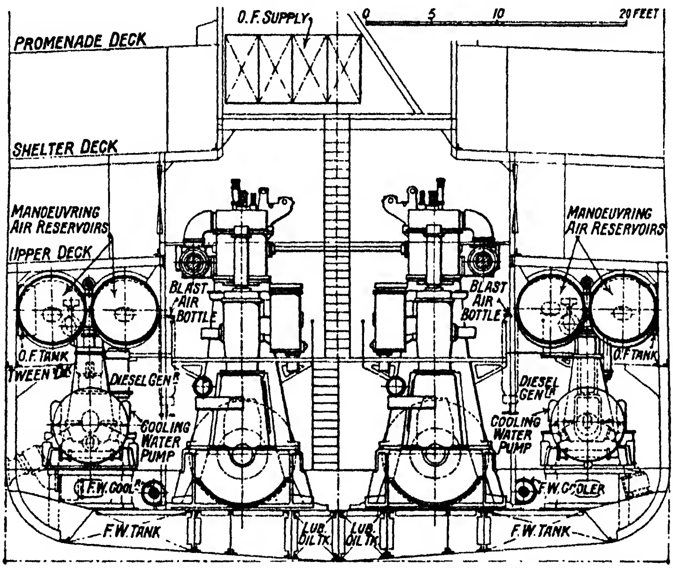
cocks in the piping enabling any individual fuel valve to be cut off when required.

The cam-shaft is situated above the cylinders, so that no push rods are needed for operating the valve levers.

Towards the end of 1928 the large twin-screw vessel *Ophir*, 370 ft. × 51½ ft. × 23 ft., 7200 tons gross, was completed by the Netherlands Shipbuilding Co. of Amsterdam. This vessel has a service speed of 15 knots, and is propelled by two eight-cylindered single-acting four-stroke Werkspoor engines of 27·6 in. bore and 47·2-in. stroke, running normally at 125 r.p.m.; at this speed each engine develops roundly 2500 b.h.p. A transverse section through the engine-room is given in fig. 72; the lubricating oil and fresh-water tanks in the double bottom will be noted, and also the large-capacity manœuvring compressed-air reservoirs.

Three three-cylindered four-stroke single-acting trunk-piston Diesel engines direct-coupled to 40-Kw. 220-volt generators supply current to the majority of the engine-room pumps and other auxiliaries, and also to the deck machinery. For maintaining the piston-cooling circulation two 10-h.p. motors are provided. The controls of the main engines are fitted at the forward ends.

In all the more recent engines of the Werkspoor type the cylinders are supported by cast-iron framework and the cylinder covers and liners are cast separately. Supercharged units are constructed with the lower ends of the cylinders closed in, the pistons acting as air pumps. A special development is that of a low-built crosshead type engine, having a very short piston rod and an extension of the liner which serves as the upper part of the guide. With this design the cylinder and liner are a single casting. (See Motor Ship Reference Book for 1938.)




By courtesy of *The Marine Engineer and Motorship Builder*

Fig. 72.—Section through Engine-room of the *Ophir*

**The Beardmore-Tosi Engine.**—A special form of the four-stroke Diesel is the Tosi design, formerly built in Great Britain by Messrs. Beardmore. The characteristic feature of this engine is the utilization of one valve only both for inlet and exhaust. The mode of working is illustrated diagrammatically in fig. 73. A is a poppet valve of the regular inlet type actuated, as usual, by a lever from the cam-shaft. A special rocking valve B, termed the "director" valve, driven by an eccentric from the cam-shaft, alternately places the passage through A—and thus the combustion chamber of the cylinder—in communication with the air through C, and with the exhaust manifold by way of D. Thus B is in its upper position when the valve A is opened for the suction stroke, and in its lower position when A is open for the exhaust stroke. Thus one poppet valve suffices both for suction and exhaust, and the head of the valve is alternately subjected to a stream of fresh air and of exhaust gases, and accordingly keeps much cooler than the exhaust valve in the normal Diesel engine, resulting in absence of " burning " and prolonged working life.

The Beardmore-Tosi engine is built both of the single-acting and double-acting four-stroke crosshead enclosed types in sizes up to 2400 b.h.p., cylinders ranging in diameter from 7·1 in. to 24·8 in., and the brake horse-power per cylinder from 15 to 400; they have been fitted to a number of submarines.

Double-acting Beardmore-Tosi engines were installed in the converted steamer *Wulsty Castle*; these ran normally at 250 r.p.m., and drove the propeller shaft at the reduced speed of 80 r.p.m. through Vulcan hydro-mechanical gearing.

The cam-shaft is fitted with the usual side-by-side cams for ahead and astern running, by which the " combined " valves A, the air-starting valves, and the fuel valves are actuated; the director valves do not require reversal. The cam-shaft has no axial movement, but the levers operating the valves have forked ends carrying pins along which the rollers can slide. The levers being lifted clear of the cams, the rollers are moved along their pins until over the astern cams, when the levers are lowered and reversal is effected.

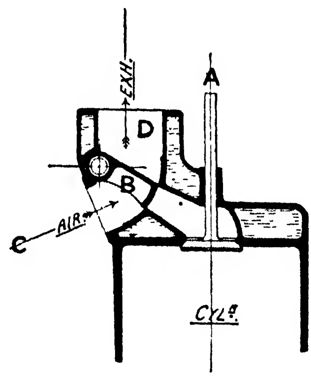



Fig. 73

The compression pressure used is about 485 lb. per square inch. Sea water is used for cooling, the piston-cooling pressure being 25 lb. per square inch, and cylinder jackets 12 lb. per square inch. The lubricating-oil pressure is 10 lb. per square inch. The fuel consumption at normal full load averages 0·42 lb. per b.h.p. hour.

Engines of this type are not now being constructed in Great Britain.

**Diesel-electric Drive.**—In 1929 Messrs. Yarrow built for the Argentine Government a Diesel-engined double-ended ferry-boat of 145 ft. in length × 40 ft. beam. This vessel has two propellers at each end, each of the four being driven at 350 r.p.m. by a 200 h.p. electric motor; the two motors of each pair of propellers can be controlled jointly or independently, but only one pair can be used at a time. Power is obtained from two generating sets each comprising a 330-b.h.p. six-cylindered single-acting four-stroke trunk-piston M.A.N. Diesel engine running at 350 r.p.m., direct-coupled to a 170-Kw. direct-current propulsion generator, and also to a 35-Kw. direct-current auxiliary generator, which supplies current to the vessel's auxiliary services.

**Sulzer Two-stroke Marine Engines.**—The experience of this firm for many years with both four-stroke and two-stroke designs led them to adopt finally the two-stroke cycle engine for all marine services, and their standard designs include 4, 6 and 8-cylindered engines developing from 300 to 7200 b.h.p., the highest continuous rating being 900 b.h.p. per cylinder

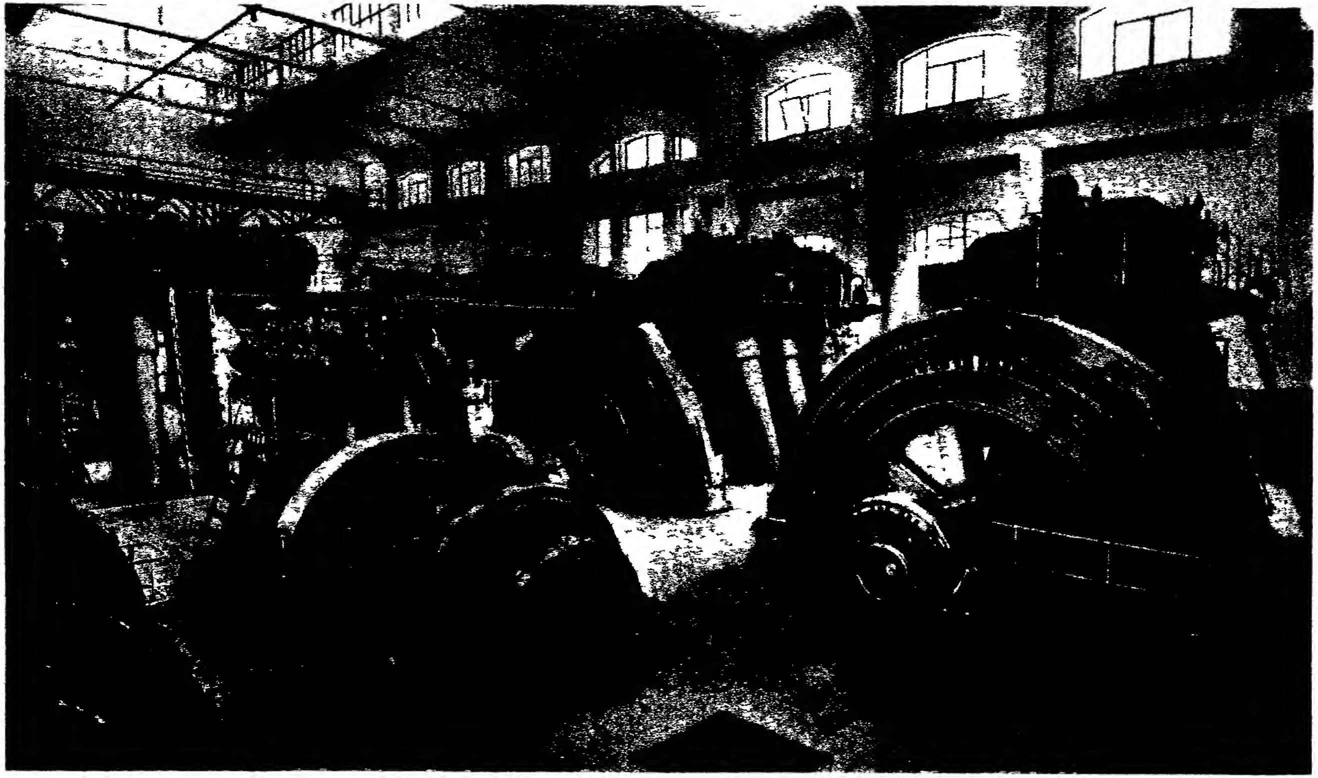


Fig. 74.—Central Power Station at Rome

Six Tosi-Diesel engines totalling 7500 h.p., including two engines of the two-cycle type developing each 2400 b h p. in six cylinders at 135 r p m.

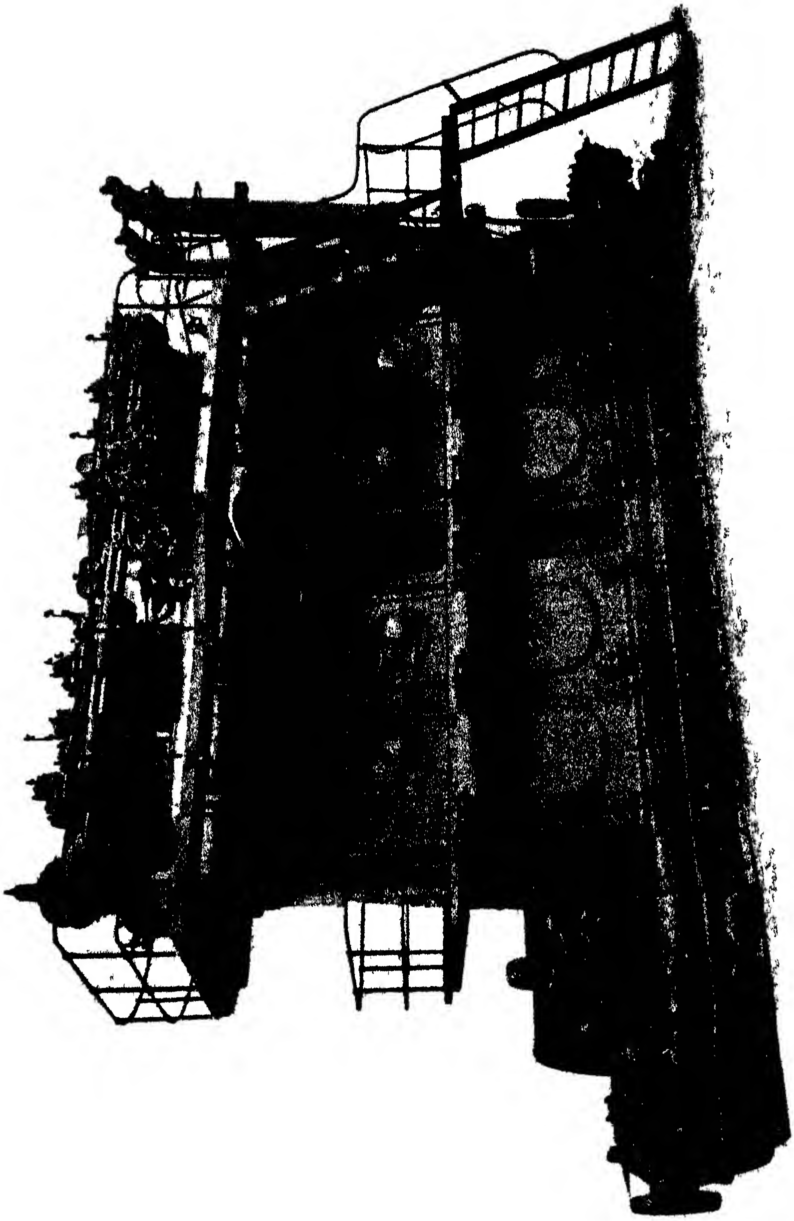


Fig 75 —Sulzer Marine type Engine 4 Cylinders, 1500 b h p.

(fig. 75). Large engines of the same type are also manufactured for land installations.

Two rows of ports are provided for the inlet of the scavenging-air. The scavenging-air which enters the top row of ports was formerly controlled by a rotary valve in the scavenging-air reservoir (now superseded, see fig. 76). The object of this Sulzer system is to effect a supercharge in the cylinder. When the piston descends, the rotary valve is in the closed position; the exhaust gases commence to pass out through the exhaust ports. When the cylinder is partially exhausted, the main scavenging-air enters through the bottom scavenging-ports, and on the upstroke of the piston the rotary valve opens, so that scavenging-air is entering the cylinder when the exhaust ports are closing, thus clearing out the exhaust gases from the cylinder more completely and filling it with fresh air for the ensuing cycle at a slight pressure, 1·5 to 3 lb. per square inch. Futhermore, this arrangement achieves the desirable object of keeping cooler the inner surfaces of those parts exposed to combustion temperature. In this way the only valves required in the cylinder head are the fuel, starting-air, and relief valves. The head, therefore, can be of relatively simple design, as will be apparent from the illustration. The frame is similar to the standard four-stroke cycle engine, except that to reduce headroom the piston, which has a shroud to cover the exhaust and scavenging-ports when at the top dead centre, is allowed to come down into the crank-chamber of the engine, so that four guide-plates per line are incorporated in the framing.

To indicate the advance made during recent years with this type, mention may be made of the Sulzer engines constructed in 1934 by the Société Anonyme John Cockerill, Belgium, for the motor-vessel *Prince Baudouin*. This ship is running on the cross-channel service between Ostend and Dover, and is designed for a speed of 24 knots. The total horse-power of the two engines installed is 17,000 b.h.p., developed when operating at 268 revolutions per minute. The machinery is of the airless injection single-acting type, each engine having twelve cylinders, 580 mm. diameter, with a stroke of 840 mm. The scavenging-air is provided by electrically driven blowers.

**The "Poelau" Class Liners.**—Early in 1928 was completed the first of four 7040 b.h.p. Sulzer engines for the single-screw *Poelau* class of 14½-knot cargo liners for the Netherlands Shipping Company. These vessels are 490 ft. in length, 61 ft. beam, and 17,450 tons load displacement. The fuel oil is stored in large side tanks in the engine-room, and also in the double-bottom tanks beneath. The daily consumption of oil is 30–35 tons, and the storage capacity enables the vessels to make the round trip from the Dutch East Indies to Europe and back without refuelling.

The Sulzer two-stroke engines for these four vessels were, at the time, the largest single-acting units that had been installed. Each comprised eight cylinders of 32·3-in. bore × 56·7-in. stroke, with an aggregate output of 7040 b.h.p. at 100 r.p.m. The fuel consumption was only about 0·40 lb. per b.h.p. hour, including the power necessary to drive the blower, cooling pumps, and lubricating-oil pumps.

A transverse section through one "line" of the engine is given in fig. 76. The two belts of upward-inclined air inlet ports will be noted, the upper belt communicating with the scavenging air trunk through a

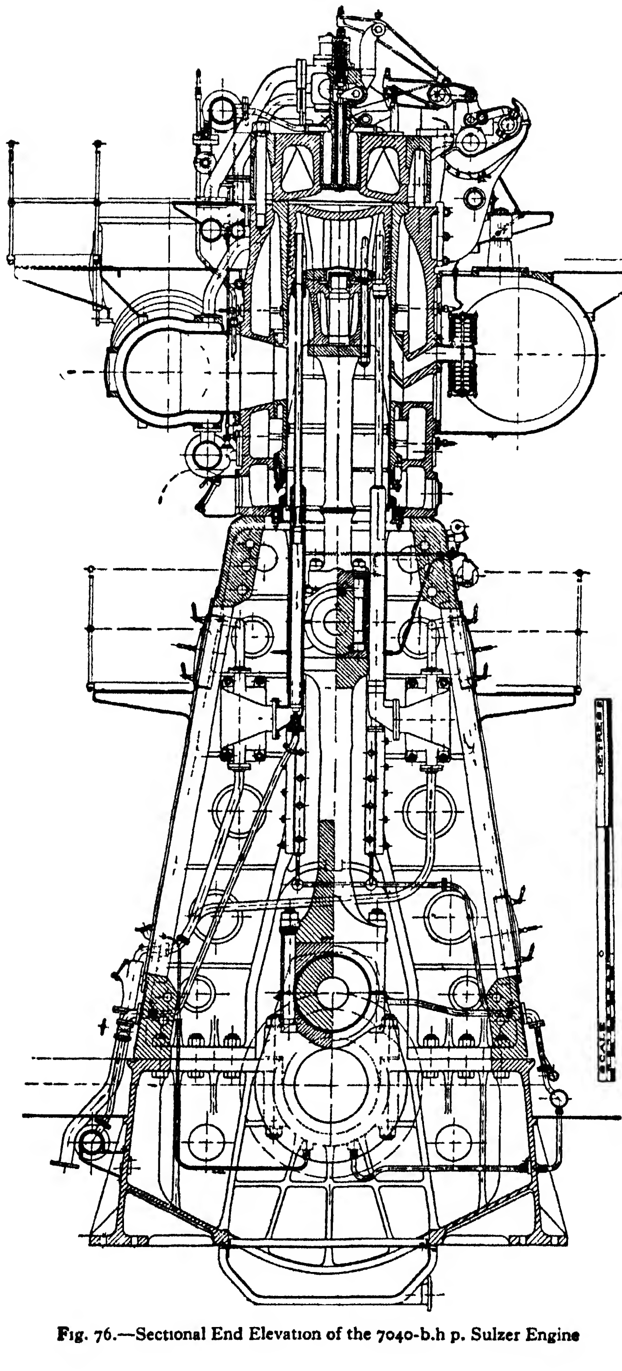


**Fig. 76.—Sectional End Elevation of the 7040-b.h p. Sulzer Engine**

nest of automatic valves which open to the cylinder as soon as the pressure therein falls below that in the trunk; the upward inclination of the ports improves the scavenging of the burnt gases by the inrushing fresh air.

The engine is of the crosshead type, and the piston is sea-water cooled; a telescopic pipe attached to the piston head is the only sliding pipe; the water is delivered to the piston at 25–30 lb. per square inch. Sea water is also used for cooling the jackets, cylinder cover, and exhaust manifold.

An external view of the engine is shown in fig. 77; the great simplicity of its appearance is noteworthy. All the controls will be observed below, and at the forward end.

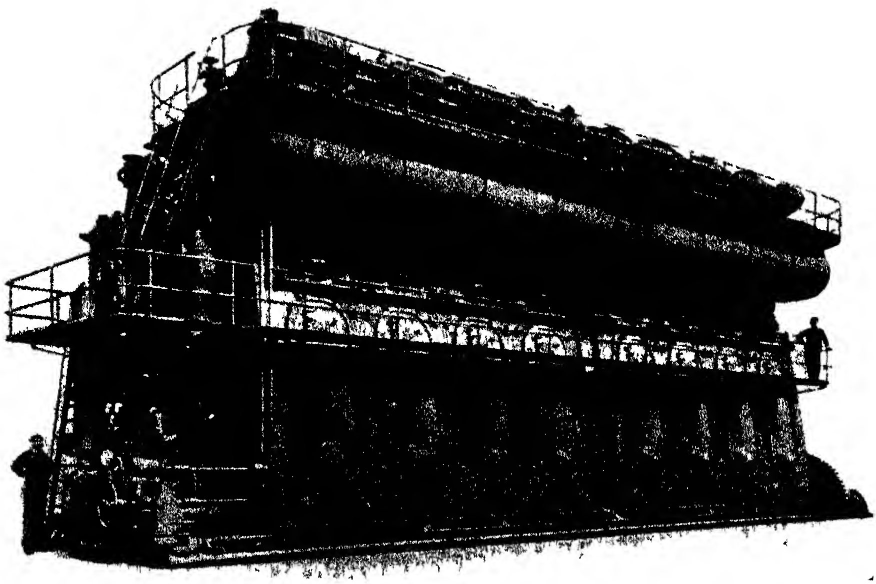


Fig. 77.—7040-b.h.p Sulzer Engine

The overhead cam-shaft carries a pair of cams, for ahead and astern running respectively, for each starting air inlet valve and each fuel valve, and reversing is effected by bringing a second set of rollers into operation by its appropriate cam by means of a hand-wheel. Starting is, of course. by compressed air. The scavenging air is supplied by a Brown-Boveri blower driven by an electric motor at each end, at 2450 r.p.m.; the capacity of the blower is 560 c. ft. of free air per second.

An airless-injection four-cylinder auxiliary engine, direct-coupled to an air compressor with four three-stage pumps of capacity 420 c. ft. of free air per minute, charges a battery of 18 high-pressure starting-air bottles, each of 28 c. ft. capacity, the maximum air pressure being 1000 lb. per square inch. There are, further, two intermediate-pressure starting-air reservoirs, each of 390 c. ft. capacity, at a pressure of 425 lb. per square inch. The exhaust gases from the main engine are taken direct to an exhaust-heated boiler. There is, in addition, an oil-fired boiler for general heating purposes.

*Auxiliary Engines.*—In addition to the engine driving the air compressor, there are three auxiliary four-cylinder single-acting two-stroke crosshead-type Sulzer Diesel engines of 13·4-in. bore × 21·3-in. stroke, each developing 360 b.h.p. at 180 r.p.m., direct-coupled to generators supplying current to the motors of the auxiliary machinery. These are of generally similar design to the main engines, with enclosed crank-chambers and a lantern piece between the cylinders and crank-case enabling the piston rods to be viewed while running; there is a stuffing-box above the cover over the crank-case. The scavenging pump and air compressor are located together at one end of these engines. Sea water is utilized for cooling both pistons and cylinders, as in the main engine.

There are two salt-water cooling pumps each with a capacity of 400 tons per hour at a pressure of about 40 lb. per square inch. The lubricating oil pumps have a capacity of 55 tons per hour, also of 40 lb. pressure. The crosshead lubricating oil pumps, which are duplicated, have a capacity of 8 tons per hour at a pressure of 250 lb. per square inch.

At the full load of 7040 b.h.p. at 100 r.p.m. the mean indicated pressure is 91 lb. per square inch. The mechanical efficiency, excluding the blower, is about 81·5 per cent.

Messrs. Sulzer build also quick-running light-type Diesel engines for use in submarines, developing up to 4000 b.h.p. in eight cylinders, at 350 r.p.m. In these engines the method of scavenging, and the design generally, follow that of the larger slower-running type already described.

For the surface propulsion of submarines the high-speed type Diesel is now generally employed.

**The M.S. " Rangitiki ".**—Early in 1929 the New Zealand Shipping Company's first motor-ship *Rangitiki* commenced her regular service. This is a twin-screw vessel, 533 ft. in length × 73 ft. beam, with a gross tonnage of 16,755, and is propelled by two five-cylindered single-acting two-stroke Brown-Sulzer Diesel engines, each capable of developing 5000 b.h.p. at 90 r.p.m. The normal aggregate rating is 9300 b.h.p. at 88 r.p.m. An output of 1000 b.h.p. per single-acting two-stroke cylinder was the highest that had been installed, of the Sulzer type, in a ship at that date. The cylinders have a bore of 35·4 in., and the stroke is 63 in.; each engine drives its own reciprocating scavenge pump and three-stage blast injection air compressor, located at the forward end and driven from an extension of the crank-shaft.

Two 780-b.h.p. six-cylinder two-stroke air-blast injection crosshead-type Diesel auxiliary engines, each driving its own double-acting scavenge pump, three-stage air compressor, and lubricating pumps, drive 500-Kw. generators at 170 r.p.m. A further two six-cylinder two-stroke 225-b.h.p. trunk-piston airless injection Diesel engines, also driving their own scavenge pumps, drive 150-Kw. generators at 320 r.p.m. and supply current to the motors of the auxiliary machinery, and for the general service of the vessel.

**The "Polar" Marine Engine.**—In the single-acting two-stroke engine of the Atlas Diesel Co. of Stockholm, known as the "Polar" engine, the scavenging air-pumps are utilized for starting and reversing also, and in this way no air starting valves are required on the engine cylinders, leaving the fuel inlet valve as the only one requiring operation on the cylinder cover, with resulting simplification of controlling gear.

The four-cylindered "Polar" engine has two double-acting air-pumps directly driven from the main shaft by two cranks at 90°. In normal running these pumps deliver into the scavenging air receiver at a pressure of about 2 lb. per square inch above atmosphere. When starting or reversing, however, the pumps are operated as compressed-air motors, and drive the crankshaft for a few revolutions, after which the engine takes up its normal working cycle. Immediately the engine starts, the fuel oil pumps deliver oil to the fuel inlet valves; thus the engine when started receives two impulses, viz. one from the scavenging pumps operating as motors, and one from the normal explosion of the charge; acceleration at starting is thus hastened.

The "Polar" marine Diesel engine is built in Great Britain, under licence, by Messrs. Swan, Hunter, and Wigham Richardson, Ltd.

A number of ships built recently are fitted with geared "Polar" engines driving a single propeller shaft through magnetic couplings. In one installation (4400 b.h.p.) four engines are geared to a shaft running at 90 r.p.m. All "Polar" engines are of the trunk piston type, and many yachts and large coasting vessels have been fitted with these.

**Fiat Marine Engines.**—Fig. 78 is a sectional end elevation of one of the four engines installed in the *Oceania*, designed for 6000 b.h.p. at 140 r.p.m. Each engine has 8 cylinders, diameter 750 mm. (29·5 in.), stroke 1060 mm. (41·8 in.). The Fiat Company is one of the very few which still adhere to blast air injection for the fuel. The engine is of the two-stroke single-acting type.

The disposition of the scavenging air and exhaust ports is shown in the sectional view, the scavenging trunk having automatically operated plate valves for admission of air. The scavenging pump is forward and comprises two double-acting pump units, one above the other.

The cylinder assembly is one of the most interesting features of the engine. There is an upper liner of cast steel, to which is bolted a lower portion of cast iron. In the upper liner is inserted a renewable liner of cast iron, so that a replacement merely involves the provision of a comparatively inexpensive and simple casting where the greatest amount of cylinder wear occurs. The cylinder cover is of cast steel and is inserted some little distance into the main upper liner, the top of the latter being thus protected from expansion stresses due to high temperatures.

Steel bolts connect the cylinders to the bedplate and relieve the engine columns from tensile loads.

**Harland and Wolff (B. & W.) Single-acting Two-stroke Engine.**—Fig. 79 is a cross-section showing the principal features of construction

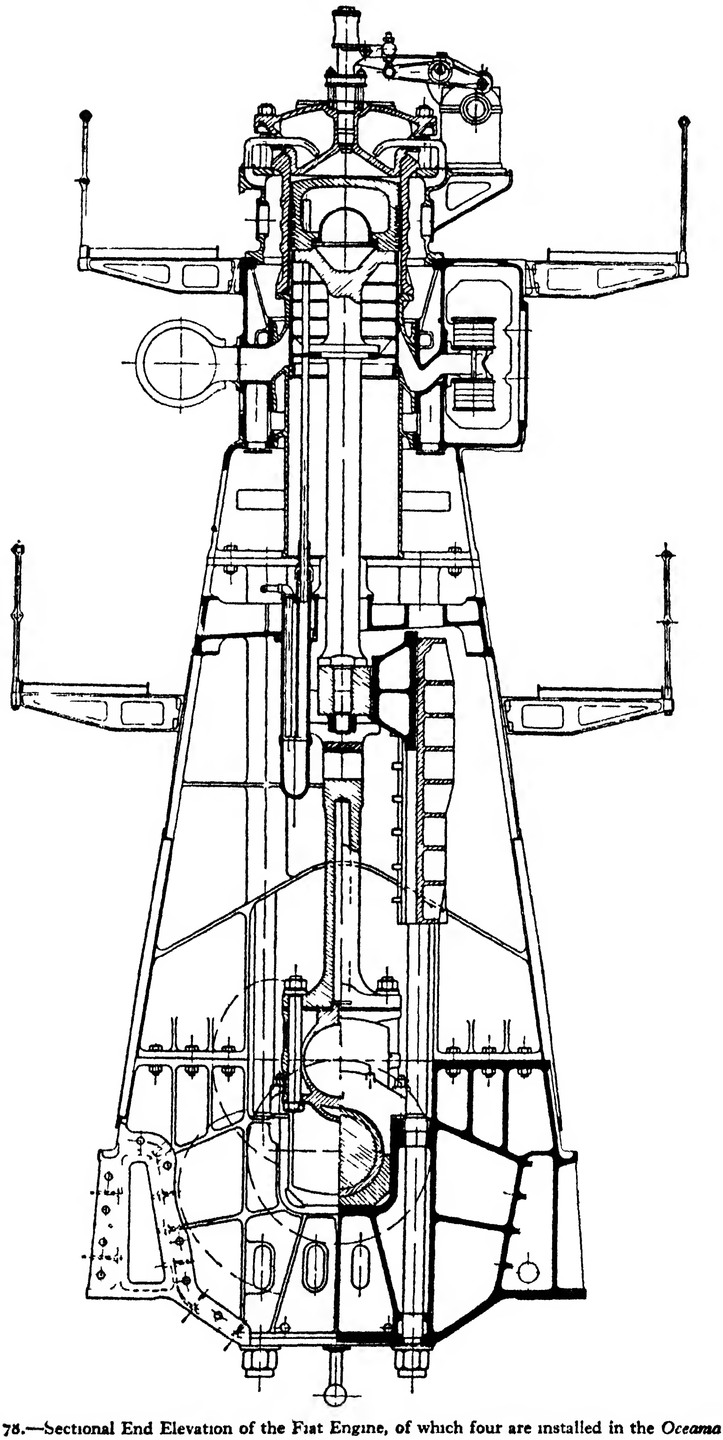


**Fig. 78.—Sectional End Elevation of the Fiat Engine, of which four are installed in the *Oceana***

of this type of engine. (See C. C. Pounder, *Proc. I. Mar. E.*) The bore is 500 mm. (19·69 in.) and the stroke is 900 mm. (35·43 in.), and the engine will develop up to 400 b.h.p. per cylinder.

Scavenge air is admitted at the bottom of the stroke by the piston uncovering ports in the liner, and the exhaust gases are expelled through a

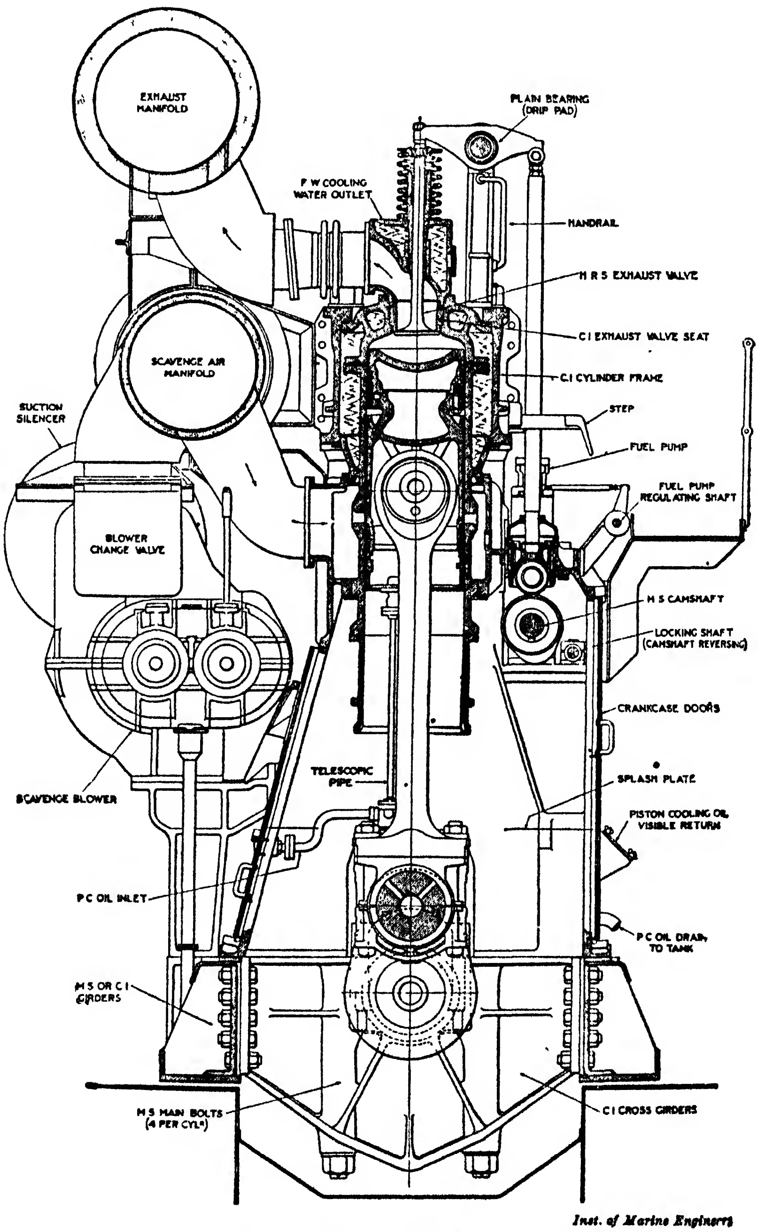



Fig. 79.—Section of Single-acting Two-cycle Engine

poppet valve centrally arranged in the cylinder cover. The approximate periods are: exhaust opens 70° before bottom centre and closes 45° after; scavenge opens 32° before and closes 32° after.

The cam-shaft, which operates the fuel and exhaust valves, rotates at crank-shaft speed, and is driven by spurwheels from the crank-shaft. There is one fuel pump per cylinder, arranged at the crank-case top. Two fuel valves per cylinder are fitted. The piston is a steel crown bolted or screwed to a cast-iron skirt.

Fig. 80 shows the fuel consumption curves of a 12-cylinder engine, 620 mm. bore and 1150 mm. stroke. The maximum power is 8100 b.h.p. at 135 r.p.m. with 114 lb. sq. in. mean indicated pressure.

The engine is of the trunk-piston type, and the piston is cooled by oil circulation, but a piston rod and crosshead can be fitted when desired.

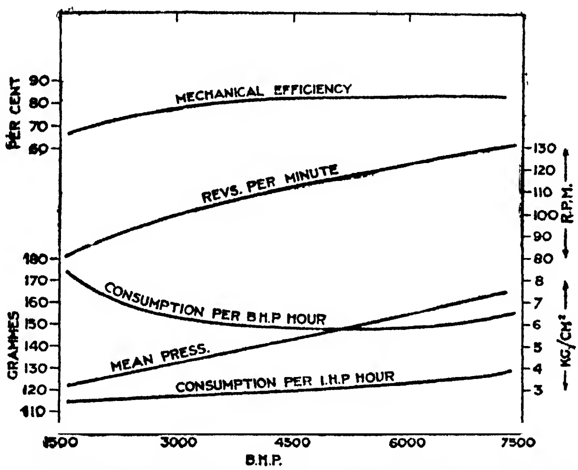



Fig. 80.—Test-bed Results for 7000-b.h.p. Double-acting Two-cycle Engine

(*Note.*—To convert grammes to pounds multiply by 2·2 and divide by 1000; to convert kg. per sq. cm. to lb. sq. in. multiply by 100 and divide by 7.)

**The Doxford Engine.**—The Doxford opposed-piston single-acting two-stroke marine Diesel, with open-ended cylinders, exhibits a departure from the conventional single-piston cylinder open at one end only. One of the earliest double-piston two-stroke gas-engines was that of Atkinson in 1885. In 1899 a 600-h.p. double-piston gas-engine designed by Dr. Oechelhauser was at work at the Hoerde Iron Works, and the type proved very successful in land installations. Oechelhauser gas-engines were built in Great Britain by Messrs. Beardmore of Glasgow, their largest design including a cylinder of 48 in. diameter, the stroke being 60 in. and output 2500 b.h.p. at 80 r.p.m. The double-piston engine was also successfully employed for some years, of inverted-vertical type operating on the four-stroke cycle with petrol as fuel, in the Gobron-Brillié motor-car.

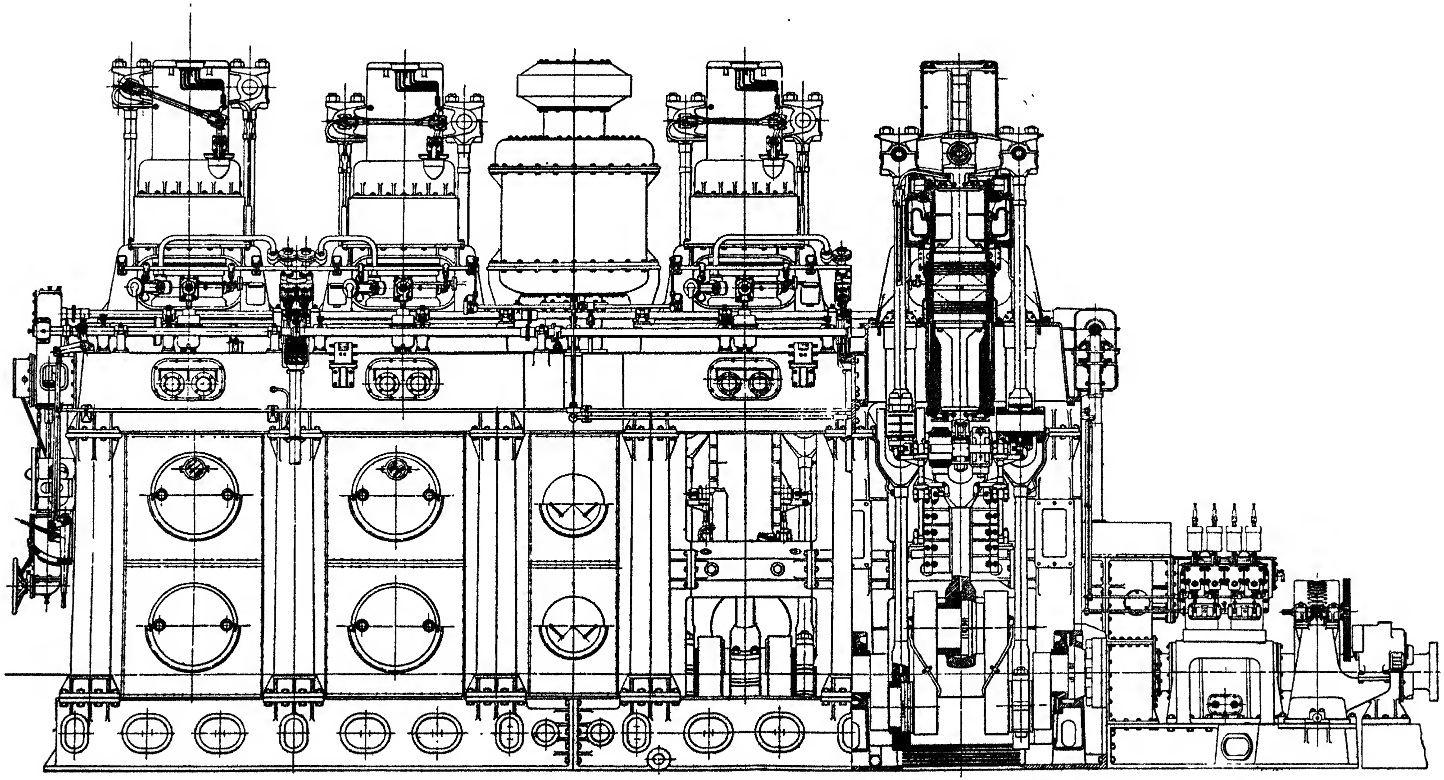
The opposed-piston type was adopted by Messrs. Doxford for marine propulsion, and by 1912 they had produced designs of the now well-known Doxford engine working on the Diesel cycle. Progress was checked by the war of 1914–18, but work was resumed in 1919, when a 3000-h.p. engine gave very encouraging results on trial. This engine was installed in 1921 in the motor-ship *Yngaren*, of the Transatlantic Co. of Gothenburg; by March, 1928, this vessel had completed roundly 250,000 miles in continuous service in the South Seas. Doxford engines have since been installed in many passenger ships, cargo boats, tankers, &c., and include cases developing as much as 5500 h.p. on a single shaft.

The advantages of the opposed-piston system are that no cylinder head is required; the liner is merely a simple tube; scavenging air enters at one end (the bottom end in this case) and sweeps out the exhaust gases through the ports in the top end. A complete end-to-end scavenging is thereby effected without the necessity for supercharging. In the Doxford engine the top piston is connected by means of a rocking beam and two side rods to the two side cranks, and the bottom piston by means of the usual piston and connecting-rod operates on the middle crank. The turning moment achieved, it will be clear, is excellent. Moreover, with a moderate piston speed, the effect of the two approaching pistons, between which combustion takes place to drive them apart, is extremely rapid, with consequently high thermal efficiency and a reduction to a minimum of the possible piston-ring leakages. The balance of the engine is good, in that the thrust, due to the piston attached to the centre connecting-rod, is balanced by the equal and opposite pull of the two side connecting-rods.

This type of engine has advanced considerably in design since 1929, and from 1933 has generally been made with a framework of fabricated construction built up of steel plates and sections electrically welded together.

**The Motor-ship "Bermuda".**—A typical installation is that for the Furness Withy quadruple-screw passenger liner *Bermuda*, 547 ft. × 73¾ ft. × 45 ft. moulded depth, 19,000 tons gross, with 144,000 c. ft. of cargo space and accommodation for 691 passengers; service speed 17–18 knots. This large vessel is driven by four Doxford balanced four-cylindered opposed-piston two-stroke Diesel engines, the pistons having in this case unequal strokes, that of the upper set being 30 in. and of the lower 41 in. The bore is 23·6 in., and normal full speed 110 r.p.m., rising to a maximum of 120 r.p.m. The aggregate power of the four engines at 110 r.p.m. is 11,200 b.h.p., rising to a maximum of 13,600 b.h.p. at 120 r.p.m. An external view of one of the four engines appears in fig. 81. The scavenging pump is situated in the middle of the engine and is directly driven by the crankshaft; it has a bore of 62·2 in. and a stroke of 34·6 in., and is shown in section in fig. 82. The scavenging piston is of phosphor bronze and is fitted with ring valves; the air enters the pump direct to the piston, through a silencer at the top, and is delivered by the pump to the entablature of the engine, which forms an air reservoir, whence it passes to the lower (inlet or "scavenging") belt of ports in the cylinders.

Messrs. Doxford were among the first successfully to employ "airless"



FRONT SECTION ELEVATION OF ONE OF THE ENGINES OF THE *BERMUDA*

[*Facing p.* 246, *I.-C. E.*

injection of the oil charge, and the four fuel pumps are fitted at the after end of the engine, just forward of the flywheel (shown in the folding plate). These deliver the oil to the two spraying injection valves, placed dia-

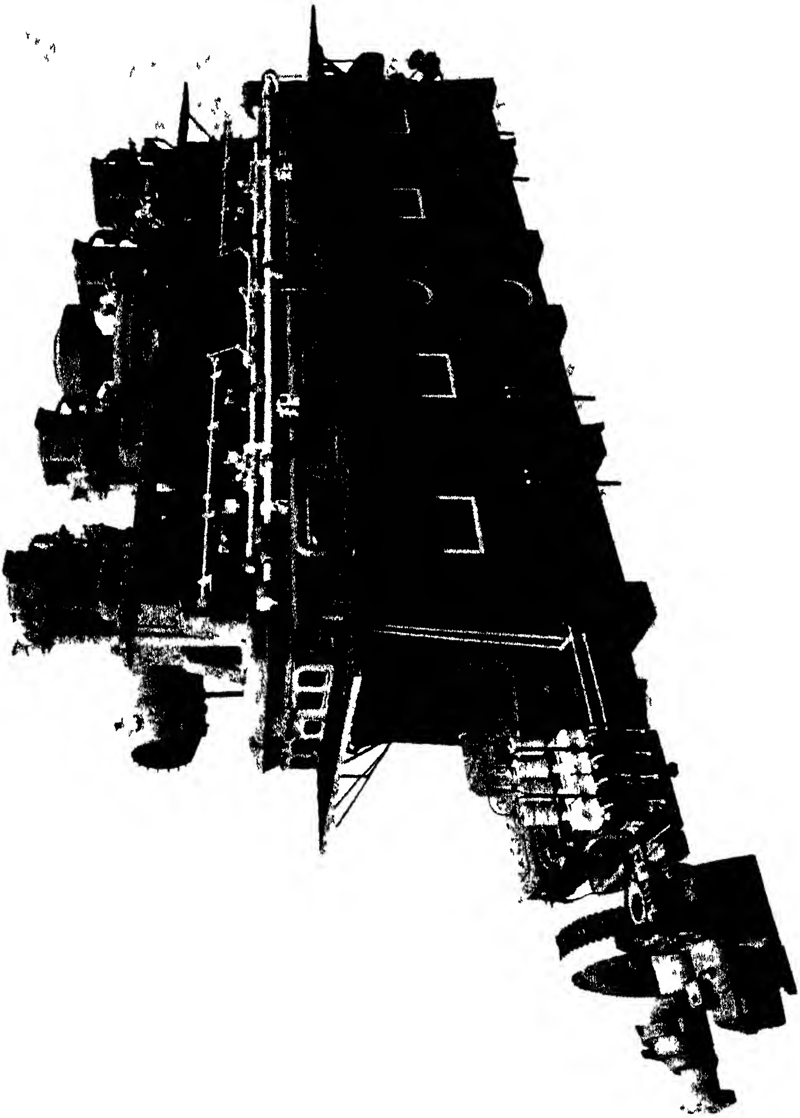


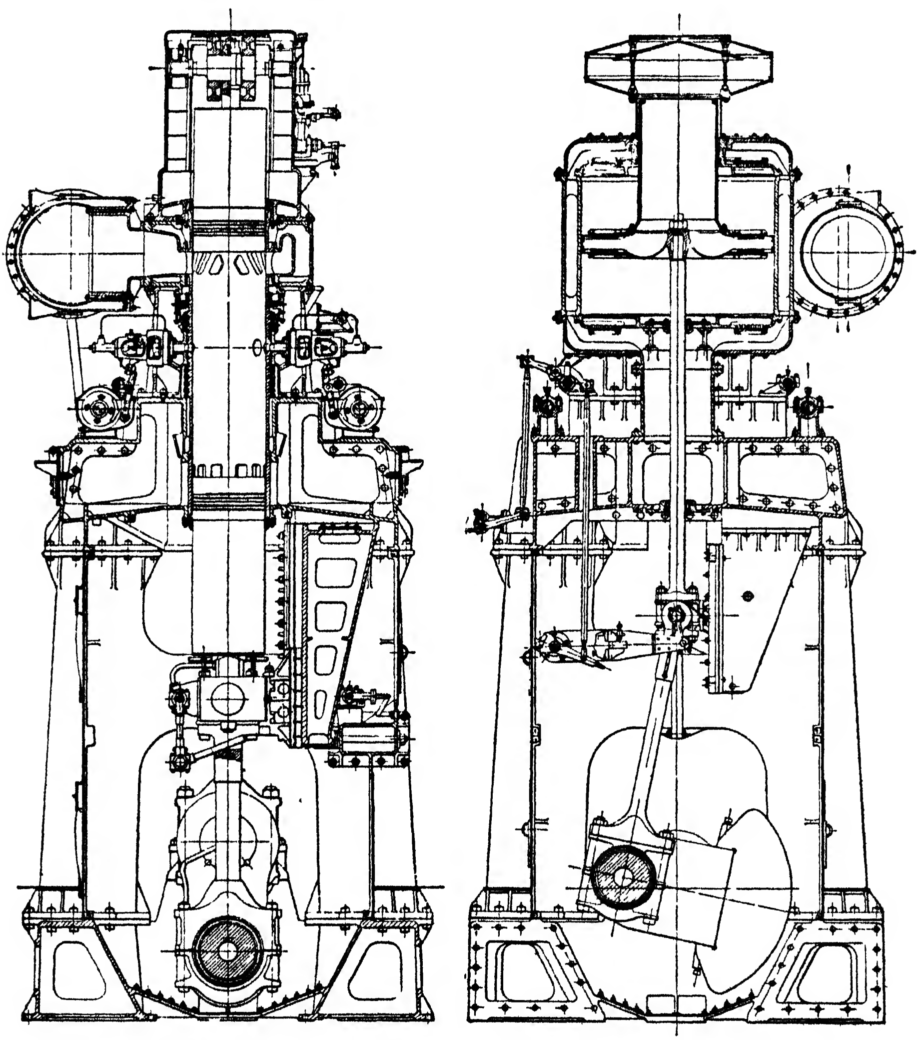
Fig 81 —One of the four Doxford Engines installed in the 19 000 ton liner *Bermuda*

metrically opposite one another in each cylinder, at a pressure of 6000–8000 lb. per square inch. The scavenging pump and the fuel-injection pumps are the only auxiliaries directly driven by the main engines.

The engines are completely enclosed, large aluminium doors, about

36 in. diameter, easily removable, giving ready access to the crank-chambers for inspection of crossheads and main bearings. The upper doors are fitted with 8-in. glass port lights through which the piston skirts, &c., can be viewed during running, the crank-chamber interiors being illuminated by electric lights.

The sectional views in fig. 82 show the arrangement of the engine in detail. Each " line ", or unit, comprises an open-ended cylinder fitted with two pistons, of which the lower acts directly, through a piston-rod, crosshead, and connecting-rod, upon the central throw of a three-cranked shaft, while the upper piston, through a cross-beam, side-rods, crossheads, and



Section through Working Cylinder looking Aft

Section through Scavenge Pump looking Forward

Fig. 82.—Doxford Airless-injection Opposed-piston Marine Oil-engine

side connecting-rods, actuates the two outer cranks of the trio. At the point of maximum compression of the air, i.e. when both pistons are in their in-stroke position, the fuel oil charge is sprayed in between them, the resulting explosion then driving them apart. The impulses on the pistons are transmitted to the three cranks largely as a pure torque, thus relieving the engine frame of nearly all stress, while by giving the upper pistons a shorter stroke than the lower the engine height is reduced, and the balance of vertical forces is attained, the piston strokes being inversely proportional to the reciprocating masses. The two pistons separate with a relative velocity which is the sum of their separate velocities, and thus the working gases are very rapidly expanded, which is thermodynamically advantageous. When near the end of its outstroke the upper piston overruns a belt of exhaust ports extending round the whole circumference, through which the exhaust gases are discharged into an exhaust manifold, and thence escape into the atmosphere up the after funnel. The lower piston similarly overruns a circumferential belt of air inlet or scavenging ports, through which fresh air—from the air reservoir in the entablature already mentioned—enters under a pressure of 1¼ to 2 lb. per square inch, and fills the space between the pistons, driving out any residual burnt gases through the upper ring of exhaust ports. The entering air has a tangential motion imparted to it by the shape of the ports, and the resulting "turbulence" is of advantage to the operation of the engine. It will be observed also that the flow of air through the cylinder is unidirectional. The exhaust and inlet ports are clearly shown in the transverse section of fig. 82, which shows also the two fuel-injection valves placed immediately opposite one another at the middle of the length of the cylinder; thus two jets of finely pulverized fuel are directed against each other, forming a widening cloud with which the whirling air under increasing compression mingles, so tending to uniformity of mixture and complete combustion of the charge. The pistons have deeply dished crowns, so that their edges when nearest are only 1½ in. apart, which confers the advantage that the length of cylinder wall in contact with the gases at the high temperature of explosion is very small. The steel piston heads are water-cooled, but the crowns are uncooled, and accordingly attain a high temperature in working, and so assist the automatic ignition of the charge. The normal compression pressure is only 285 lb. per square inch, while the normal maximum explosion pressure is 570 lb. per square inch; the combustion is of "mixed" type, i.e. partly at constant volume and partly at constant pressure (see fig. 6), so that, in strictness, the Doxford engine comes under the definition of a "semi-Diesel" engine. The two cam-operated fuel valves open at 25° before, and close 25° after, the top dead centre. In addition to the two fuel valves, each cylinder is fitted with a non-return air starting valve, an indicator valve, and a (water-cooled) relief valve loaded to 710 lb. per square inch.

Two cam-shafts are provided, one at the front and one at the back of the engine; these are driven by spur and bevel gearing from the main shaft. The front cam-shaft has two sets of cams for ahead and astern running respectively, and reversing is effected by sliding this shaft axially by means of a large hand lever. The back cam-shaft does not slide, and actuates the back fuel inlet valves, indicator valves, and lubricator cams. When running astern the back fuel valves are cut off, ample power then being obtained from the front valves alone.

The engine is started, and speed controlled, by a large hand-wheel suitably connected with the reversing lever. By this wheel all four cylinders are started on compressed air, while a continued rotation of the hand-wheel cuts off the air and simultaneously brings the front fuel valves into action. A still further rotation increases the lift of the front fuel valves and also brings the back fuel valves into operation. The fuel pressure is regulated by a small hand-wheel on the left of the operator, which controls the trip gear on the suction valves of the fuel pump.

The pistons and cylinders are cooled by distilled water, the water being preheated to about 140° F. before starting, thus warming up the engine and ensuring quick starting under all atmospheric conditions. The temperature of the water at outlet is 155° F. to 170° F.; all cooling water returns to visible flow hoppers on the middle platform, each cylinder and piston having its own thermometer. Thus the operator can at once assure himself that all the cooling arrangements are functioning correctly. The water pressure in the jackets is 15–20 lb. per square inch, and in the pistons about 40 lb. per square inch. The lubricating oil pressure is about 25 lb. per square inch.

At a normal load of 2800 b.h.p. and speed 110 r.p.m., the mean effective pressure averaged about 90 lb. per square inch, and the fuel consumption 0·385 lb. per b.h.p. hour. The mechanical efficiency was 90 per cent, and the temperature of the exhaust gases 450° F. Under an overload trial 3400 b.h.p. was developed, the corresponding mean effective pressure ranging from 100 to 103 lb. per square inch, and speed 118–210 r.p.m. The fuel consumption rose to 0·395 lb. per b.h.p. hour, and the exhaust gas temperature to from 600° F. to 650° F. A slow-running test showed the engine to be capable of turning steadily at only 18 to 19 r.p.m.

To give an idea of the size of the engine, the 13-throw crank-shaft is 38 ft. in length and weighs 38 tons. The main journals are 17 in. diameter, and the crank-pins 18½ in. diameter. The crank-shaft is carried in six spherical bearings. All controls are at the forward end so that all four operators can be seen from a central position, and orders more readily transmitted. All operations are carried out by hand power. A view of the *Bermuda* herself is given in fig. 83.

*Test Results.*—A trial of a three-cylindered two-stroke single-acting opposed-piston Doxford marine engine was made in March, 1929, by Professor Hawkes. The cylinder bore was 15·75 in., stroke of each upper piston 21·26 in., and of the lower pistons 29·92 in. To avoid the necessity of giving the engine a preliminary heating by the jackets the compression pressure was raised to 435 lb. per square inch (at 150 r.p.m.), enabling the engine to be started from cold.

The double-acting scavenging pump was 44·1 in. dia. and 13·78 in. stroke, and was driven by rocking beams and links from the lower cross-head of No. 2 power cylinder; the pressure maintained in the scavenging air trunk ranged from 1·8 lb. per square inch at full load to 3·2 lb. per square inch at quarter load, and the power absorbed by the pump from 62 to 81 i.h.p. Airless or "solid" injection was employed, the injection pressure ranging from 8100 lb. per square inch at full load to 4950 lb. per square inch at quarter load; the three fuel injection pumps were gear-driven from the shaft at about two-thirds engine speed. The lubricating oil pressure was about 25 lb. per square inch at all loads.

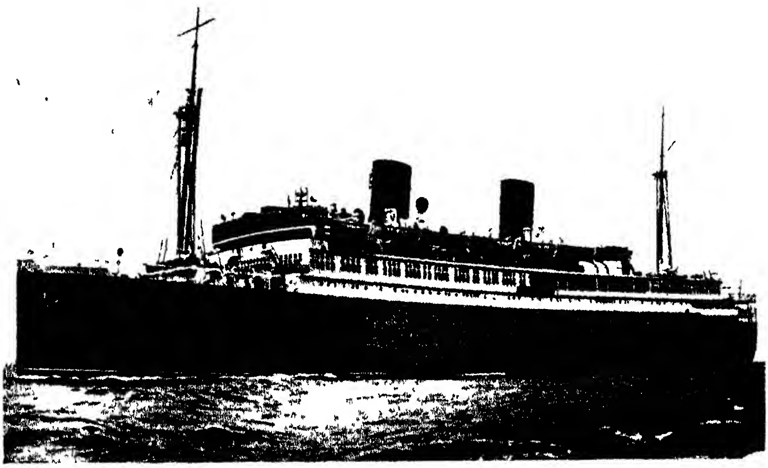


Fig 83.—The *Bermuda*

*Temperature.*—Atmosphere, 49° F. Cooling-water inlet to jackets at full load, 103° F.; outlet, 121° F. Piston-cooling outlet water temperature, 158° F. Lubricating-oil inlet at full load, 84° F.; at outlet, 101° F. The cylinder liners and upper pistons were cooled by distilled water; the lower pistons were cooled by lubricating oil; the outlet temperature of the piston cooling oil averaged 154° F. at full load. The temperature of the exhaust ranged from an average of 563° F. at full load to 291° F. at quarter load.

The brake horse-power was absorbed by a Heenan and Froude hydraulic dynamometer. Indicator diagrams were taken both from the power cylinders and from the scavenge cylinder. The results obtained are exhibited in the table on page 252.

The engine was put on full load 1½ hr. before the commencement of Trial No. 1A. Before commencing each subsequent trial the engine was given time to reach stable conditions at the load necessary for that trial.

Fig. 84 shows the general arrangement of the improved Doxford engine, as fitted to the cargo motor-ships *Devon City* and *Houston City*. The scavenge pump is placed in this case at the rear of the engine for convenience. All of the frames are of welded construction and upper and lower pistons "float" in the cylinder, with a gain in mechanical efficiency and reduction in lubricating-oil consumption in addition to reduction of cylinder wear.

Results of test-trials of the engine, which has a bore of 600 mm. and combined stroke of 2320 mm. (1340 mm. lower, 980 mm. upper), are tabulated on p. 254.

It will be noted that the stroke of the upper piston is less than that of the lower piston, and there are two reasons for this. In the first place, the head room required for the engine is reduced. Secondly, the inertia forces due to reciprocating parts are proportional to the product of the weight of the reciprocating parts and the stroke. Since the weight of the upper set of reciprocating parts is greater than that of the lower set, better balance is obtained if the stroke is reduced in proportion.

Fig. 85 (p. 255) shows one of the four 8000-b.h.p. Doxford engines for the 27,000 ton passenger liner *Dominion Monarch*. It represents the highest powered engine of this type that has yet been built. It is designed to develop its normal power at 133 r.p.m., and has five cylinders with a diameter of 725 mm. and a combined stroke of 2250 mm. (1300 mm. upper, 950 mm. lower). The exhaust gases from the inboard starboard and inboard port

SUMMARY OF RESULTS*

SERIES 'A' (Engine controlled by governor). Date of Trial: 2/3/29.

| | | | | | | |
|---|---|---|---|---|---|---|
| Number of trial | .. | 1a | 2a | 3a | 4a | 5a |
| Approximate load | .. | Full | 3/4 | 1/2 | 1/3 | 1/4 |
| Duration of trial | hr. | 1 | 1 | 1 | 0 | 0 |
| | min. | 21 | 0 | 0 | 17 | 53 |
| | sec. | 49 | 32 | 6 | 19 | 41 |
| Brake load | lb. | 2768 | 2076 | 1384 | 923 | 692 |
| R.P.M. | .. | 183·8 | 185·6 | 185·7 | 186·3 | 186·5 |
| Total fuel used during trial | lb. | 503·5 | 280 | 199·6 | 44·6 | 119·6 |
| Brake horse-power | .. | 1018 | 771 | 514 | 344 | 258 |
| Fuel per b.h.p. hour | lb. | 0·363 | 0·36 | 0·388 | 0·449 | 0·518 |
| Condition of exhaust | .. | Clear | Clear | Slightly shaded | | |
| Mean indicated pressure, | | | | | | |
| Lb. per sq. inch, No. 1 Cylinder | .. | 82 | 63 | 47 | 37 | 34 |
| Lb. per sq. inch, No. 2 cylinder | .. | 85 | 67 | 52 | 38 | 31 |
| Lb. per sq. inch, No. 3 cylinder | .. | 85 | 68 | 46 | 35 | 26 |
| Total i.h.p. | .. | 1166 | 925 | 677 | 551 | 427 |
| Mechanical efficiency, per cent | .. | 87·3 | 83·4 | 75·9 | 66·8 | 60·4 |
| Fuel consumption, lb./i.h.p./hr. | .. | 0·32 | 0·3 | 0·29 | 0·3 | 0·31 |

* By courtesy of Messrs. Doxford.

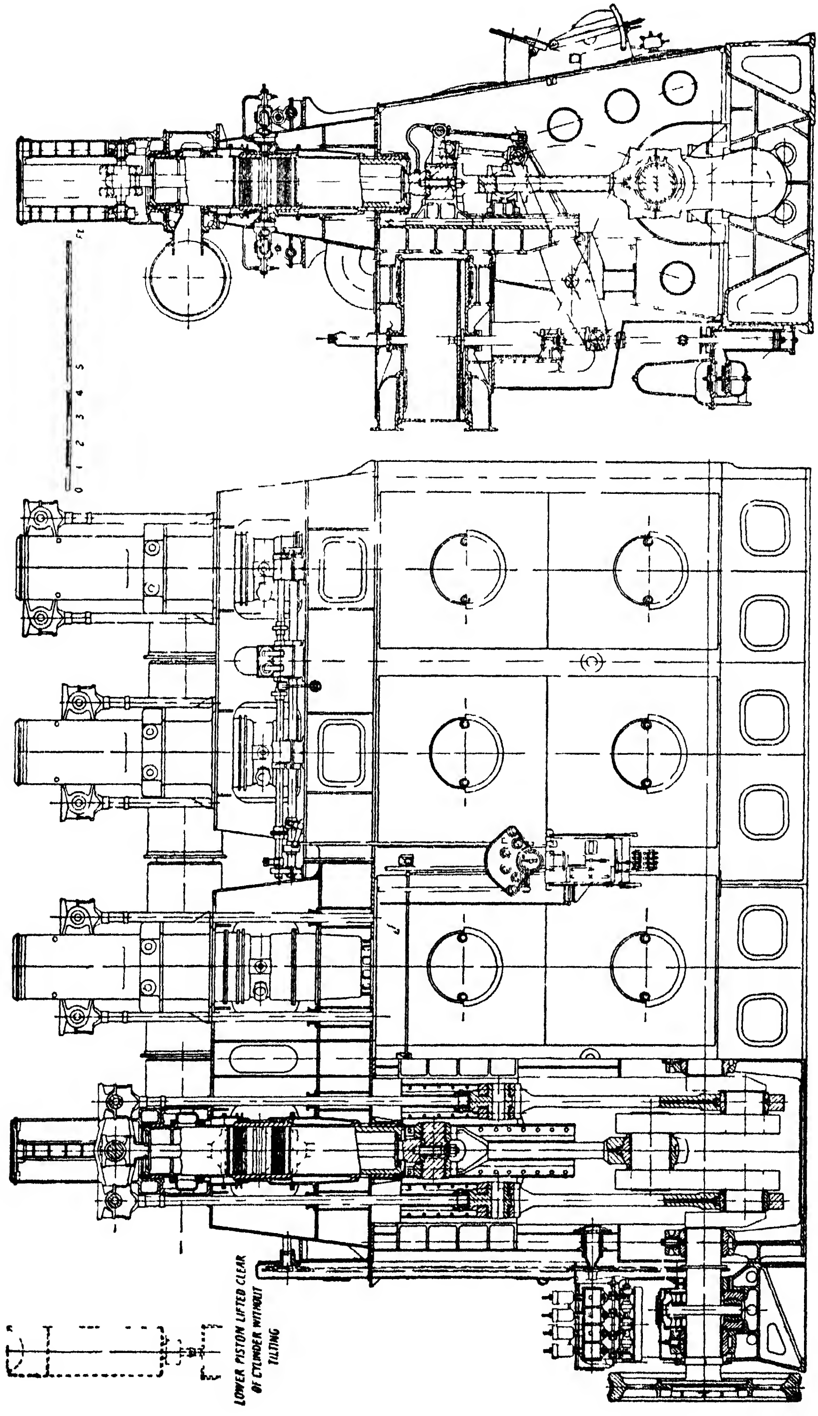


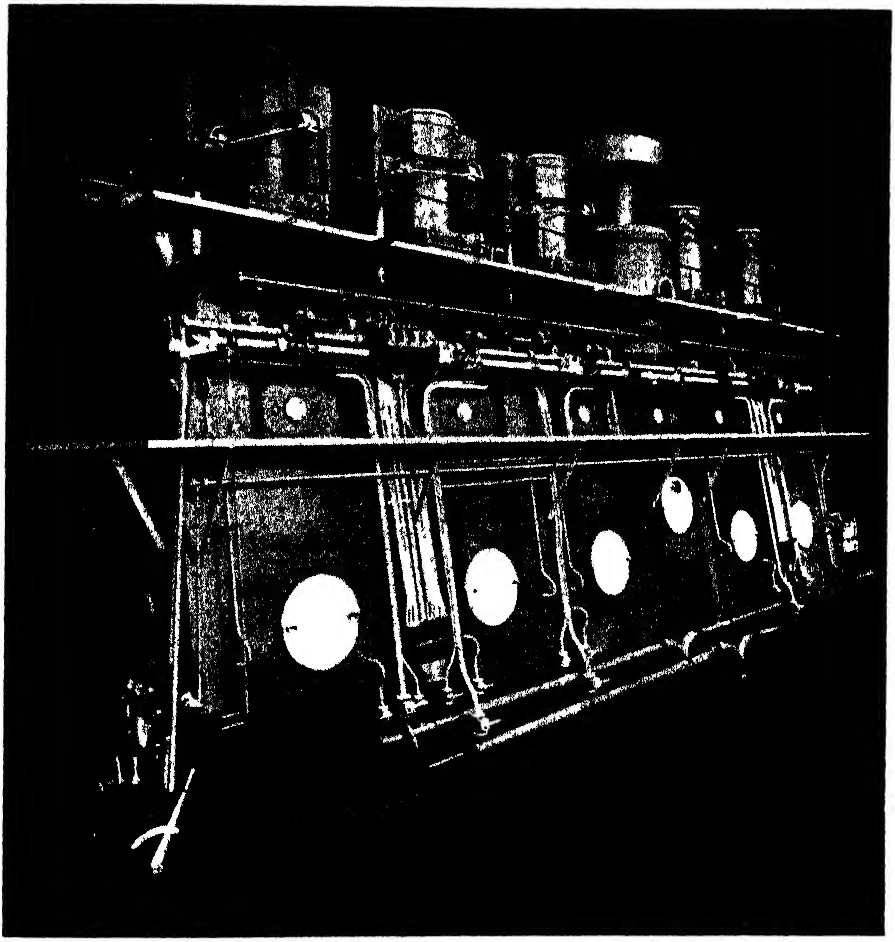

Fig 84 —General Arrangement of the Improved Doxford Engine

RESULTS OF THE TEST-BED TRIALS OF THE DOXFORD ENGINE OF THE "DEVON CITY"

| R.P.M. | S.H.P. | Exhaust Temp., deg. F. | Average M.I.P. | | | | Total I.H.P. | Mech. Efficiency, per cent | Average Injection-oil Pressure, lb/sq. in. | Fuel-oil Consumption | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | No. 1 Cyl. | No. 2 Cyl. | No. 3 Cyl. | No. 4 Cyl., Aft. | | | | lb/S.H.P./hour | lb/I.H.P./hour |
| 93·2 | 2945 | 705 | 85·6 | 87·8 | 90·0 | 85·6 | 3279 | 89·6 | 5000 | ·360 | ·323 |
| 92·5 | 2920 | 707 | 88·9 | 90·0 | 86·7 | 88·9 | 3309 | 88·3 | 5000 | ·359 | ·317 |
| 93·7 | 2960 | 707 | 87·8 | 88·9 | 88·9 | 86·7 | 3331 | 88·6 | 5000 | ·357 | ·317 |
| 93·3 | 2950 | 710 | 88·9 | 88·9 | 88·9 | 86·7 | 3330 | 88·4 | 5000 | ·357 | ·316 |
| 93·4 | 2955 | 710 | 85·6 | 87·8 | 86·7 | 87·8 | 3279 | 89·7 | 5800 | ·355 | ·317 |
| 92·5 | 2920 | 700 | 86·7 | 86·7 | 86·7 | 86·7 | 3292 | 88·8 | 5800 | ·355 | ·315 |
| 92·0 | 2905 | — | 86·7 | 88·9 | 86·7 | 86·7 | 3240 | 89·5 | 5500 | ·355 | ·317 |
| 101·5 | 3200 | 700 | 87·7 | 88·9 | 88·9 | 90·0 | 3620 | 88·2 | 5800 | ·357 | ·315 |
| 100·8 | 3185 | 720 | 88·9 | 88·9 | 88·9 | 88·9 | 3614 | 88·1 | 5800 | ·357 | ·315 |
| 101·3 | 3195 | 720 | 87·8 | 88·9 | 91·1 | 91·1 | 3661 | 87·4 | 5800 | ·357 | ·312 |
| 102·0 | 3215 | 730 | 84·6 | 88·9 | 92·2 | 91·1 | 3668 | 87·6 | 5100 | ·358 | ·314 |
| 102·0 | 3215 | 730 | 86·9 | 91·1 | 90·0 | 91·1 | 3684 | 87·1 | 5100 | ·359 | ·314 |

By courtesy of *The Shipbuilder and Marine Engineer* engines are discharged through two Clarkson waste-heat boilers, each designed to raise 5000 lb. of steam per hour at 100 lb. sq. in. from feed water at 60° F. (See *The Motor Ship*, September, 1938.)

The newer engines are appreciably shorter than the earlier designs for the same power, the reduction in length ensuring greater rigidity of the crank-shaft and structure. A considerable saving in weight has also been made.



By courtesy of *The Motor Ship*

Fig 85.—8000-b.h.p. Doxford Opposed-piston Diesel Engine

**The Fullagar Engine.**—This is an ingenious arrangement of the two-stroke single-acting double-piston Oechelhauser type engine. The power unit is illustrated diagrammatically in fig. 86; it includes two vertical open-ended cylinders placed side by side, the pistons A and D, and also B and C, being connected across by diagonal tension-rods GH and LK respectively, thus enabling a flat two-throw crank-shaft to be used.

The advantages of balanced action, unidirectional scavenging, and simple-torque drive are retained, while the side rods, return connecting-rods, and three-throw crank of the Oechelhauser and Doxford designs are avoided (see fig. 82).

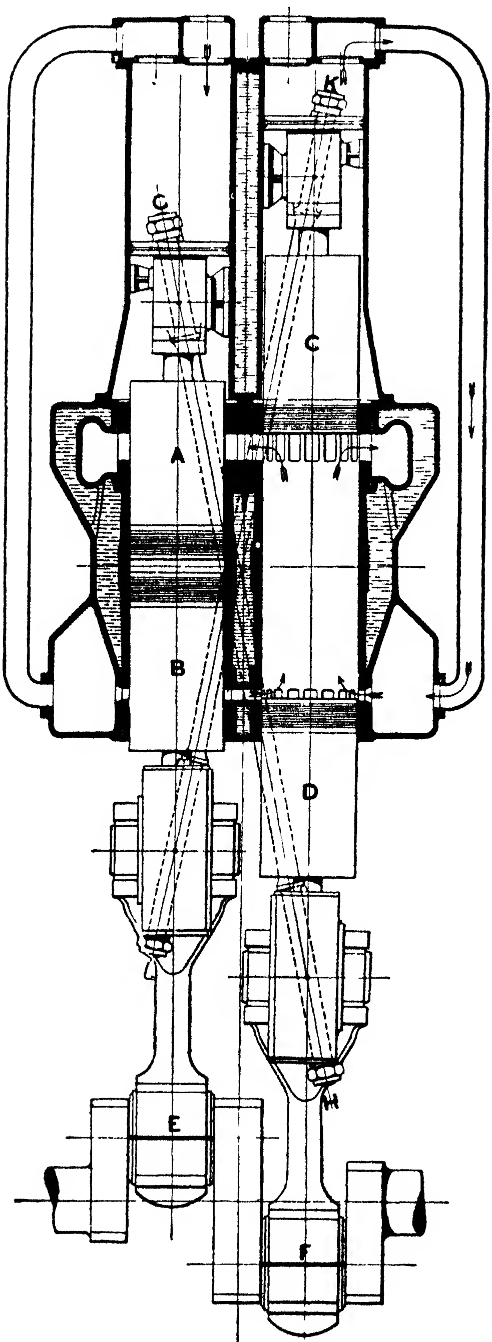


Fig. 86.—The Fullagar Engine

The obliquity of the diagonal rods is not great, and the side thrust of these is taken by crossheads running on water-cooled guides as indicated. Each power unit furnishes two torque impulses per revolution, and the engine is capable of very slow-speed running.

In the diagram the cross-heads are shown boxed in and made to serve as scavenging pumps (of rectangular section), supplying air at 1·5–2 lb. per square inch pressure to the working cylinders, but in some cases separate scavenging pumps are used. As no bearing is needed between the two cranks of a unit, the length of the engine can be kept low, though its height is necessarily increased by the presence of the cross-heads, and the necessity of an equal stroke for both the top and bottom pistons.

The Fullagar engine has proved a valuable type in land installations for driving large generators, a 550-h.p. engine being installed at Gateshead-on-Tyne in 1914, while 1125-h.p. engines are at work in Bermuda, the Sudan, &c.

Marine Fullagar engines have also been built by Messrs. Cammell Laird, and by Messrs. Palmer. The motor-ship *Malia* was fitted, for example, with two two-unit 1000-b.h.p. Fullagar engines with cylinders 18·5 in. diameter, each piston

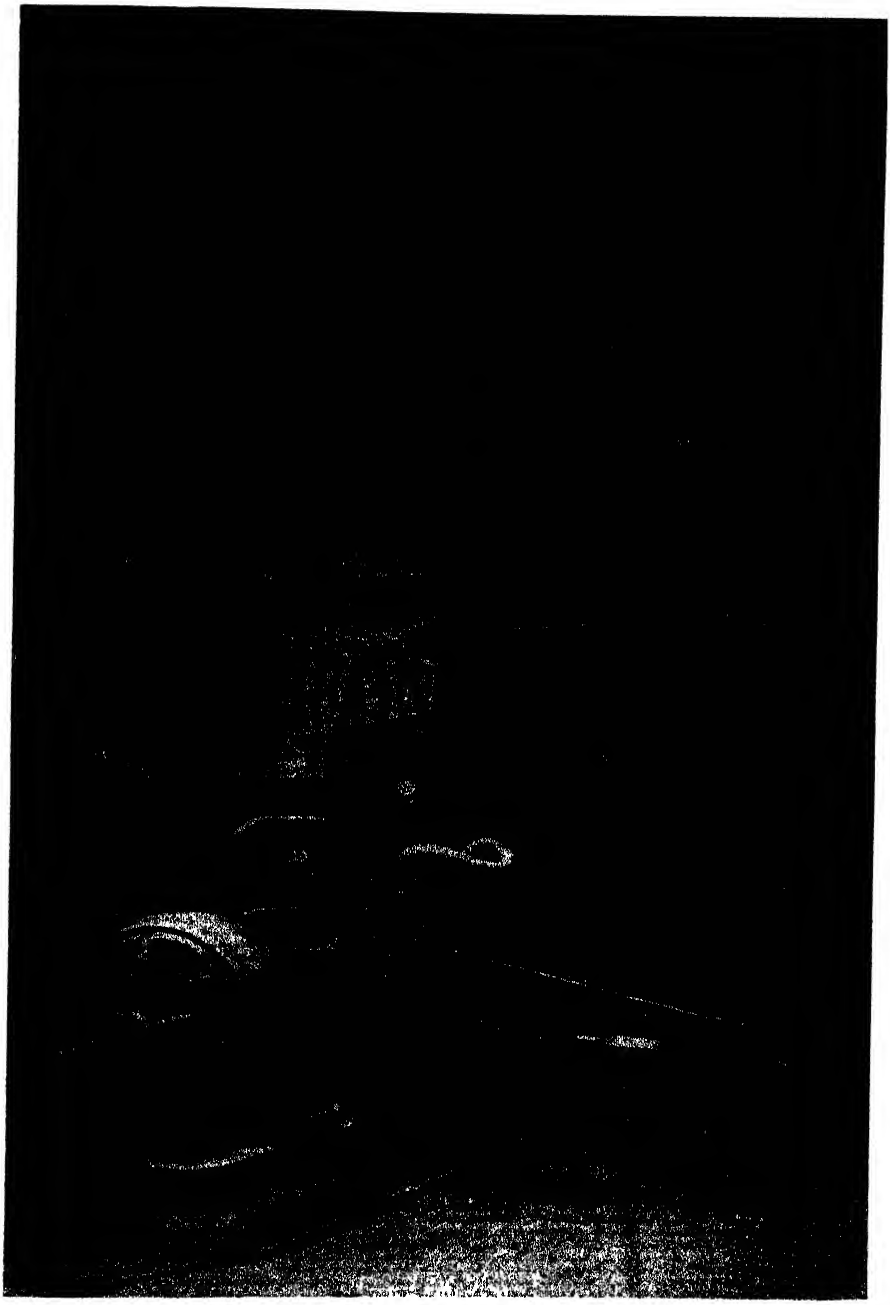


Fig. 87.—Single-cylinder Scott-Still Oil-engine. 2 Cycle, 22 in. diameter, 36 in. Stroke, developing 350 b.h.p. at 120 r.p.m.

having a stroke of 25 in; speed 110 r.p.m. The air-starting and oil-fuel valves were in the centre of the cylinders, and were operated by a horizontal cam-shaft, gear-driven from the crank-shaft. The jackets and pistons were water-cooled.

Tests of a 100-b.h.p. Cammell Laird Fullagar engine gave 1000 b.h.p. and 1400 i.h.p. at 115 r.p.m., the corresponding mechanical efficiency being 71·4 per cent. The fuel consumption was claimed to have the very low value of 0·391 lb. per b.h.p. hour, corresponding to only 0·28 lb.* per i.h.p. hour.

No marine engines of this type appear to have been constructed in recent years.

**The Scott-Still Engine.**—As is well known, a considerable proportion of the heat of combustion of the fuel in internal-combustion engines finds its way into the cylinder jackets, in order that the inner surfaces of the cylinder may be maintained at a temperature to permit of reliable working, and also into the exhaust pipe, because of the mechanical impossibility of abstracting from the products of combustion more than a limited quantity of heat.

With a two-stroke Diesel engine, including blower, in normal full-load running, roundly 35 per cent of the total heat of combustion appears as brake horse-power, 30 per cent is discharged in the exhaust gases at a temperature of 500°–800° F., 25 per cent appears in the cooling water, 5 per cent is lost in engine friction, 3 per cent in driving the air compressor, and 2 per cent in the blower. Special types of boiler have long been employed for utilizing some of the heat of the exhaust gases, and commonly 1¼ to 1½ lb. of steam (from and at 212° F.) is obtained from them per b.h.p. hour.

The diagram (fig. 54, p. 203) and illustration (fig. 87) show the principle of the Still engine. On the top of the piston it can be said that the normal Diesel combustion takes place. That portion of the heat of combustion which passes to the water-jacket is utilized to raise steam in this water-jacket, the construction of the parts, the liner, and the jacket being suitably designed and proportioned to withstand the pressure therein contained. The heat of the exhaust is further harnessed to generate steam. The total steam generated is led to the underside of the piston, and serves to augment the main work derived from the combustion of unit quantity of fuel in the main cylinder. The gain in economy is not the only advantage; the jacket is not the only boiler, but is coupled to an ordinary auxiliary boiler fitted with an oil-burner similar to oil-fired steam plants, which auxiliary serves to generate the steam with which the engine is started. In this way it will be appreciated that the engine is not started as a combustion motor in the cold condition, but is always hot. Manœuvring is effected by steam operating on the bottom of the main piston.

With this motor the piston is not cooled in the ordinary sense. A certain proportion of the heat of combustion passes through the piston

* Compare Professor Hawkes' trial results with Doxford engine, *supra*.

into the steam, superheating this steam, and so permitting it to do more useful work than otherwise would be possible with saturated or wet steam.

The compression ratio used in this engine is 8·6, which is appreciably lower than in other types. This lower compression ratio is practicable owing to the higher working temperatures in the cylinder. It must be borne in mind that only about 15 per cent of the heat in the steam generated can at best be converted into work, so that if the waste heat in the steam generated is 50 per cent of the heat value of the fuel only 7·5 per cent of the heat of the fuel is returned as work. If the engine utilizes 34 per cent of this heat, the percentage increase of power would thus be 22. Actual trial results show about 10 per cent increase of indicated power, but the mechanical efficiency is 90 per cent as against 80 per cent with the other types, so that the increase in brake horse-power is about 11½ per cent.

The motor-ship *Dolius*, built in 1924 and fitted with a Scott-Still engine of 2500 b.h.p., was still in service in 1938.

Early in 1928 the twin-screw M.S. *Eurybates*, 425 ft. × 54·5 ft. beam, 12,130 tons displacement, was fitted with a pair of improved Scott-Still combined Diesel and steam-engines each with a normal output of 2500 b.h.p. at 105 r.p.m.

In the engines of the *Dolius* the steam generated was employed on the underside of the pistons, this being the original Still design, but in the *Eurybates* each main engine comprises five single-acting two-stroke Diesel cylinders of 27-in. bore and 45-in. stroke, and two double-acting steam cylinders of 24-in. bore and 45-in. stroke at the forward end, all working on a common crank-shaft. Fig. 88 (p. 260) gives an external view of the port engine. Both the internal-combustion cylinders and the steam cylinders are, generally, of normal type. In the former a compression pressure of 375 lb. per square inch is employed, which is found sufficient to cause spontaneous ignition of the fuel with airless injection, as the jacket water is kept hotter than usual in the Still engine. Moreover, as starting and manœuvring are effected by the steam cylinders, the only valve required in the Diesel cylinder cover is the automatic spring-loaded pump-opened fuel inlet. The scavenge air is supplied by an independent steam-turbine-driven blower, operated by the exhaust from the main engine steam cylinders.

A noteworthy feature is the provision of rotary valves in the exhaust. There are five such valves, one for each Diesel cylinder, driven by spiral gear at half speed from the main shaft, and these close the exhaust on the upstroke earlier than usual, thus enabling the blower to effect a slight supercharge.

The combustion cylinders' liners are jacketed by water in circuit with a simple vertical multi-tubular boiler or "regenerator" working at about 15 lb. per square inch above atmosphere, so that the jacket water is maintained at the corresponding temperature of about 250° F. Steam is raised and maintained in this regenerator by passing the exhaust gases from the main engine through it, and this steam is augmented by the steam formed in the jackets of the internal-combustion cylinders. A special feature claimed for this system is that all the heat added to the boiler water is added as latent heat and wholly utilized in steam raising, and owing to this the amount of steam produced is of commerical value.

As a preliminary step in the recovery of the heat carried off in the exhaust gases, these are first passed through a high-pressure boiler, capable, when oil-fired, of supplying steam at a pressure of up to 180 lb. per square

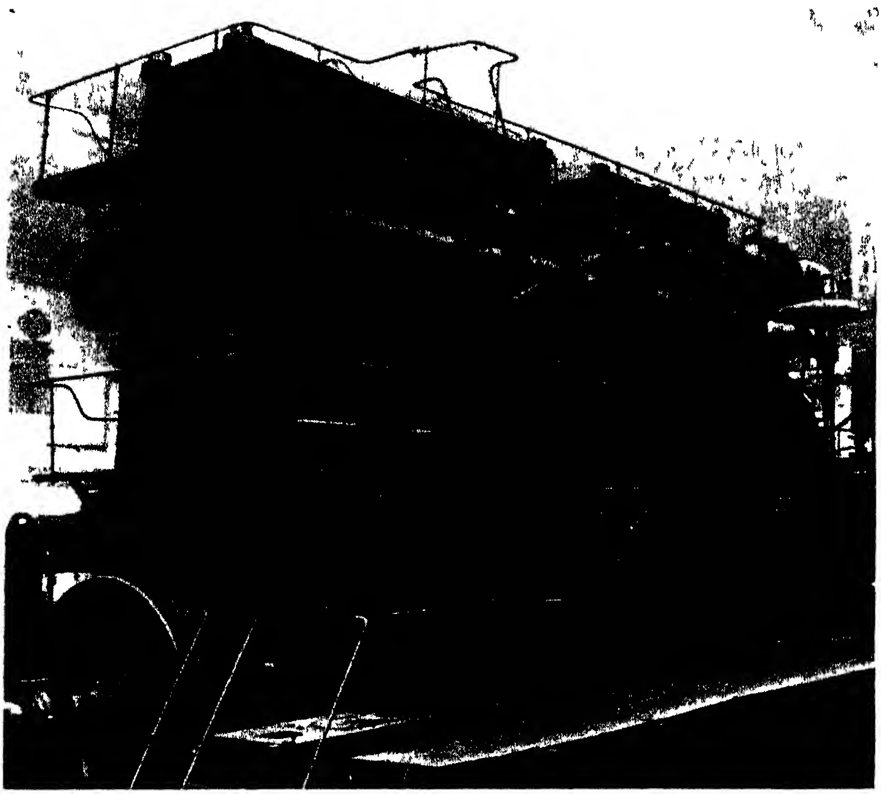


Fig 88 —Scott-Still Engine Port Engine of the *Eurybates* on Test-bed

inch; here some of the heat of the exhaust is given up. The gases are next passed through the low-pressure regenerator, and are thence discharged through the funnel into the atmosphere. The steam thus obtained is used in the two double-acting steam-engine cylinders at the forward end of the main engine, and the total power output is thus increased.

In order to start the main engines the high-pressure boiler is oil-fired, and steam is raised; the engine is then started as a steam-engine. At the same time the jacket water of the five Diesel cylinders is also heated, and this accelerates the starting of these internal-combustion motors. When all is well warmed up, the fuel supply to the boiler is cut off and the only steam then used is that produced from the heat derived from the exhaust gases, at about 15 lb. per square inch pressure.

In the event of the failure of the internal-combustion cylinders the boiler is capable of supplying sufficient steam to drive the vessel at a moderate speed; and again, a heavy overload can safely be taken by oil-firing the high-pressure boiler and thus greatly increasing the power output of the steam end of the main engines The exhaust steam from the engine cylinders, at about 18-in. vacuum, passes through the steam-turbines driving the scavenge blowers and finally enters the condenser at 28-in. vacuum.

**Large Double-acting Four-stroke Engines.**—A notable installation of a large double-acting four-stroke marine Diesel was that of the Swedish-American twin-screw liner *Kungsholm*, completed towards the end of 1928, and engined by Burmeister & Wain This important vessel, 609 ft. long by 78 ft. beam, 21,530 tons gross, accommodating 1575 passengers in addition to cargo, has a service speed of 17·5 knots, and is propelled by a pair of eight-cylindered four-stroke double-acting Burmeister & Wain Diesel engines of 33·1-in bore and 59·1-in. stroke, running normally at

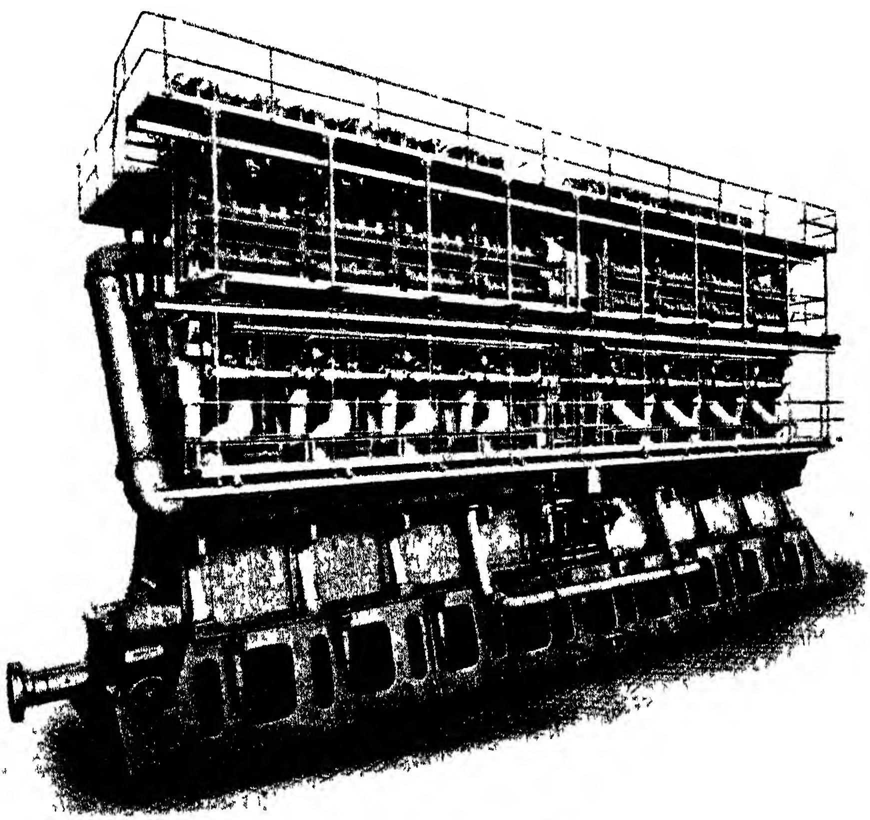
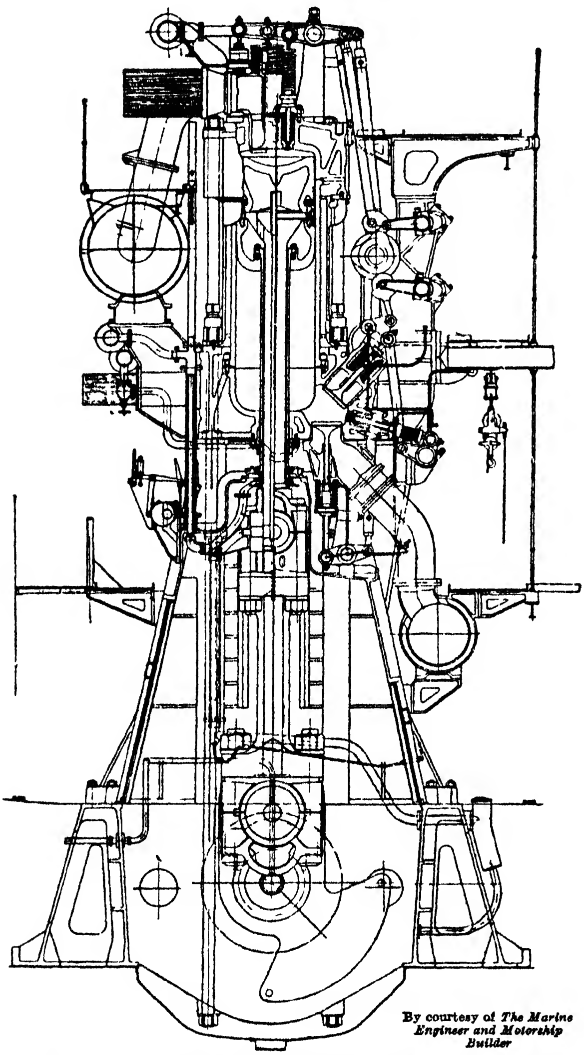


Fig 89 —Port Engine of the *Kungsholm*

100 r.p.m., and developing a total at this speed of 18,000 i.h.p. By increasing the speed to 120 r.p.m. the output is raised to 20,000 i.h.p. An external view of the port engine is given in fig. 89. Air-blast injection is used. Three auxiliary Diesel engines each driving a tandem two-crank three-stage air compressor (one being a stand-by) charge the reservoirs from which the air necessary for the fuel injection and air starting is supplied; these auxiliary Diesels are each four-cylindered four-stroke single-acting Burmeister & Wain engines running at 160 r.p.m. and developing 850 b.h.p.



By courtesy of *The Marine Engineer and Motorship Builder*

Fig. 90.—Sectional View of the B. & W. Double-acting Engine as installed in T.M.S. *Kungsholm*

There are also three three-cylinder trunk-piston Burmeister & Wain auxiliary engines, each of 700 b.h.p., driving 450-Kw. generators at 175 r.p.m. which supply current to a complete set of motor-driven pumps serving each main propelling engine. There are two pumps for sea-water cooling, and two for forced lubrication-oil circulation for each main engine, one of each being a stand-by. The cooling-water pumps are centrifugal, each with a capacity of 300 tons per hour, and deliver sea water to the oil coolers, fresh-water coolers, and exhaust manifold; the remainder of the main engine is cooled by fresh water circulated by two electrically-driven pumps each with a capacity of 250 tons per hour. In case of these failing, provision is made for the admission of sea water to the fresh-water circulatory system. The formidable character of the cooling problems which have had to be overcome in these large double-acting engines will be apparent from consideration of the costly and powerful plant which experience has shown it to be necessary to install for this special service. A transverse section through one of the main propelling engines is shown in fig. 90. It is of the usual inverted-vertical enclosed type with large easily removable doors giving access to the crank-chambers. The piston and piston-rod are water-cooled. One of the long tension bolts passing right down from the top cover of the cylinder to the crank-shaft main bearing casting is shown on the left. The cam-shaft is situated about half-way up the length of the cylinder, and the valves are operated by rocking levers worked by roller-ended push rods actuated by the cams.

The great difficulty of arranging the valves in the bottom cover of double-acting four-stroke engines will be appreciated from a study of fig. 90; as there shown, the valves are all grouped in a separate side casting bolted to the lower end of the cylinder, and in this position they are all easily accessible.

Three oil-fuel daily-supply pumps each of 30 tons per hour capacity are installed in the auxiliary engine-room; these pump the fuel oil from the double bottom, and deep fuel tanks in the ship, to the settling service tanks from which, in turn, the main engines, auxiliaries, boilers, and galley range are supplied.

*Trial Results.*—On a 12-hr. full-power trial an average output of 18,357 i.h.p. was obtained at 100·25 r.p.m. Speed tests showed a mean rate of 18·06 knots. On her first voyage from Gothenburg to New York at the end of 1928, the speed averaged 18·2 knots, and mean revolutions of 102 per minute were maintained.

The double-acting engine is necessarily higher than the single-acting trunk-piston type on account of the necessity of a crosshead, although the piston may be shorter when double-acting, having no longer to take the side thrust of the connecting-rod.

**The North-Eastern Marine Engineering Co., Ltd.**—This company, in collaboration with Messrs. Werkspoor, were designers and builders of large double-acting four-stroke marine Diesels in addition to their earlier single-acting types. The Blue Funnel Line *Stentor* was the first British cargo boat to be propelled by a double-acting four-stroke Diesel. The vessel was 449 ft. in length × 54·9 ft. beam, 7820 tons gross, and had a service speed of 13 knots. It was driven by one six-cylindered 32·25-in. bore × 59-in. stroke North-Eastern-Werkspoor double-acting four-stroke engine giving an output of 4000 b.h.p. at 95 r.p.m. Blast air injection was adopted, two high-pressure air compressors being driven from an extension of the 21-in. diameter crank-shaft at the forward end of the engine (fig. 92, p. 265). A transverse section through one engine " line " is given in fig. 91; the upper portion of the cylinder follows generally the normal Diesel practice with the several valves located in the deep water-cooled top cover. The lower portion of the cylinder, however, presents two features of special interest: (1) the compression pressure employed is only about 250 lb. per square inch, and (2) the shape of the underside of the piston conforms to that of the bottom cylinder cover, which it very closely approaches when at the bottom of its stroke. The valves at the lower end are all located in a separate side casting, somewhat as in the Burmeister & Wain engine already described; this casting forms the combustion chamber, and communicates with the cylinder through a relatively narrow passage. The advantages gained are: (1) the temperature of the explosions at the bottom end is reduced, and (2) the piston-rod and gland are protected from the explosion temperature, thus avoiding damage from overheating, which has proved a source of trouble in some double-acting engines. It is stated that the temperature of the piston-rod does not exceed 140° F. The presence of the piston-rod and the diminished compression pressure cause the power output of the lower end to be considerably less than that of the upper, but this loss is considered to be fully compensated for by the practical advantages gained; the mean effective pressure at the bottom end is about 80 per cent of that at the top in normal running.

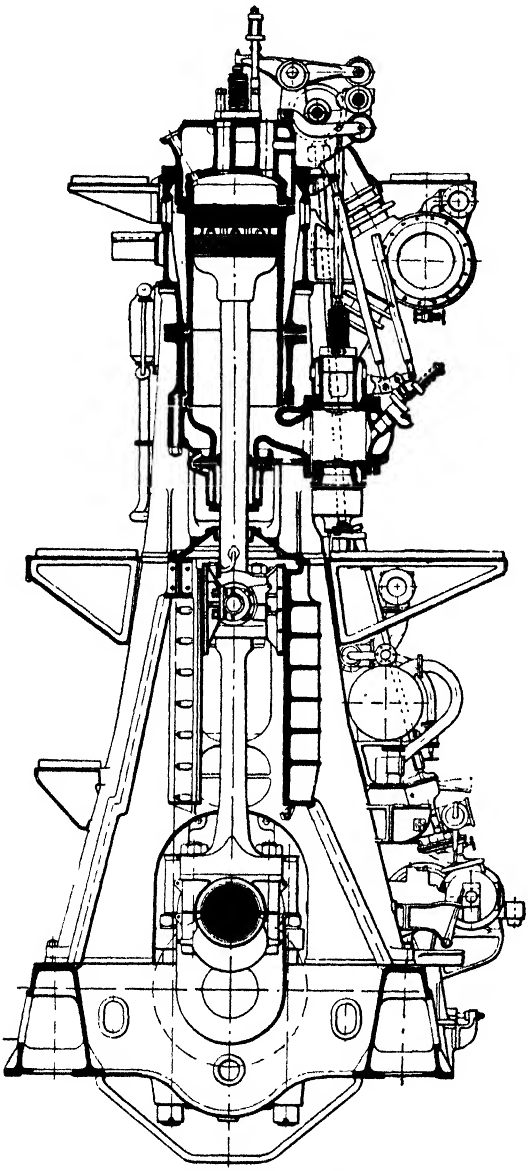


Fig. 91.—Sectional End Elevation through one Cylinder of the "North-Eastern" Double-acting Four-stroke Diesel Engine

The hollow piston-rod, 11 in. in diameter, is attached to the piston by nickel-chrome steel bolts; both piston-rod and piston are fresh-water cooled, the supply entering by a 2·5-in. diameter hole in the rod, and leaving the hollow piston through heavy bronze telescopic piping.

The piston stuffing-box is provided with combination packing, permitting a slight degree of flexibility. In addition to the stuffing-box, the crank-chamber is isolated by a removable diaphragm through which the piston-

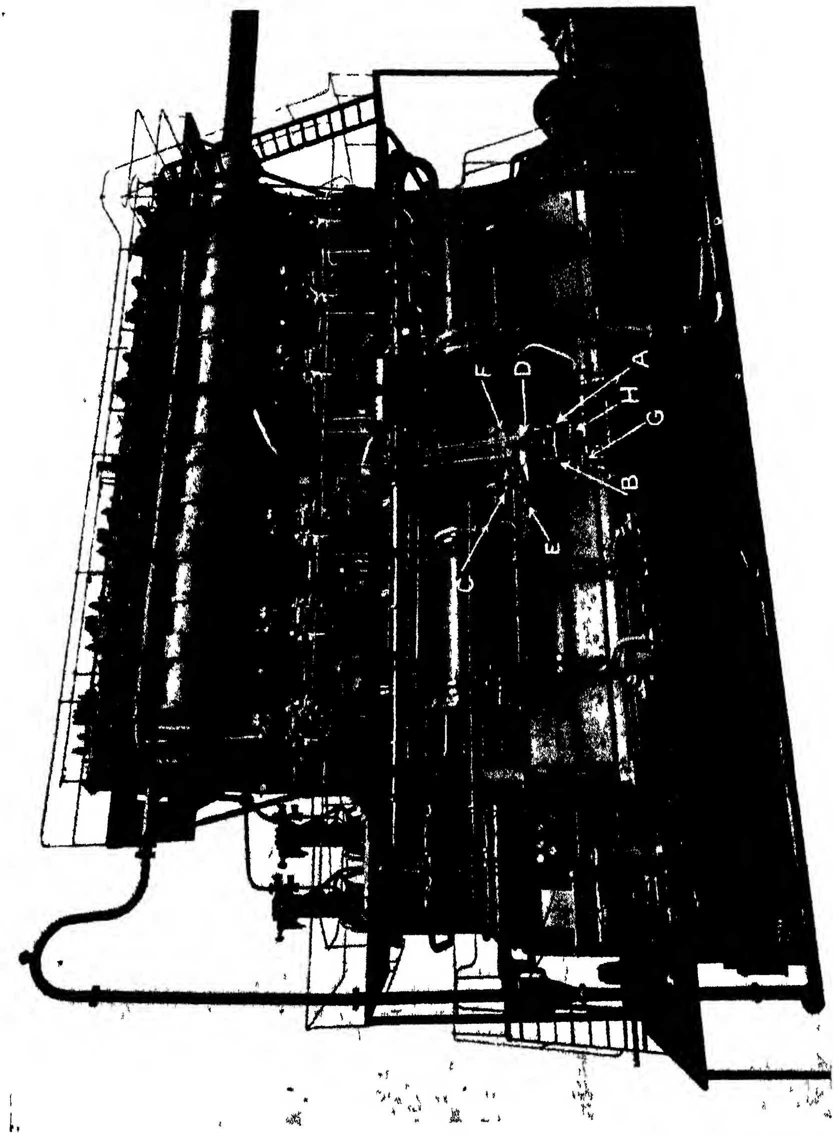



Fig 92 —4000-b h p. North-Eastern Werkspoor Engine of *Stentor*

rod passes as shown. Each crosshead has three guide shoes, two on the astern side and one on the ahead; the upper end of the connecting-rod may be easily disconnected, on removal of a crank-case door, when the rod can be swung outwards between the two astern guides; the lower cylinder cover next being unbolted and lowered, the piston-rod and piston can be brought down so as to expose the latter for examination and such treatment as may be needed. In this way the top cylinder covers, with their high-pressure joints, remain untouched, together with all their valves and gears, and valuable head-room is also saved.

As may be noted from fig. 91, the cylinder liner is made in two parts, with an expansion joint between them, the clearance left being such that it is just taken up when the engine has attained its normal working temperature. To prevent the expansion from affecting the timing of the lower fuel valve a compensating device is fitted.

An external view of the engine is given in fig. 92; the upper valves are operated through rockers by a cam-shaft at the top of the cylinders, the lower being actuated by push rods, as shown in fig. 91. Referring to fig. 92, when the starting lever B is in the "stop" position, the bottom air and fuel valve levers are raised clear of their cams by suitable eccentrics, so that these remain closed; the bottom exhaust valves, however, are held *open*. The starting lever B acts through a small compressed-air motor C. The engine is started as a single-acting motor by admitting compressed air to the top ends of all six cylinders; as soon as motion is well established a further movement of the lever B cuts off the compressed air, and admits fuel to three of the cylinders; the next movement of B admits fuel to the remaining three top ends, and the engine then runs as a single-acting motor. The exhaust valves at the lower ends being held open, the hot exhaust gases from the upper ends have free ingress to the lower combustion chambers, which are thus heated up until their temperature is sufficiently raised to cause automatic combustion of the fuel, as the temperature corresponding to the low compression pressure of only 250 lb. per square inch at the lower ends is insufficient to effect this in a cold cylinder. A gauge E indicates to the engineer in charge when the lower combustion chambers are hot enough to permit the final operation of moving the starting lever into the position which enables all the lower valves to function; the engine then runs as a double-acting four-stroke motor.

The reverse lever A, acting through a servo motor, first raises the valve levers clear of the cams and next slides the cam-shaft axially until the astern cams come into position, when the levers, &c., are lowered.

The several positions of the starting lever are indicated on the dial D. The small compressed-air motor C may be declutched by a lever F, and the starting shaft must then be actuated by a hand wheel. Two levers working in a quadrant G control the lift of the suction valves of the fuel pumps; their lift is also independently controlled by an Aspinall governor located on the beam-lever pump drive at the forward end of the engine. The double-acting fuel pumps, one for each cylinder, are at H.

On a trial trip in heavy weather with the propeller repeatedly out of the water the combined effect of the governing, flywheel inertia, and cushioning effect of the compression was to cause the "racing" speed of the engine only to exceed the normal speed by about 10 per cent.

During the past few years the North-Eastern Marine Engineering Co.

have specialized on single-acting four-stroke engines with supercharging (see p. 296).

**Double-acting Two-cycle Engine.**—Obviously the two-cycle double-acting principle makes a powerful appeal to marine engineers, and holds out at first sight great hopes of developing large powers for a minimum expenditure of weight and space. The thermal problems, however, associated with this design are of an extremely complex nature, and much research and experimental work, connected as much with the qualities of material available as with any other prime factor, has been carried out. At the present time there are a number of firms manufacturing two-cycle double-acting engines of large power.

**The M.A.N. Double-acting Two-stroke Engine.**—The Nuremberg Co. (M.A.N.), for many years famous for its large horizontal tandem double-acting four-stroke gas-engines in land installations, is also prominent in the production of very large and successful Diesels. In 1927 there was in operation at the Hamburg Electricity Works a nine-cylindered double-acting two-stroke M.A.N. engine of 33·8-in. bore and 59·2-in. stroke, direct-coupled to a large alternator, and giving an output of 15,000 b.h.p. at 94 r.p.m. At the date of its installation this was the largest heavy-oil engine in existence.

In 1927 also was completed the great Italian motor passenger liner *Augustus*, 710 ft. in length, 32,650 tons gross displacement, furnished with accommodation for 2178 passengers and a crew of 450, and propelled by four M.A.N. six-cylindered double-acting two-stroke Diesel engines of 27·6-in. bore and 47·3-in. stroke, having an aggregate normal full-power output of 25,000 b.h.p. at 120 r.p.m., and a maximum of 28,000 b.h.p. at 125 r.p.m. The normal full speed of the vessel is 19 knots, and maximum about 21 knots.

The mechanical efficiency of the engine is roundly 80 per cent, so that the maximum of 28,000 b.h.p. implies an indicated horse-power of 35,000. The fuel consumption at normal full power amounted to 0·38 lb. per b.h.p. hour.

The overall length of each engine, excluding the flywheel, is 38·5 ft., maximum height from engine-room floor 28 ft., and width 11·2 ft. The main engine-room is 75 ft. in length, and the auxiliary engine-room 63 ft. An external view of a pair of the main propelling engines is given in fig. 93. About half-way up is the cam-shaft, driving the fuel inlet valves, while the inlet and exhaust manifolds are on the outboard sides of the engines. The arrangement of the air inlet, or "scavenging", and of the exhaust ports adopted by the M.A.N. in their double-acting two-stroke engines has already been described (see fig. 9, p. 144).

In the auxiliary engine-room are installed three six-cylindered four-stroke M.A.N. engines, each of 1200 i.h.p. at 215 r.p.m., direct-coupled to 600-Kw. generators, and also five similar 600-i.h.p. engines coupled to 300-Kw. generators; these supply current, at 220 volts, to the motors driving the auxiliary machinery of the vessel.

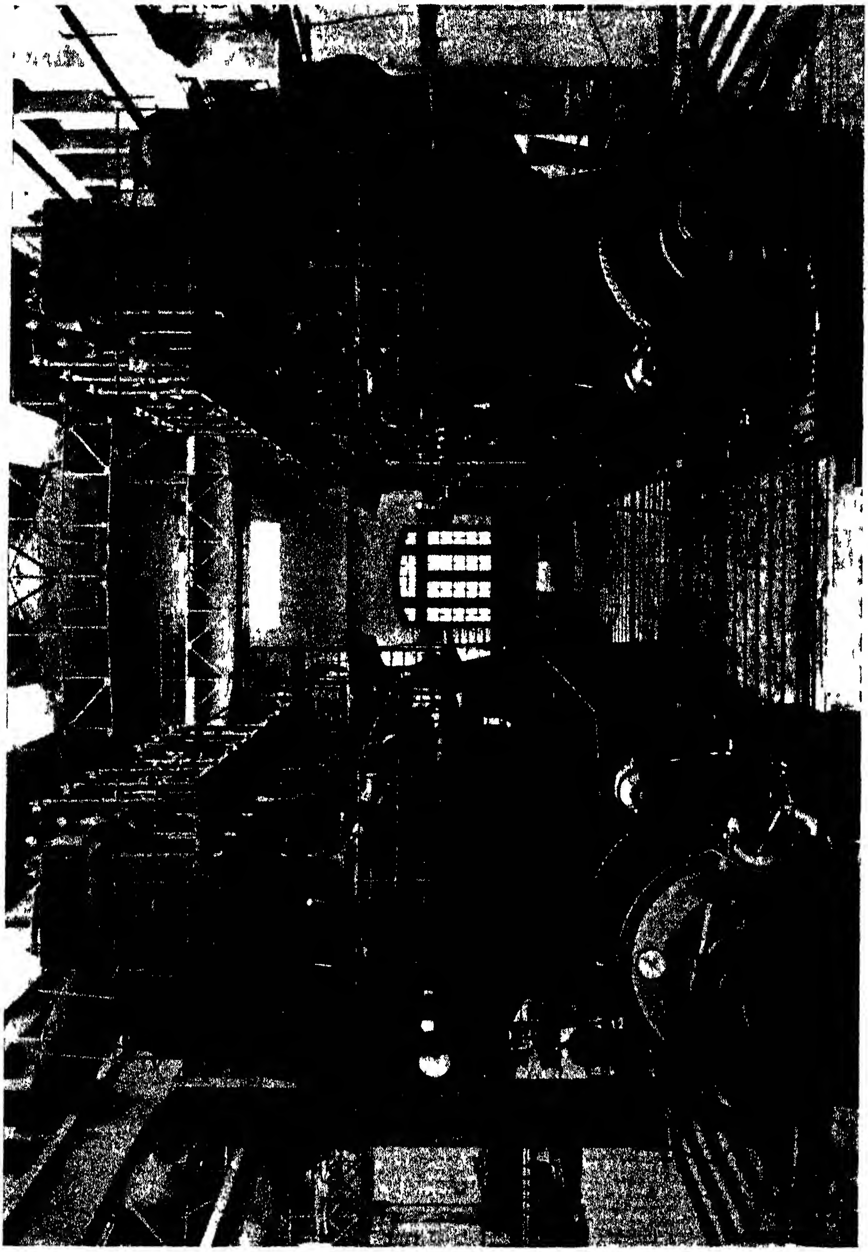


Fig 93 —End View of two of the Main Engines of the *Augustus*

The three larger dynamos are employed mainly in supplying current to the motors driving three Brown-Boveri scavenging blowers (one being a stand-by), each of which is direct-coupled to a 750-h.p. motor running at 1800–2450 r.p.m. Each of these large blowers is capable of delivering from 60,000 to 67,000 c. ft. of air per minute into the scavenging trunk at a pressure of 1·3 to 2·3 lb. per square inch.

Air-blast injection is used in this installation, the high-pressure blast air being supplied by two groups of reservoirs in which it is stored at a pressure of 1100 lb. per square inch, while for starting purposes nine large vertical air reservoirs are provided in which the air is stored at 440 lb. per square inch. Sea water is normally employed for cooling purposes, but provision is made for fresh-water cooling when the vessel is in muddy waters, e.g. the La Plata River; six electrically-driven water-circulating pumps are installed in the main engine-room.

For the forced lubrication four oil-pumps and oil-cooling apparatus are provided, while seven de Laval centrifugal separators are employed to cleanse both the lubricating and fuel oils. The oil is stored in deep tanks, and also in the double bottom of the vessel.

**Geared M.A.N. Double-acting Two-stroke Engines.**—In the constant effort to reduce the weight and size of marine motors, designers have in some cases adopted the device of gearing-down from the engines to the propellers, the reduction gear employed being of either mechanical, electrical, or hydraulic type. With a reduction gear the engine may be run at a much higher speed, with consequent reduction in weight and size, the reduced head room being an undoubted advantage. The reduction in weight is, however, largely offset by the additional weight of the transmission system, while it is still doubtful if the cost is not actually increased. In 1929 two 16,000-ton twin-screw motor-liners were built for the Hamburg-Amerika Line (the *Milwaukee* and *St. Louis* respectively), each propelled by four double-acting two-stroke M.A.N. engines driving two propellers through mechanical reduction gear; these were the first high-speed marine engines of this type, and represented the highest-powered geared-down installation at the date of their completion. In direct-driving marine Diesels where the engine revolution speed is determined by that proper for the propeller, engine weight commonly ranges from about 200–350 lb. per b.h.p.; in these two sets the weight, excluding the reducing gear, is only 115 lb. per b.h.p., including the weight of the turbo blower.

An external view of the reducing-gear drive is given in fig. 94; each propeller is driven by two engines through a " Vulcan " clutch, by enclosed gear-wheels as indicated. The normal speed of the engines is 225 r.p.m., while that of the propeller is only 110 r.p.m. In general design the engines are of the normal M.A.N. double-acting two-stroke type; each has six cylinders of 19·1-in. bore and 25·8-in. stroke, with a normal full-power output at 225 r.p.m. of 3150 b.h.p., the aggregate, for each vessel, thus being 12,600 b.h.p., for a service speed of 16 knots Each engine has two air compressors mounted at its forward end. The overall length is 28 ft., and between the centre lines of the port and starboard pair of engines the distance is only $10\frac{3}{4}$ ft.

The length of the main engine-room is 53 ft., and this is also the length of the auxiliary engine-room, but the forward end of this is occupied by fuel-oil tanks. The ships themselves have an overall length of 574 ft.

In the auxiliary engine-room there are three six-cylinder four-stroke M.A.N. Diesels each driving a 450-Kw. 220-volt dynamo at 245 r.p.m., and one three-cylinder similar engine driving a 220-Kw. dynamo and an air compressor. These supply current to the various auxiliary pumps and

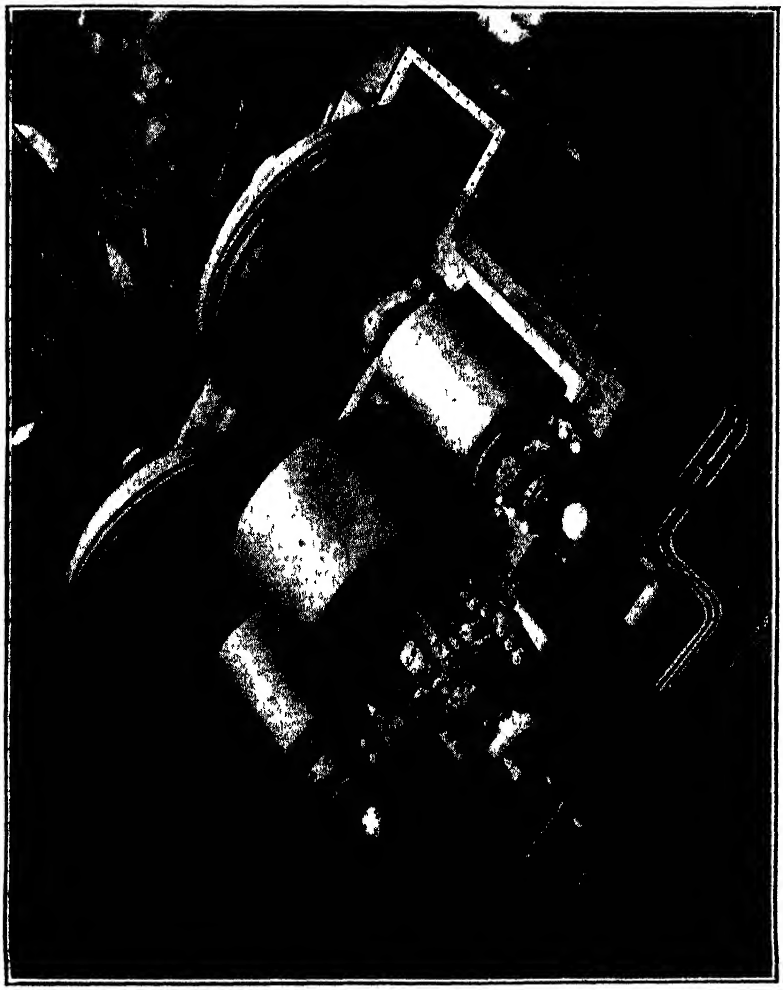


Fig. 94.—The Gear Drive from two 3150-b.h.p. Engines on to a Single Propeller through Vulcan Clutches

machinery. This room also contains three turbo blowers, each electric driven at up to 2950 r.p.m. by a motor of 320–400 h.p.; these blowers supply the scavenging air to the four main engines.

Cooling is by fresh water, circulated by two rotary pumps driven by 160-h.p. motors at 1100–1450 r.p.m.

The four air-starting reservoirs are also in the auxiliary engine-room; they have an aggregate capacity of 2360 c. ft., and the air in these is at 370 lb. per square inch.

Air-blast injection is employed; there are four main air-injection reservoirs, and two reserve " air bottles ", having an aggregate capacity of 98 c. ft., holding the blast air at 1100 lb. per square inch pressure.

The latest type of Vickers-M.A.N. two-stroke cycle double-acting engine is illustrated in fig. 95. In this engine the arrangement of the valve gear has been simplified and airless injection adopted, thus dispensing with the blast-air compressor. The scavenging process is effected entirely by

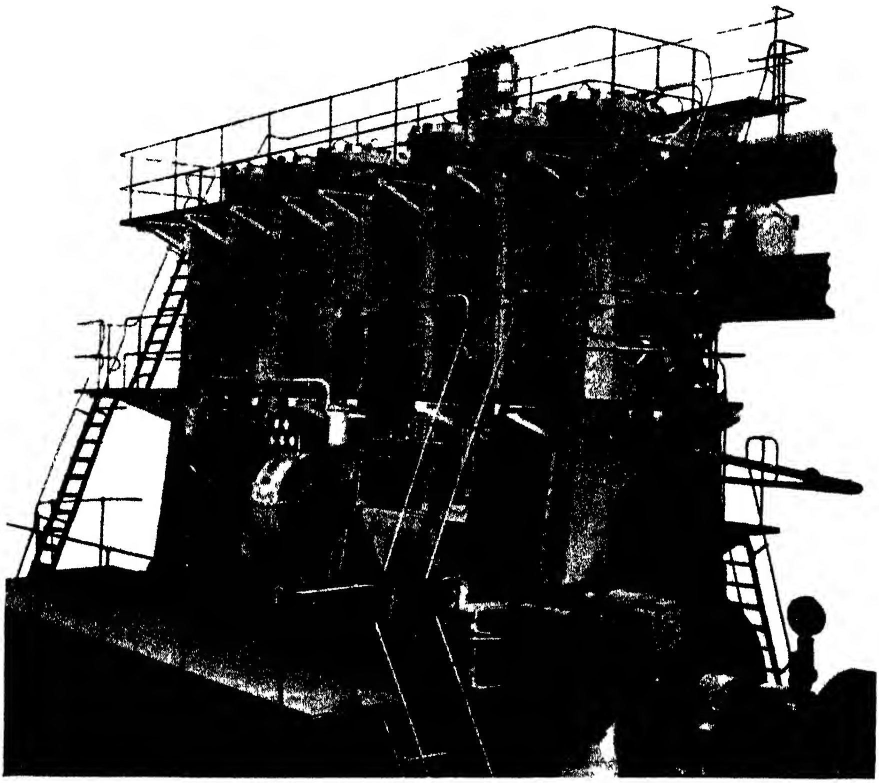


Fig 95 —Vickers-M A N Six-cylinder Two cycle Double-acting Airless Injection Engine, 4000 b h p at 130 r p m

ports in the cylinder, no valves being used. The exhaust and scavenging ports are situated on the same side of the cylinder and occupy about half the circumference, the scavenging ports being so formed that the air-stream first sweeps across the piston-head and is then deflected along the opposite wall of the cylinder to the cylinder cover. From the cover it is reversed and flows back in the opposite direction to the exhaust ports, where it escapes. In so doing, the scavenging air pushes the exhaust gases out of the cylinder and clears the latter completely, so that on the commencement of the compression stroke pure air is available for the ensuing working stroke. This may be termed " loop " scavenging and has the distinct advantage of giving good scavenging of the cylinder with a very low air pressure.

The engine shown in fig. 95 is a six-cylinder unit with a cylinder diameter of 600 mm. and stroke of 900 mm. and gives an output of 4000 b.h.p. when operating at 130 revolutions per minute. The engine is arranged for ship propulsion and is very easily handled as regards ahead and astern running.

**Burmeister & Wain, Copenhagen, with Harland & Wolff, Belfast, and J. G. Kincaid, Greenock, as Licensees in Great Britain.**—This engine is the most recent of the two-cycle double-acting type to come into the marine field. Airless injection is used and end-to-end scavenging, which is claimed to allow a considerably higher mean pressure to be carried in the cylinder than is possible with the normal two-cycle scavenging arrangement. The cylinder is provided with a ring of scavenging-air ports around the circumference at the centre, the exhaust gases being expelled through piston valves, one for each end of the cylinder. The two piston valves are rigidly connected together and are operated by rods from a chain-driven auxiliary shaft. The scavenging-air pressure required is comparatively low, being from 2·5 to 3 lb. per square inch. On account of the good scavenging conditions it is claimed that good combustion can be maintained with mean pressures 5 per cent higher than the maximum for four-cycle unsupercharged engines. A six-cylinder engine of this type with cylinders 620-mm. diameter and 1400-mm. stroke at 100 revolutions per minute gives an output of 7200 b.h.p.

Since the introduction of this type in 1929, it has made remarkable progress, and a notable installation (1936) is that fitted in the motor-vessel *Stirling Castle*, owned by the Union Castle Mail Steamship Co. This vessel is 725 ft. over-all length with a gross tonnage of 25,550. Her machinery consists of twin engines of a total power of 24,000 b.h.p. Each engine has 10 cylinders, cylinder diameter 660 mm. and piston stroke 1500 mm., operating on the double-acting two-cycle airless injection principle.

**Harland-B. & W. Double-acting Two-stroke Engine.**—Fig. 96 shows sections of the latest type of Harland-B. & W. engine. The method of operation of the two exhaust pistons at the ends of each cylinder is clearly shown. The engine illustrated has a bore of 530 mm. (20·87 in.) and a stroke of 1250 mm. (49·2 in.). The fore-and-aft section in fig. 97 and the cross-section in fig. 98 show in more detail the construction and method of cooling of the cylinders and pistons.

The scavenge and exhaust ports are respectively controlled by the main and exhaust pistons. The exhaust pistons are a fixed distance apart and reciprocate together, being driven by eccentrics which are integral with the crankwebs, the angle of advance being usually 187° ahead, becoming 173° astern. When going ahead there is a large lag between opening of exhaust ports and scavenge ports. The exhaust area thus becomes so great that the pressure at end of expansion falls to atmospheric or below. The scavenge air attains a velocity which, with the whirling action, effectively scavenges the cylinder.

The piston consists of two end castings of chrome steel and a centre piece of special cast iron, all strongly bolted together. The piston-rod flange is sandwiched between the bottom and centre components of the piston. There are six piston rings at each end with one scraper ring adjacent to the scavenge ports. Cast-iron carrier rings are caulked into the piston. The mild-steel piston-rod is protected from the hot gases by a cast-iron

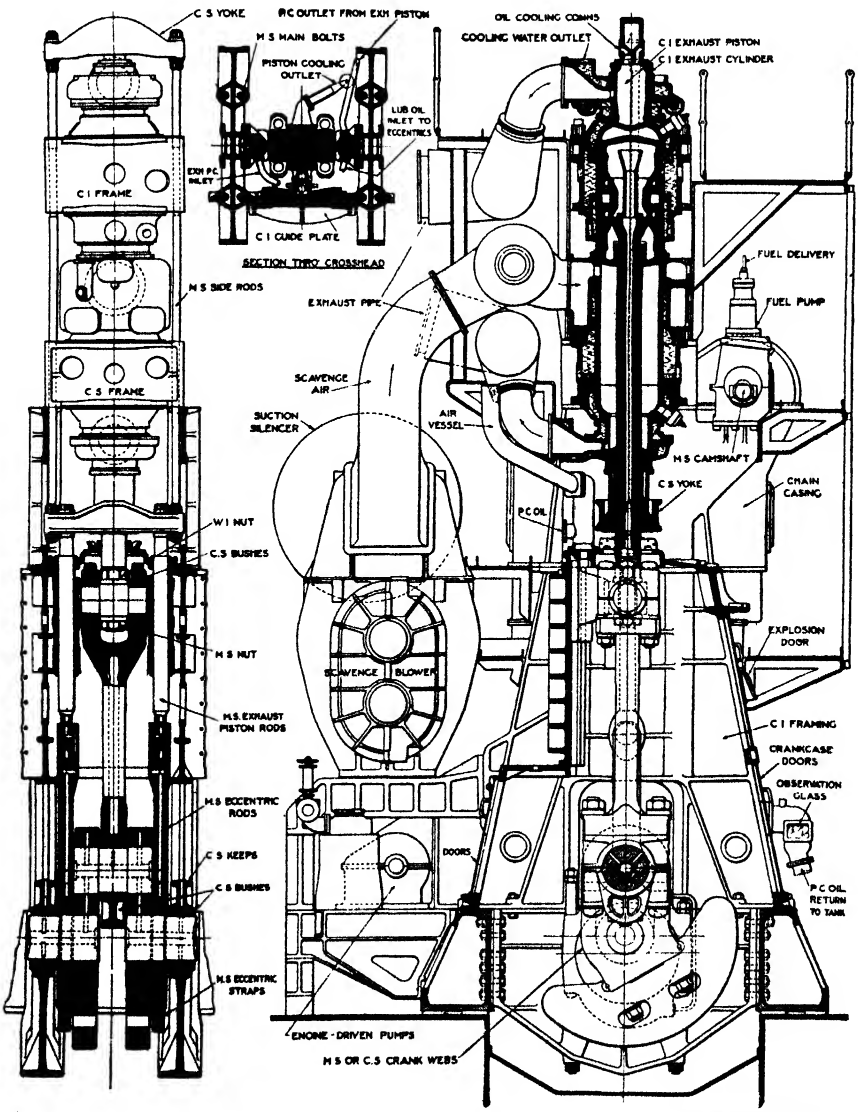



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Fig. 96.—Double-acting Two-cycle Engine

sleeve screwed into the lower part of the piston and sliding freely in a bush at the crosshead end of the rod.

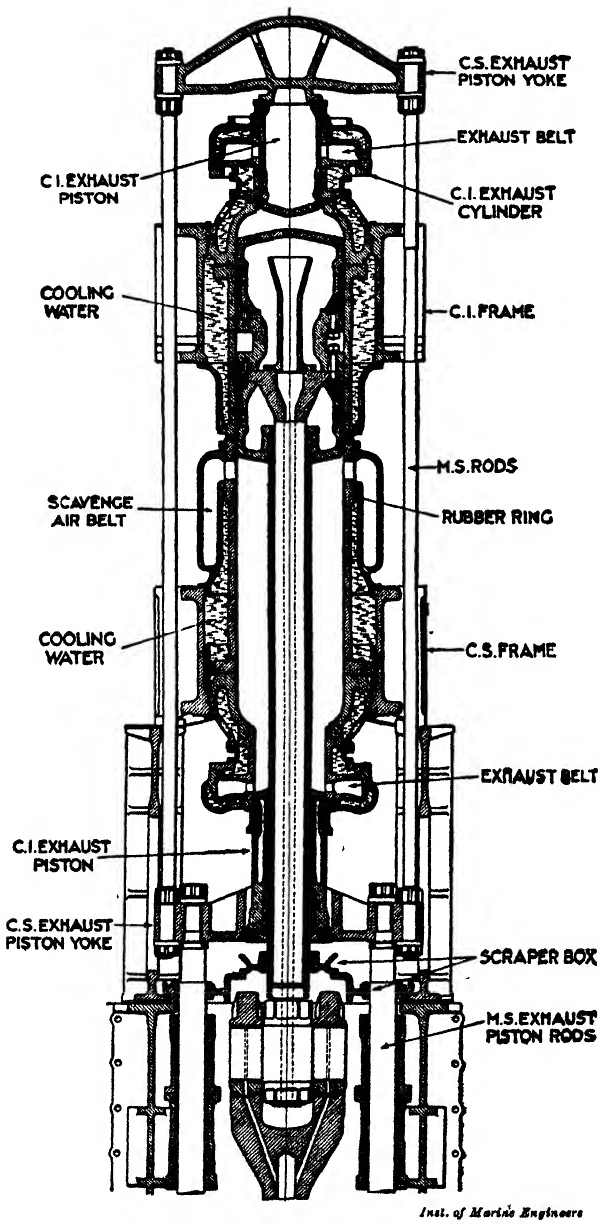


Fig. 97.—Fore-and-aft Section of Double-acting Two-cycle Engine

Each complete cylinder, consisting of covers, liner and frames, is bound together by four long bolts, which continue through the main framing and terminate in nuts under the bedplate. The elasticity of the main bolts and the compressibility of the framing allow expansion of the cylinder liner, &c., to be taken safely.

The cylinder is fresh-water cooled from a closed circuit. The main piston and the exhaust pistons are oil cooled.

A pair of positive displacement blowers at the back of the engine supply scavenge air.

Fig. 99 shows the test-bed results of a 6-cylinder 620/1400 engine tested in the shops of Messrs. Burmeister & Wain. At the designed rating of 7000 b.h.p. at 105 r.p.m. the indicated mean pressure is 96 lb. sq. in., based on the total stroke volume of main and exhaust cylinders, and the fuel consumption is 0·326 lb. per b.h.p. hour.

The double-acting engines shown in figs. 96–98 have been fitted to a total of 1,000,000 s.h.p. during the last few years.

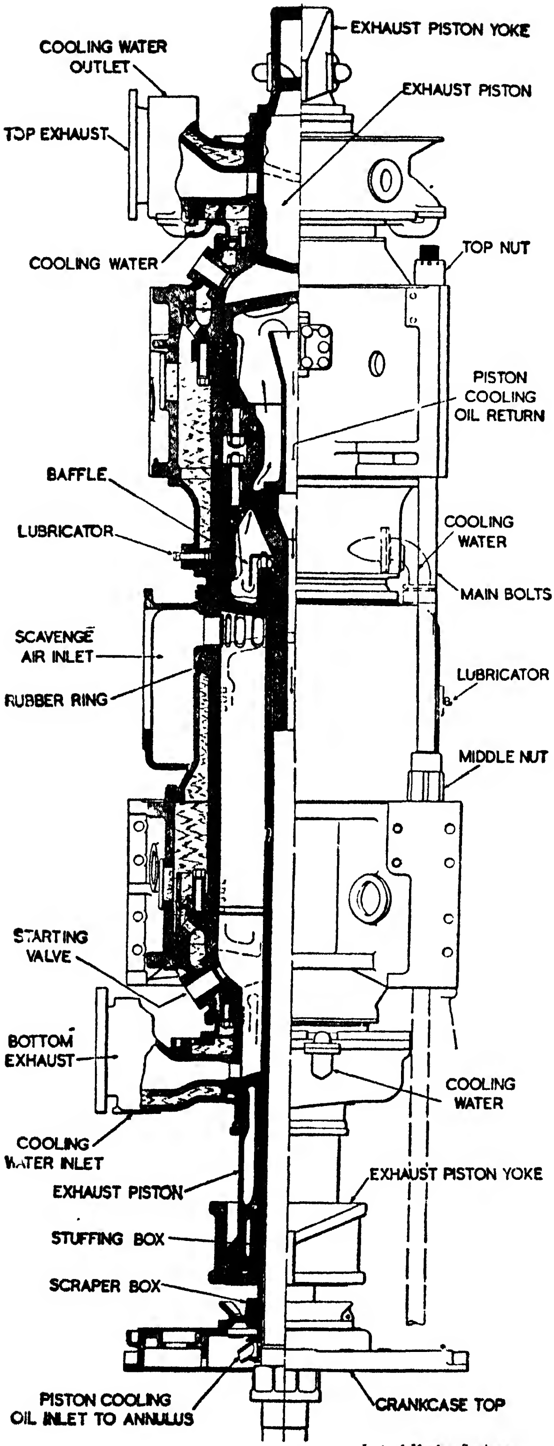


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Fig. 98.—Section of Double-acting Two-cycle Cylinder

**The Richardsons-Westgarth Engines.**—This British firm build a series of three-, four-, five-, and six-cylindered double-acting two-stroke marine Diesel engines, ranging from a three-cylindered design of 21·5-in. bore × 38-in. stroke, rated at 1425 b.h.p. at 105 r.p.m., to a six-cylindered 29·75-in. bore × 52·5-in. stroke design rated at 6000 b.h.p. at 85 r.p.m.

An external view of the three-cylindered Richardsons-Westgarth engine of the tanker *Irania* is given in fig. 100.

The *Irania*, completed in 1929, and still in service in 1938, is a single-

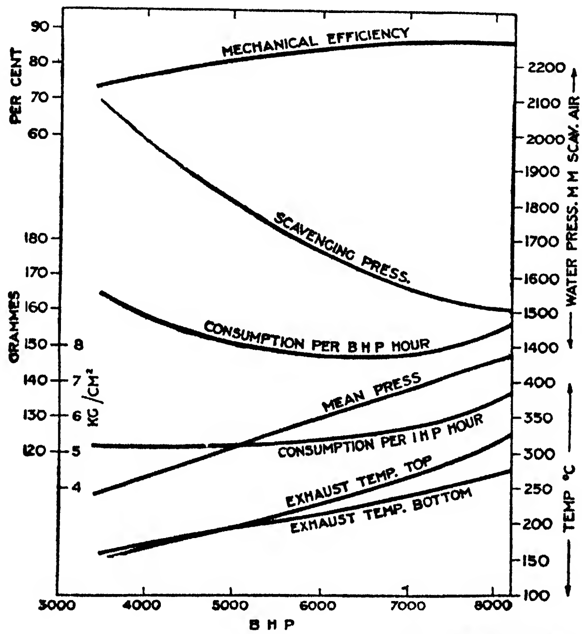


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Fig. 99.—Test-bed Results, 7000-b.h.p. Double-acting Two-cycle Engine

screw bulk-oil carrying ship of about 3000 tons deadweight, and the double-acting two-stroke airless-injection engine of 21·5-in. bore × 38-in. stroke is rated at 1200 b.h.p. at 90 r.p.m., corresponding to an indicated mean effective pressure of about 80 lb. per square inch, and piston speed of only 570 ft. per minute.

The scavenging pump is a double-cylindered horizontally-opposed unit with rotary inlet and delivery valves carried in ball bearings, and driven from the crank-shaft by duplex roller chains. It is spur-gear driven from the main engine shaft at 270 r.p.m., and, as shown in fig. 101, is located at the other end of the engine and encased. The delivery valves are timed to function in phase with the opening of the scavenging ports in the power cylinders, and by this means the scavenging air main, or receiver, may be made of small diameter. All the fuel inlet and starting-air valves are of the automatic spring-loaded type. The fuel pumps, starting-air mechanism, and controls are collected together at the floor-level at the forward end of the engine as shown.

A sectional view through one " line " of the engine is given in fig 101. The upper and lower covers of the cylinders, upper and lower cylinder

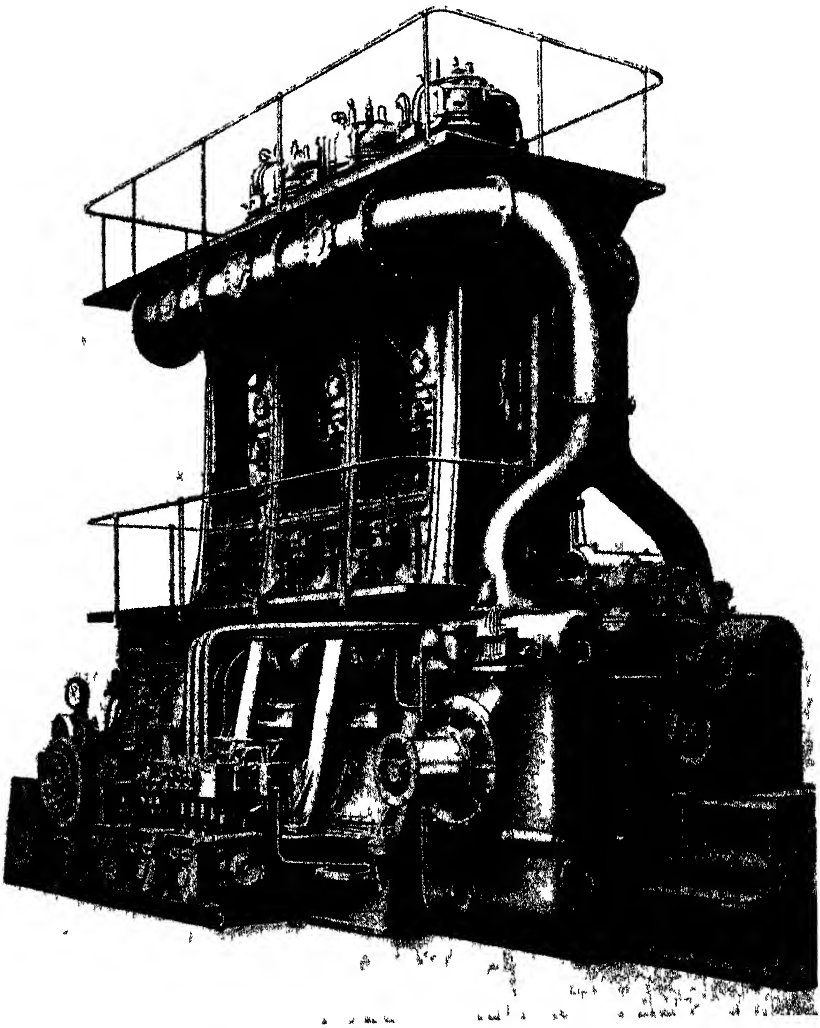


FIG 100 —Richardsons-Westgarth Engine of the *Irama*

liners, the piston halves, and other minor details are all severally interchangeable, thus reducing the first cost and number of spare parts carried, while one size of fuel pump and control unit is used for the whole series of engines built.

The plain ribless cylinder covers are of cast iron. An aluminized steel heat-resisting " shroud " is fitted as indicated at the junction of cover and liner, and extends a short distance downwards into the combustion chamber. This ring fits the liner closely and reaches to the highest point attained by the piston, and thus protects the liner surface from the direct action of the very highly heated flaming fuel jets, and reduces the heat lost to the cooling water.

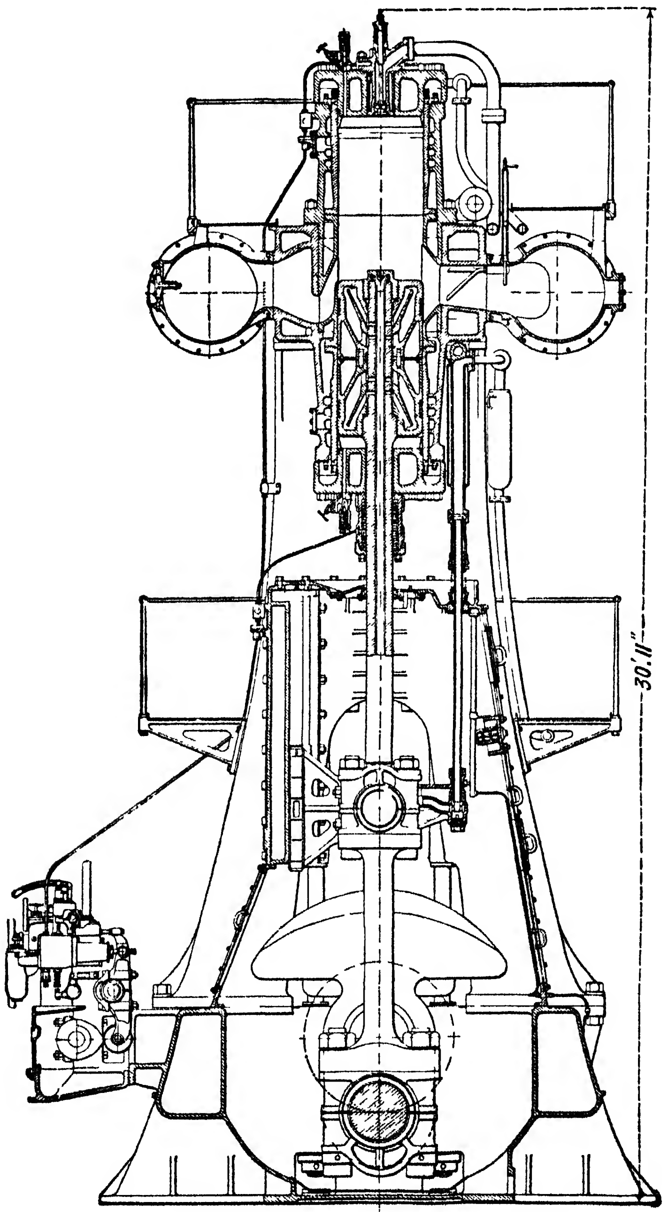



Fig 101 —Richardsons-Westgarth Engine of the *Irama*, Section

Fig. 103A shows in more detail the "shroud" for the lower end of the cylinder and piston rod.

To the lower cover the piston-rod gland is attached externally, thus keeping cool. Each cover is fitted with two vertical fuel-injection valves of automatic spring-loaded type opened by the fuel pump at a suitable pressure. The cover carries also a starting air valve, and a relief (and indicator) valve. An external view of a top end cover showing the arrangement is shown in fig. 102.

The pistons are made up of two similar halves with flat crowns. Fresh water for cooling is passed up the hollow piston-rod, around the lower half of the piston, back into the rod, and then around the upper half of the piston and thence through a bell-mouthed pipe fitted in the top of the hollow rod.

The cylinder liner is in three pieces as shown in the sectional view (fig. 101), but more clearly in the external view of fig. 103. The central portion contains the belts of ports, having on one side two belts of scavenging ports for the top and bottom cylinder ends respectively, and on the other a belt of exhaust ports serving both ends of the cylinder. The scavenge ports are directed steeply upwards and downwards respectively, so that when just opened the side of the piston causes the inrushing stream of air to be deflected almost vertically upwards, and as the port opening increases the stream becomes less vertical, and sweeping downwards thus ensures satisfactory scavenging. The top and bottom pieces of the liner are interchangeable, as already stated, and being separate from the port piece, each can be made of the material best suited to its particular service. The liner is supported at the centre, the ends being free to expand, while the ribless cylinder covers are also designed to permit of expansion when heated without stressing the engine. The recessed rings and the holes by which oil is

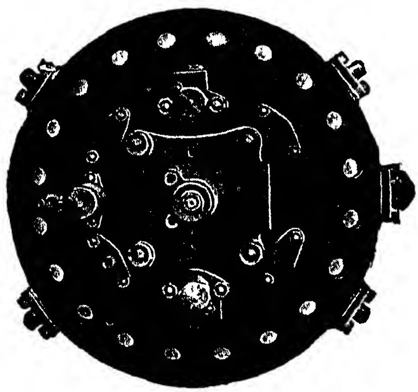


Fig 102.—Top Cylinder Cover

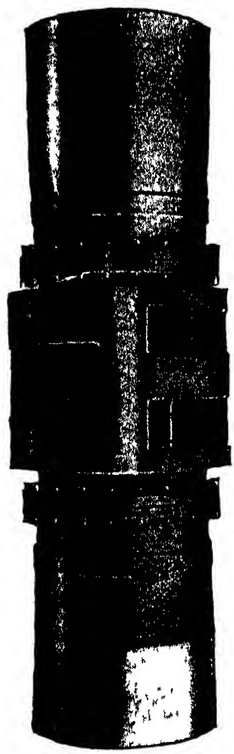


Fig 103 —Cylinder Liner

directly supplied under pressure to the pistons will be noted in fig. 103.

The bars of the exhaust ports are of large section, and as all the cylinder cooling water passes through them they are well cooled, their temperature approximating to that of the partially water-cooled scavenge ports on the opposite side of the liner, thus minimizing any tendency to distortion from unequal expansion. Circumferential ribs are provided in the cylinder jacket to ensure an even distribution of the cooling water around the combustion-space ends of the liner. The cooling water enters the cover at several points (see fig. 102), to ensure uniform cooling of the heated parts.

The combustion chambers are simple annular flat-ended spaces, the volume being such as to give a compression pressure of 350 lb. per square inch. The airless-injection fuel inlet valves are so arranged as to spray the charges of oil in a practically horizontal direction around the piston-rod

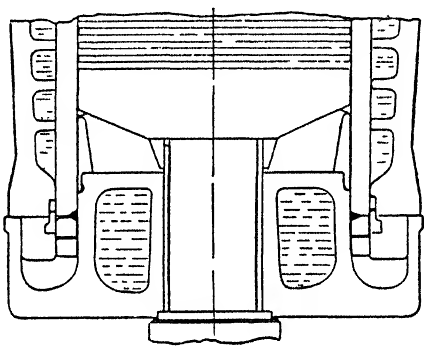


Fig. 103A

nut at the top, and the piston-rod at the bottom, of the cylinder; in this way a very good distribution of the fuel spray is attained. The injection pressure is about 6000 lb. per square inch, and a separate and independent fuel pump is provided for each cylinder end; the pump plungers are operated on their delivery strokes by cams and rockers, the suction strokes being effected by aid of springs.

To start the engine compressed air is admitted to the top cylinder ends only, while fuel is injected into the lower ends when motion has become established; thus the initial firing occurs in the lower combustion chambers, which have not been cooled by the admission of the starting compressed air; this arrangement facilitates starting. Separate ahead and astern cams are provided for the fuel pumps and starting-air valves as usual.

Figs. 104 and 105 (by courtesy of *The Marine Engineer*) show the more recent improvements in the design of the Richardsons-Westgarth engine as fitted in the modernized Silver Liners. The scavenge pumps run at twice engine speed and their discharges are timed in relation to the cylinders, the air being delivered to the cylinder through two double rows of ports as shown. The uppermost and lowermost ports are arranged to give a swirling effect, and their opening and closing are controlled by multiported

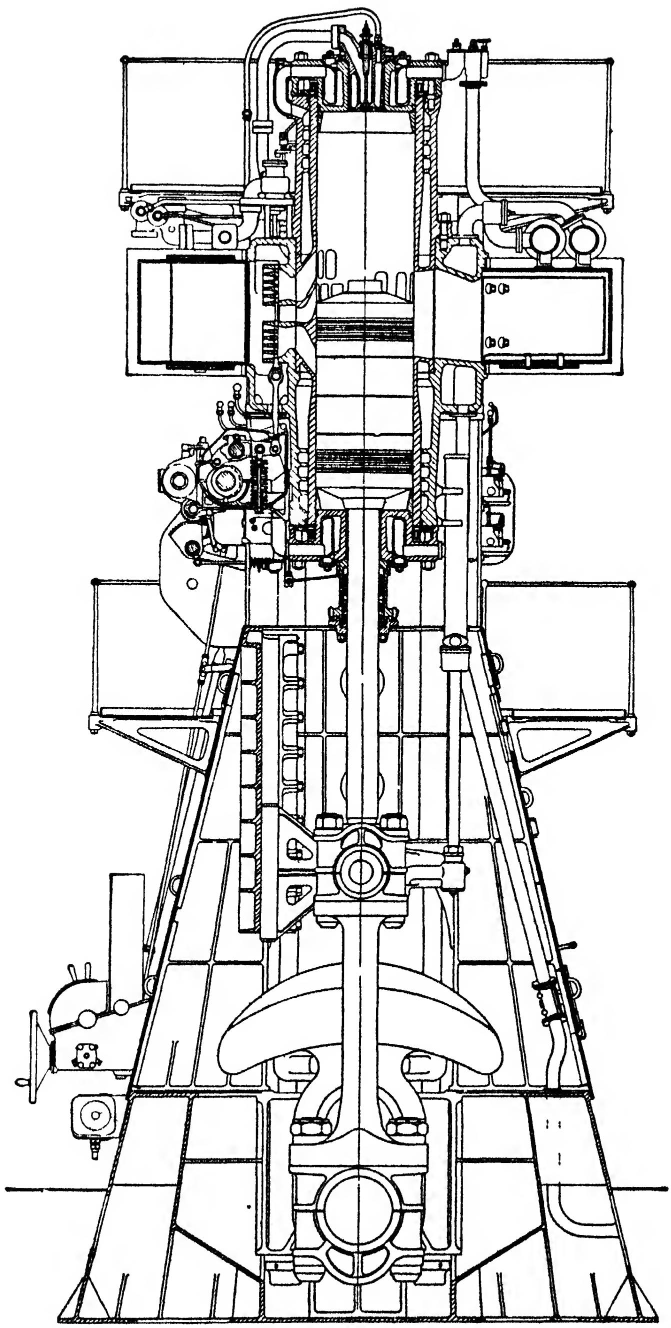


Fig 104 —Transverse Section through a Cylinder of the latest R.-W Engine, showing such original features as the supercharging valves, linear construction, combustion chamber, insulation, &c.

slide valves operated by a cam, a supercharging effect in addition to the swirling movement being thus produced.

The engine has four cylinders $27\frac{1}{2}$ in. diameter with a piston stroke of $47\frac{1}{4}$ in. The rated service output is 4000 b.h.p. at 110 r.p.m. This gives a brake m.e.p. of about 65 lb. sq. in., which corresponds to a mean indicated pressure of about 76 lb. sq. in. The engine is well within its capacity at the rating mentioned and runs quite sweetly at 120 r.p.m., when 4400 b.h.p. is developed. Actually on trial no less than 5350 i.h.p. was developed at 117·5 r.p.m. The scavenge pressure with this output of nearly 1350 i.h.p. per cylinder was no higher than 2·1 lb. sq. in., while the exhaust temperature was 650° F.

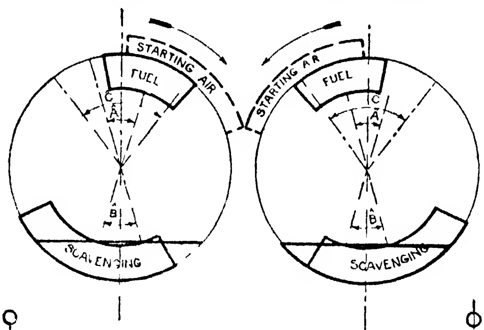
FIG 105.—The upper of these two models shows how the latest arrangement of alternate ports gives a compact lay-out with, at the same time, the greatest possible total circumferential port width. The lower model shows a section through the water space of an exhaust port, and also shows lower cylinder scavenge ports and upper cylinder supercharging ports.

# CHAPTER IX

## Reversing of Diesel Engines

For marine work, the main engine is preferably made reversible, except in small sizes, such as up to 200 h.p. per shaft, where reversing gears external to the engine are often used. By this arrangement these small engines run in one direction for ahead and astern motion of the vessel, thus keeping the

Diagram of Valve Settings for Ahead and Astern Running of early type of Reversible Two-cycle Engine



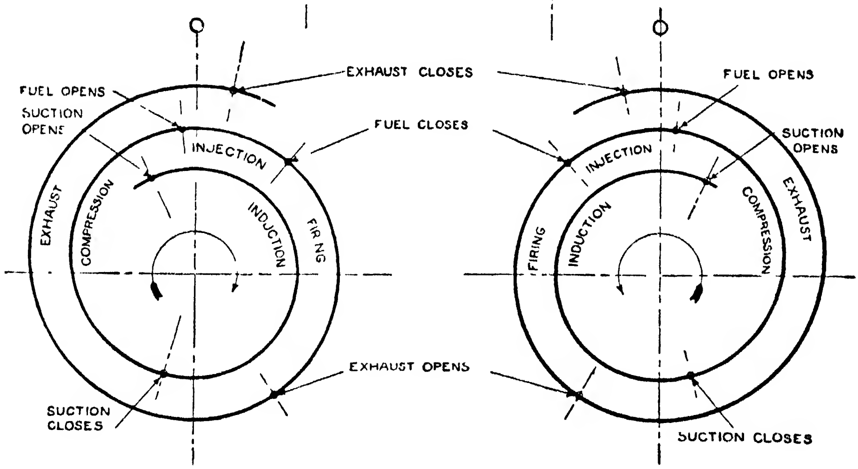




Diagram of Valve Settings for Ahead and Astern Running of Typical Reversible Four-cycle Engine

Fig. 106

valve gear of the most simple type. This subject is one to which a very considerable amount of study has been given, and many and ingenious are the various methods of reversing Diesel engines. The problem is very much less complicated than it would appear at first sight. Advantages of easy reversibility were attributed to earlier two-cycle engines, and the valve-timing diagram given herewith (fig. 106) illustrates this point, where it will be seen that the scavenging and the fuel-injection phases are so arranged to be symmetrical about an axis of 20° from the vertical. In this way a slight alteration to the angular position of the cam-shaft relative to the crank-shaft would put the valves in the cylinder head for fuel injection and scavenging in such a position that they would operate for the opposite direction of rotation. Separate cams were required for starting ahead and astern. It was found, however, that whilst this was extremely simple, it did not give the best actual running results, since it necessitated certain compromises in the designed setting of the valves.

Two-cycle practice in respect to reversing at the present time differs but little from four-cycle practice, although the valves to be operated in the case of the two-cycle engines are not so heavy nor so numerous as with the

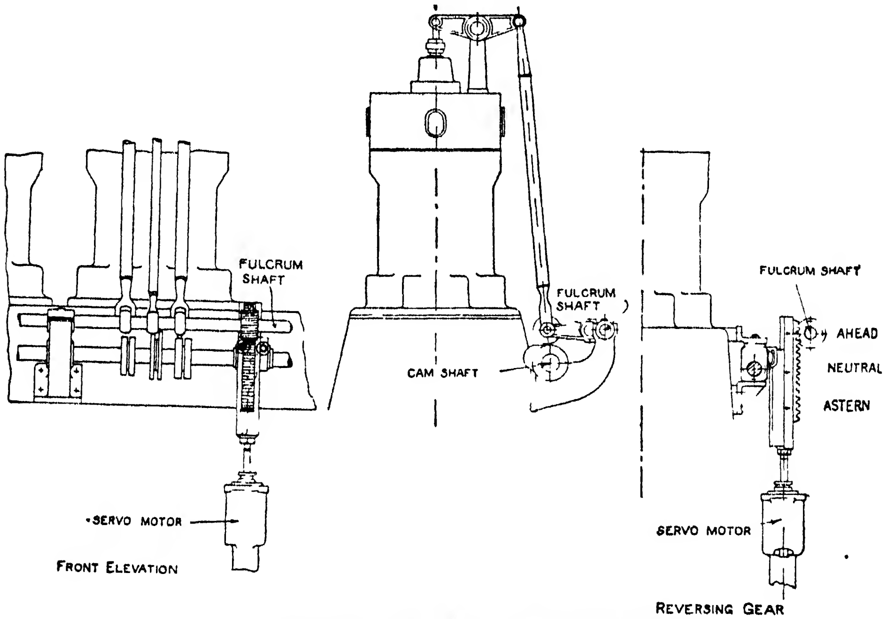



Fig 107—Types of Valve Gear, &c, Burmeister & Wain System

four-cycle, and therefore the gear can be made somewhat lighter If the four-cycle system is described in full, it will serve to show the underlying design.

Generally two sets of cams are provided on the cam-shaft, one cam for ahead running and one for astern running for each valve. The valves are driven by means of a mechanism of levers and rollers for driving from this cam-shaft (see figs. 107, 108, and 109). The rollers, therefore, must be lifted off one set of cams and replaced on the other. This can be done in a number of ways. (*a*) Fig. 107: the rollers are swung clear of the cams, the cam-shaft is shifted longitudinally, and the rollers are replaced on the cams. (*b*) Fig. 108: the valve lever fulcrum shaft, on which the levers are mounted on eccentrics, is rotated, so lifting the rollers from off the cams, and as a result of these levers being mounted skewed on the eccentrics, they move longitudinally, and the roller is replaced on the adjacent cam for the opposite direction of rotation. (*c*) A third method (fig. 109) is to lift the rollers from off the cams, and then to move the rollers only fore and aft, so that on being replaced they fall on to the cam for the opposite direction of rotation. These alterations required for the running in the opposite

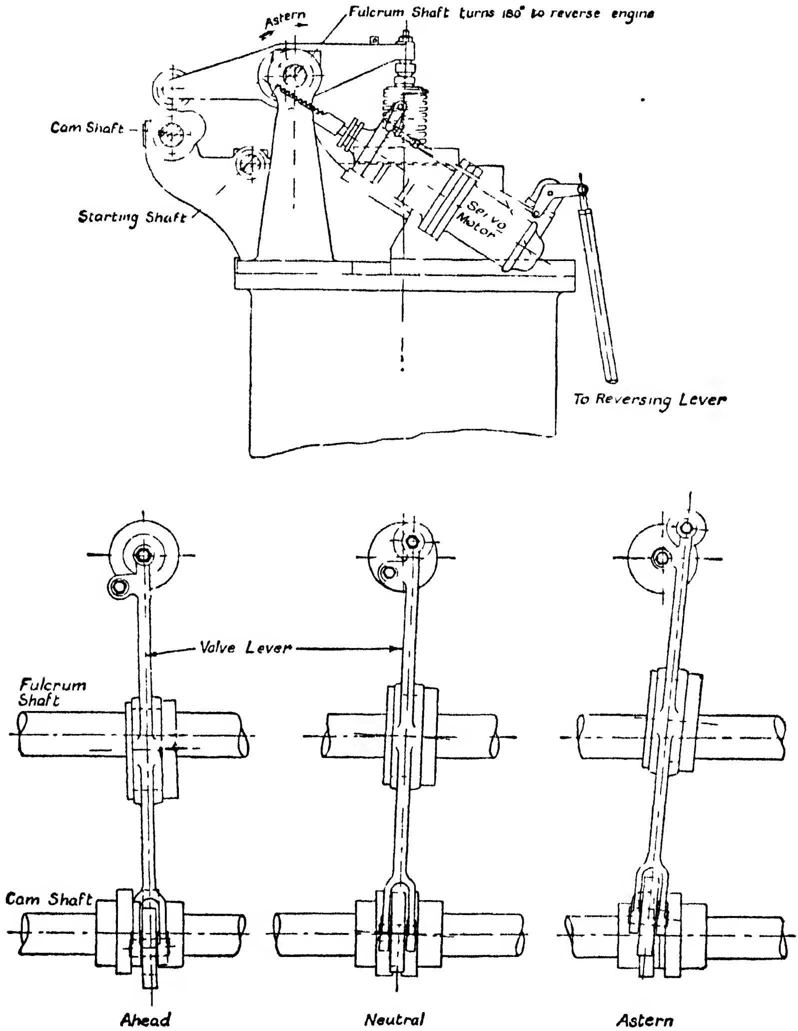



Fig. 108.—Position of Valve Levers for Ahead and Astern Running, Werkspoor System

direction of rotation can all very simply be carried out, and, in fact, are very much simpler than the work necessarily involved in changing a steam-engine with Stephenson link motion from the ahead to the astern position. In Diesel engines they are usually carried out either by (*a*) an air servo-motor of the rotary or plunger type, or (*b*) an electric motor as in fig. 109, and since the electric driving of auxiliaries is becoming a very common feature with Diesel vessels, the latter method is gaining ground.

It will be appreciated that since the fuel-injection air compressor and the fuel-injection pumps both have automatic valves, no gear whatsoever is required to permit these to function equally in either direction of rotation.

With fig. 109 is given a complete description of the Beardmore-Tosi

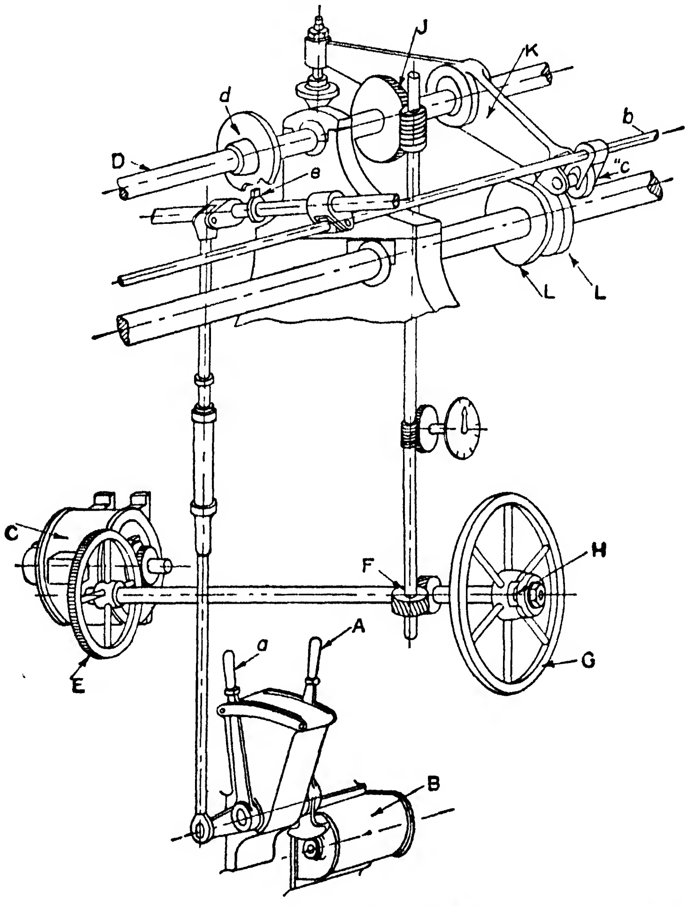



Fig. 109.—Beardmore-Tosi Marine-type Diesel Engine. Diagrammatic Perspective Arrangement of Manœuvring Gear

reversing gear, and generally the same principles apply to all gears, so that a complete understanding of one makes all others relatively simple to follow.

*Starting*.—The *hand lever* A controlling the *starter* B of the 5-h.p. *electric motor* C, driving the *fulcrum shaft* D through the media of *spur gearing* E, *spiral gearing* F, to which *hand wheel* G—the speed of revolution of this shaft being suitable—is clutched for hand starting at H, and *worm gearing* J effects the starting of the engine by causing one half-revolution of the fulcrum shaft D. The valve-operating *levers* K are eccentrically mounted on the fulcrum shaft D, the rotation of which causes the rollers for starting to be depressed on to, or for stopping raised from, the respective *cams* $L_1$ and $L_2$ (one for ahead and one astern). This lowering or raising of the roller end of the valve levers K is carried out in such a sequence as controlled by the angle of the eccentrics to cause the order of starting to be as follows: Stop (all rollers clear of cams), 6 on air (6 air-valve rollers on cams), 3 on air, 3 on fuel (3 air rollers raised and 3 fuel rollers depressed), 6 on fuel (3 remaining air rollers raised and 3 remaining fuel rollers depressed). *Reversing.*—The *hand lever a* through the medium of links and levers causes fore and aft movement of the *reversing shaft b* upon which *forks c* are mounted. Fore and aft movement pulls the rollers along their pins in the wide fork end of the levers K, from opposite the ahead cams such as $L_1$, to opposite the astern cams such as $L_2$. A *disc d* with a gate is keyed on to the fulcrum shaft D, through which *pointer e* must pass before rollers can be moved fore and aft for reversal. The gate in the disc *d* is only opposite the pointer *e* when the valve gear is in the stop position, i.e. all the rollers clear of the cams. This is the only interlockment in the gear.

Fig. 110 shows the starting and reversing gear for the Harland-B. & A. double-acting two-stroke engine. The following description is quoted by permission of The Institute of Marine Engineers from the paper by C. C. Pounder referred to on p. 224.

Fuel cam 1 is keyed to cam-shaft 2, on which there are two clutch toes 3—opposite each other—formed solid with the cam-shaft. Chain-wheel 4, loose on the cam-shaft, carries a lost-motion clutch ring 5, with two toes 6. These toes are of such size that when one side engages cam-shaft toe 3, as shown, fuel cam 1 is in phase with the crank-shaft for ahead running; when chain-wheel 4 and clutch ring 5 are rotated until toes 6 engage the opposite side of toes 3, the fuel cam is in phase for astern running.

To prevent chatter the lost-motion coupling 5 has bolted to it a friction plate arrangement. This comprises casing 7, which carries weight levers 8 and alternate friction plates 9—having projections 10 engaging slots in 7, also plates 11—which have projections 12 engaging slots 13 in 2. The movement of 8 is limited by studs 14, also by springs inside 8 so adjusted that no force is exerted upon the clutch plates when the engine is turned slowly. This allows the chain-wheel to move freely on the cam-shaft, to take up the lost motion when reversing. As the engine gathers speed, the centrifugal force of 8 overcomes the spring on 14 and exerts pressure on adjusting pins 15 which is transmitted through pins 16 to the clutch plates. The lost-motion rotation of 4 carries with it a nut 18 on the chain-wheel sleeve. Rotation of 18 slides threaded sleeve 17 along splined cam-shaft, thereby moving 20, 21 and sleeve 22—which is free on shaft 23—and operating fuel locking cam 19.

The air cylinder, acting through the reverse retaining gear and 24, 23, 25, 26, operates the blower change valves, held firmly against their faces by spring rods, not shown—and the retaining gear provides the reaction.

A slot in plate 27 moves shaft 28, rotating at engine speed, which carries ahead and astern distributor cams. By reason of differential pistons on the distributor valves, these make contact with the cams only so long as starting air is on the distributor, the springs holding off the valves at other times.

Reversing air and brake cylinder piston-rod moves locking bar 29, which has two slots so placed that, when full-up or full-down, one is oppo-

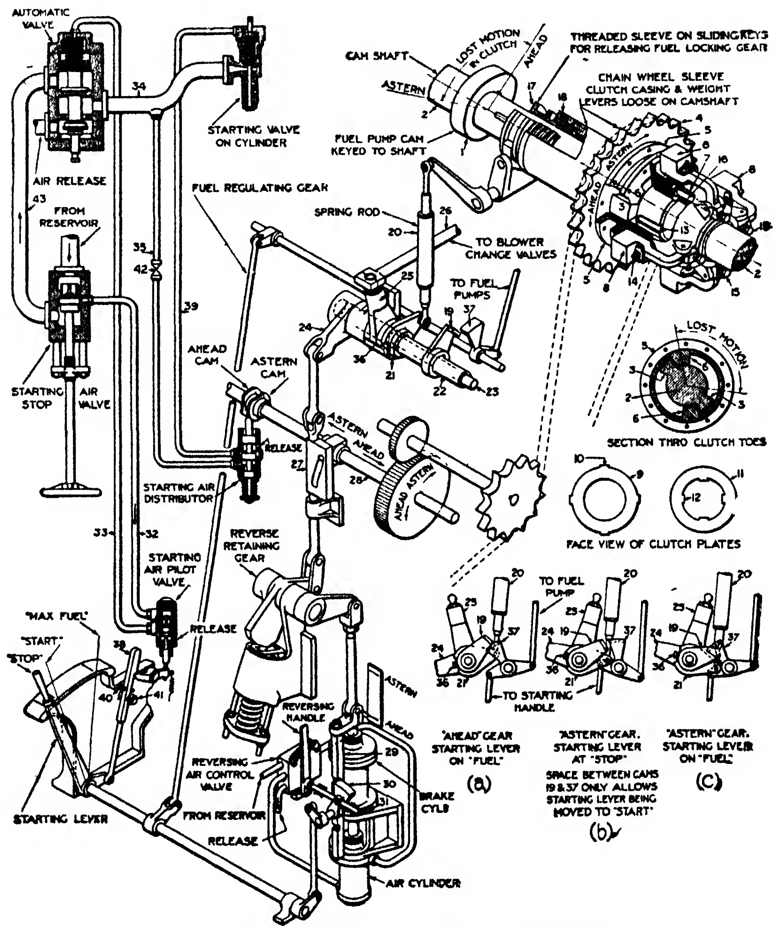



Fig. 110.—Reversing Gear of Double-acting Two-cycle Engine

site to cam 30, which is operated by starting lever. This prevents starting air being put on the engine before the air cylinder piston has completed its stroke. The movement of the starting lever which operates cam 30 also operates pin 31; this locks the reversing handle, except when the starting lever is at stop.

Fig. 110 shows the starting lever at " Stop ", engine in ahead gear, and fuel locking cam 19 as at (*a*). The starting air stop valve is closed. In opening this valve, before pressure can reach the automatic valve by pipe

43, the air passes through 32 to the pilot valve and 33 to the automatic valve top chamber, keeping it closed. Starting air cannot, therefore, reach 34 or 35.

To reverse the engine: bring starting lever to " Stop ", move handle on reversing air control valve to " Astern " and hold it there. This supplies air to the air cylinder bottom. When the pointer comes to " Astern " on the index plate, the reversing handle is moved to the vertical or neutral position; otherwise when attempting to bring starting lever to " Start ", locking pin 31 will come against reversing handle boss instead of entering the hole. The astern cam is now over the air distributor valve, the blower change valves are reversed, and key 36 on lever 25, acting on lever 21, has extended spring rod 20 somewhat, bringing cam 19 into position shown at (*b*).

The space between regulating lever 37 and cam 19 permits the starting lever being moved to " Start ", cam 30 travelling into slot in bar 29, and pin 31 into hole in reversing handle. Draw-bar 38 lifts the air pilot valve spindle, releasing the pressure from the automatic valve top chamber, which automatic valve thereupon opens and starting air enters 34 and 35, forcing the air distributor valves against the cams. The engine crank-shaft begins to move astern; chain-wheel 4 with its clutch gear, all driven from the crank-shaft, takes up the lost motion—about one-third of a revolution—until clutch toes 3 and 6 come into contact for astern running. During this time 18 moves 17 towards 1, relieving the tension in 20 and moving 19 to position shown at (*c*), releasing 37 and so allowing starting lever to be moved past " Start " to " Fuel ". As soon as starting lever is moved past " Start ", pin 40 engages roller 41 and trips the air pilot valve, thus closing the automatic valve and allowing the air in 34 and 35 to escape.

Valve 42 is closed only for testing starting valves.

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# CHAPTER X

## The Internal-combustion Turbine

For high powers the limitations of all reciprocating engines are well known, and were intensified by the success which the steam turbine rapidly achieved, especially in high-powered installations.

A number of attempts were made about 1910, principally on the Continent, to produce an internal-combustion turbine, but these ended in failure, and the project was generally abandoned as impossible.

Since these early attempts there has been a considerable advance in the knowledge of combustion of liquid and gaseous fuels, and in the production of materials suitable for high temperatures. Consequently further experimental work has been undertaken, and it is on record that a 500-h.p. oil-turbine was constructed for the Prussian State Railway in 1920.

In 1922 a 5000-Kw. turbine was under construction, and designs for higher powers under consideration.

These experiments have not yet led to the production of a machine suitable for commercial service.

Internal combustion to be efficient must take place in highly compressed air. Reciprocating air compressors are very much more efficient than rotary machines when the pressure of the delivered air is as required for such combustion. If a rotary compressor is used the efficiency of the whole plant is immediately limited, and if a reciprocating air compressor is fitted the case for the turbine is seriously weakened. Further, the difficulties (owing to the high temperatures of combustion in the turbine) in finding materials which will maintain the requisite strength at the temperatures involved, if not insuperable, are very great. These are the two main factors limiting research and experiment.

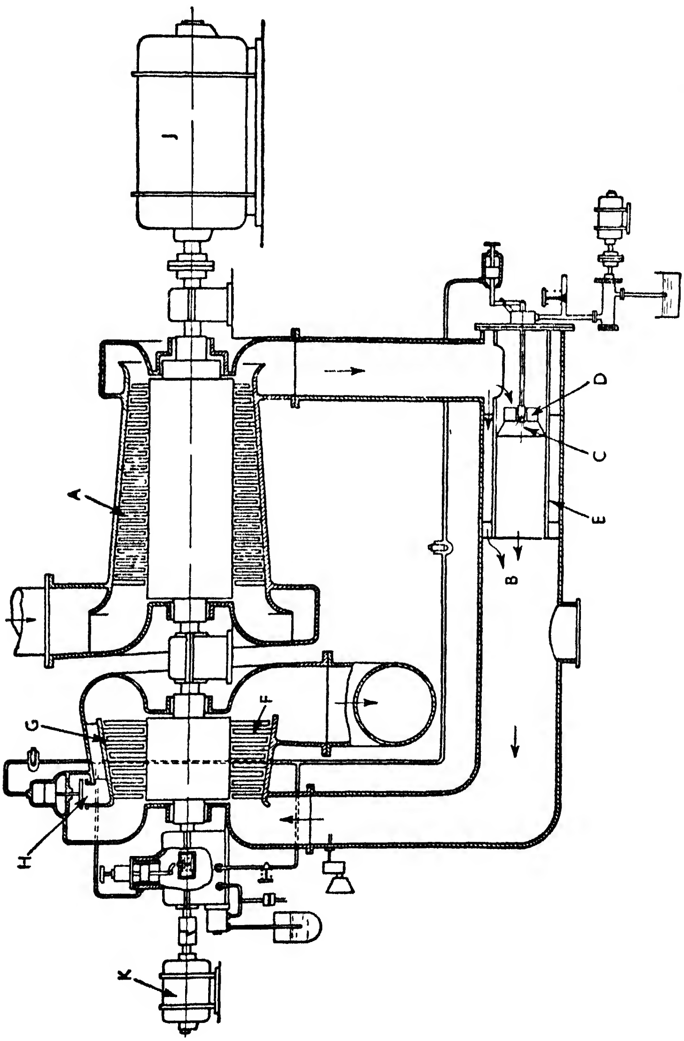
Much progress has been made in the development of the " Combustion " Gas Turbine, which is actuated by the steady flow of the products of continuous combustion under pressure in a combustion chamber. Information on the development, limitations and prospects is given in a paper by Dr. Adolf Meyer (*Proc. I. Mech. E.*, Feb., 1939). Fig. 111, reproduced by permission from this paper, shows the essential features of this type of turbine.

Oil is used as a fuel in this case, and a large excess of air is also used in order to reduce the effect of high temperature of the combustion products on the creep strength of the turbine blading.

Air is taken from the atmosphere and compressed by the axial flow compressor A (fig. 111), the pressure of delivery being 20 to 30 lb. sq. in. by gauge, and is delivered to the combustion chamber B. Part of the air passes through the burner D and is used for the combustion of the oil entering at C, the remainder of the air passing through the annulus B. The gases then pass through the blading F of the turbine G and thence to atmosphere. The turbine drives the compressor A and the generator J either directly or through gearing. Governing is accomplished by controlling the fuel supply and also by the bypass valve H, which also acts as a safety valve. K is a small starting motor, which ensures the compressor supplying the air for starting up and also operates the lubricating and fuel oil pumps.

Dr. Meyer shows that with a temperature of 1000° F. at entry to the turbine and an overall efficiency of 73 to 75 per cent (compressor plus turbine) the thermal efficiency for turbines of 2000 to 8000 Kw. could be expected to attain 17 to 18 per cent. As the efficiency of a modern steam turbine is 25 per cent or more, a gas turbine of this type could not therefore be expected to compete with the steam turbine for power station base loads. It would have, however, advantages as a stand-by plant or for peak loads, as it is of simple design, cheap, light and not dependent on any water supply, so could be installed at any convenient point. Improvements in the cycle could be obtained by utilizing the heat of the exhaust gases to pre-heat the compressed air, giving a possible increase of efficiency from 16·5 to 21 per cent. Other possible arrangements are dealt with by Dr. Meyer.

If materials capable of withstanding continuously higher temperatures than 1000° F. become available at reasonable cost the efficiency can be




*Proc Inst of Mech Engineers*

Fig. 111 —Diagram of the Simplest Form of Combustion Turbine Plant. With reaction type gas turbine and axial compressor for oil fuel, and with excess air cooling

raised still higher, and this type of turbine may then become a serious competitor of the steam turbine.

**The Coal-dust Engine.**—Experimental work on internal-combustion engines using coal dust as fuel has been in progress for many years on the Continent. During a discussion on Coal as Fuel for Internal-Combustion

Engines,* it was stated that the coal-dust engine can now be regarded as being ripe for trial on service. Troubles associated with its development have been ash content of the fuel, excessive wear of cylinder liner and piston rings, erosion and burning at the exhaust valve, and carbonization and erosion of the injection nozzles. Difficulties in cylinder lubrication and sludge removal have also been experienced, but most of the difficulties have been reduced to reasonable proportions by research and experiment.

The fuel is introduced into the combustion chamber in two stages. First, the dust stream is pneumatically regulated during the suction stroke through a dust valve and is carried by an air stream into a combustion chamber which acts as a kind of weir. During the following compression stroke the dust is prepared for combustion by the inrush of compressed air from the cylinder until at the upper dead centre or thereabout a sudden rise of pressure in the pre-combustion chamber is caused by a partial ignition. This forces the contents of the pre-combustion chamber into the combustion space, at first quickly and then at a decreasing rate. For high efficiency it is essential that the dust should pass as soon as possible into the cylinder after ignition in the pre-combustion chamber and that combustion should be completed as soon as possible after the commencement of the stroke.

Much information on constructional details of coal-dust engines, accompanied by indicator diagrams and particulars of tests, are recorded in the report of the discussion referred to above. Research on the coal-dust engine is now in progress in this country, and the results of further tests will be awaited with interest.

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# CHAPTER XI

# Concluding Notes

In the preceding chapters the progress of the internal-combustion engine has been traced, necessarily somewhat briefly, from the appearance of the first small single-cylindered single-acting Otto-Crossley engines of 1876, to the great multi-cylindered heavy-oil marine engines of the present day with an output of fully 1000 h.p. per cylinder, employed in propelling ocean-going vessels of the largest class in installations aggregating in some cases upwards of 35,000 i.h.p.

The very wide range of application of the internal-combustion engine to the propulsion of watercraft has also been indicated; at one end is the smallest type of outboard motor pleasure boat accommodating two persons and driven by a 2- or 3-h.p. petrol-engine of the simplest type; at the

* *Proc. I. Mech. E.*, Vol. 141, No. 4, 1939.

other, the fast great liner whose *auxiliary* engines alone develop thousands of horse-power. Between these extremes are found the launches, small harbour craft, yachts, flying-boats, ferries, shallow-draught vessels of all kinds, tugs, canal barges, submarines, dredgers, coasters, trawlers, patrol boats, ocean " tramps ", tankers, &c.

The problems of chief importance that remain have reference to the best direction of development of the great engines required for propelling ships of the largest class. The intense and ever-increasing world competition in overseas transport necessitates continuous effort to obtain more and more power with less and less bulk and weight of engines, and several ways of effecting progress in this direction are indicated.

Thus, as in large vessels propelled by single or twin screws the propellers necessitate a low revolution speed, increased output may be sought by increasing cylinder dimensions and piston strokes. In the early days of marine Diesels many cases occurred of installations of large-diametered cylinders before the necessary preliminary experience had been gained of their reliability in continuous service, and many costly and occasionally disastrous failures resulted from cracked and seized pistons, cracked liners, burst covers, and other breakdowns. The formidable character of the cooling problems, and the great difficulty of preventing over-stressing of cylinders and pistons through unequal heating, was only fully realized after many unfortunate experiences.

Substantial progress has, however, been made, as is evident on comparing the single-acting four-stroke engines of the 1912 *Selandia*, with 20·875-in. bore × 28·75-in. stroke, with an output per cylinder of 146 b.h.p. at 129 r.p.m., with the 1929 *Rangitiki* engines of single-acting two-stroke type, having cylinders 35·4-in. bore × 63-in. stroke, and capable of an output of 1000 b.h.p. per cylinder at 90 r.p.m.

It is significant that in 1928 Messrs. Sulzer Bros., after their very great experience with the two-stroke heavy-oil engine, ventured upon the construction of an experimental double-acting two-stroke Diesel of 35·4-in. bore and 55·2-in. stroke, developing 2000 b.h.p. from its single cylinder, at 100 r.p.m. An external view of this interesting engine appears in fig. 112; its size may be estimated by comparison with that of the attendants shown on the three platforms.

Though indicated mean effective pressures of 120 lb. per square inch, or even more, can readily be obtained, it is customary in regular service to work with from about 80 to 100 lb. per square inch in order to prevent any risk of overheating troubles arising.

The stroke of the largest slow-speed marine engine in some cases exceeds 60 in., and piston speeds of rather over 900 ft. per minute are successfully employed, as e.g. in the case of the *Rangitiki*, referred to in Chapter VIII (p. 241).

**Revolution Speed.**—A second means of increasing power output is by increasing revolution speed. With large single or twin screws, and no reducing gear, the engines must run at from 80 to about 140 r.p.m. If reducing gear be installed, the engine speed may be increased. Instances of vessels fitted with reducing gear have been given in preceding chapters.

The main difficulties encountered in running Diesel engines fast have

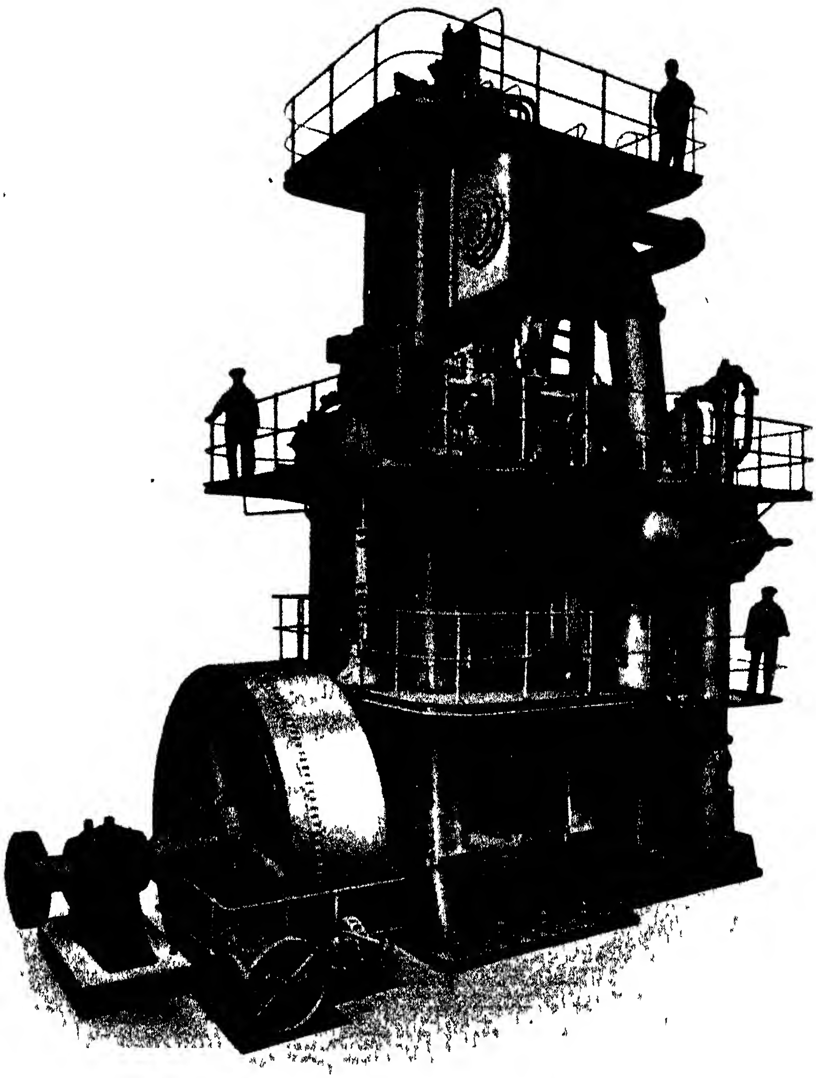

Fig. 112.—Sulzer Double-acting Diesel Engine—Experimental 2000 b.h p. in One Cylinder at 100 r.p.m.

been: (1) that of obtaining sufficiently complete combustion of the oil fuel in the very short interval of time available, and (2) that of keeping bearing pressures down to a sufficiently low intensity.

Fast-running Diesel engines driving rail cars have proved very successful, and are now quite firmly established; it is possible the confidence thus created may become manifested in marine practice in the future. Messrs. Beardmore, for example, have specialized in the production of this type, and in 1928 two of their high-speed Diesels were installed in a locomotive. Each engine comprised twelve cylinders of 12-in. bore × 12-in. stroke, with an output of 1330 h.p., and the two were jointly capable of hauling a load of 2800 tons at 40 m. per hour. Again, early in 1929, Vickers-Armstrong were developing designs of six-cylindered four-stroke Diesels of 14-in. bore by 14·8-in. stroke, developing 750 b.h.p. at 450 r.p.m. The Ricardo-Brotherhood 300-b.h.p. sleeve-valve Diesel running at 900 r.p.m. has already been described.

In the 16,000-ton 16-knot *St. Louis*, built in Germany in 1929 for the North Atlantic service, four Diesel engines, aggregating 12,000 h.p. and running at 215 r.p.m., drive two propeller shafts, through reducing gearing, at 110 r.p.m.

Late in 1928 also was completed the tanker *Brunswick*, 12,500 tons, the first British-built large ship to be equipped with Diesel-electric transmission. In this vessel four high-speed Diesel engines, direct-coupled to generators, supply current to the propelling motors.

In addition to the matter of deciding upon the type of propelling engine in marine services, on which point various comments will be found in preceding chapters, much attention has in recent years been devoted to the supercharging of engines as a means of increasing power output. As already described, Sulzer Bros. have for long employed a special valve-controlled third belt of scavenge ports whereby a slight supercharge is obtained (fig. 8, p. 142), and by this means claim to obtain an output more than double that of a similar four-stroke cycle cylinder. Turbo-driven blowers are very commonly employed with large two-stroke engines. But since 1928 notable progress has been made with the Buchi system of turbo-supercharging large four-stroke engines. In this system the exhaust gases from the main engines actuate a small high-speed turbine direct-coupled to a two-stage blower by which cool air under small pressure is delivered to the engine cylinders.

"The influence of exhaust-turbine supercharging can be shown clearly at ¾ load by running the engine with and without the turbine. This can be done by by-passing the turbine. With the engine tested at a constant load of approximately 850 b.h.p., the difference was 137° F. When turbine-supercharged, the heat transfer per brake horse-power per hour is also greatly reduced. This means that in spite of the 50 per cent increase in power output due to efficient scavenging, the heat lost to the cooling water is approximately the same as that for an ordinary Diesel engine giving only two-thirds of the power output. This point is important, as it gives a measure of reliability, large engines requiring heavy wall thicknesses, which cause a big temperature drop and correspondingly high stresses in the cylinder walls." *

* Buchi, World Power Conference on Fuel, 1928.

Tests carried out by Dr. Stodola at Winterthur in 1928 showed that by aid of a Buchi supercharger the normal output of an ordinary four-stroke engine could be increased by 50 per cent, and that the maximum power output could be even increased by 100 per cent, with no increase either in the temperature of combustion or in that of the exhaust gases, and with reduced fuel consumption per horse-power hour. Though these large increases in output are of great interest and value, they have not, so far, been taken advantage of to their full extent in marine installations.

The first British ship (fig. 68, p. 230) fitted with the Buchi system was the *Raby Castle*, equipped with an eight-cylindered single-acting four-stroke North-Eastern-Werkspoor Diesel engine giving normally 3000 i.h.p. at 92 r.p.m.; during her three years' continuous service prior to the installation of the Buchi supercharger in 1928 the vessel had averaged roundly 11 knots in deep load condition. Though the additional apparatus increased the total displacement of the vessel by 300 tons, the speed on trial was found to have been increased by 12 per cent, viz. from 11·9 knots (test) to 13·339 knots, while the maximum pressure during combustion was found to have remained practically unchanged. It was estimated that by aid of the Buchi system the power output of the engine was increased by 30 per cent; the Buchi supercharger ran at 4000 r.p.m.

The Buchi system has undoubtedly given a fillip to the large four-stroke marine engine. On the other hand, after many years' experience of the four-stroke design, the great firm of Burmeister & Wain decided in 1928 to build large two-stroke marine engines. Although it is quite impossible to venture any prediction as to the type of Diesel marine engine which will ultimately prevail, it is abundantly evident that immense progress has been made within the past ten years with both the four-stroke and the two-stroke type.

Difficulties with vibration have been met with in some cases. Vibration may be caused by (1) unbalanced secondary forces and those of still higher frequency due to reciprocating parts, (2) torsional vibrations of the crank-shaft and attachments. If any part of the ship has a natural frequency of vibration corresponding to one of these, resonance will occur and this part of the ship will be uncomfortable for passengers. Much has been done in modern vessels, however, to insulate these portions, and serious trouble is rarely experienced where sufficient care has been taken with the design of both ship and propelling machinery. In some recent vessels the engines are insulated from their foundations by rubber chocks.

In his Howard lectures before the Royal Society of Arts in 1931, Ricardo dealt very fully with recent developments in Diesel engines, and particularly expressed the opinion that the large marine Diesel engines would ultimately give place to smaller units. The following are his remarks on this subject:

" We are now, I believe, on the eve of seeing the light high-speed Diesel turned out in bulk production like the petrol engine, and it behoves us, I think, to review and revise our ideas as to how it is to be treated. In the first place, we must keep it always before our minds that the efficiency of the Diesel engine is absolutely independent of size. In the past the Diesel engine's chief competitor was the steam engine, and the steam engine is efficient only in very large units. From long acquaintance with the steam engine, we have grown accustomed, therefore, to think in terms of few and large units; we have struggled hard and, to my mind, quite unnecessarily to produce very large Diesel engines, and in doing so we have run into all manner of troubles which might easily have been avoided. For a given aggregate horse-power we are bound, on the score of efficiency, to employ the largest possible units when steam is the motive power, but where Diesel engines are used we should, I contend, employ the largest possible *number* of units each of the smallest possible size. By doing so we shall, in fact, gain in efficiency, since the units in operation can be run always at their most economical load factor. We shall gain enormously in first cost; we shall gain enormously both in weight and in space occupied; and, above all, we shall gain, hands down, in reliability. I have never been able to discover whether the aversion to a large multiplicity of units is due to steam tradition or to mere fear of the unusual. As a rule, the arguments advanced against it appear to me to lack weight. One is asked to view with horror the vision of a power plant containing, say, 1000 pistons and 2000 valves, all of which have to be maintained in working order, and to turn with comfort and assurance to, say, a single turbine doing the same work. At first glance the comparison may seem appalling, but let us consider it a little more closely. Of the 1000 pistons we can afford to allow anything up to, say, 100 or 150 to go wrong even simultaneously and still be able to carry the full load, for the remaining units will have at least a 10 to 15 per cent temporary overload capacity. We could afford to crash even 200 or 300 units at the same instant without any serious inconvenience. Now let us consider the single turbine. Inside its simple casing, unseen but not forgotten, are several thousand blades, the failure of any single one of which will bring the entire plant to a complete standstill. The chance of a turbine blade coming adrift is, happily, fairly remote, but the chance of a thousand Diesel engines all breaking down at the same moment is almost beyond the bounds of possibility. One is told, again, that the engineer in charge of such a Diesel plant would be so worn out with anxiety that he would never sleep a wink. One of the largest and probably quite the most important power plant in the world is the four hundred thousand horse plant which is responsible for the above-ground passenger traffic of London. A failure of this plant would certainly cause annoyance and dislocation to more people and to more business interests than of any other I can conceive. This plant consists of over 5000 engines with 30,000 pistons and 60,000 valves, yet the engineer in charge of the London General Omnibus Company is anything but a nervous wreck, and, I am told, sleeps like a child.

"It is my confirmed belief that the Diesel engine of the future will be a small high-speed unit of a size which can be turned out very cheaply by bulk production methods, and that where large power concentrations are required we shall employ large batteries of such units. We must, however, revise also our ideas as to how such engines are to be handled. The large slow-running engine is far from self-supporting—it requires sympathetic care and constant attendance; the high-speed engine requires none of these things; in fact, there is nothing whatever that the most conscientious or sympathetic attendant can do to minister to its needs. So long as it has oil and fuel it will run, for its lubrication is entirely automatic, and whether tended with skill or neglected, it will continue to run for a similar period until wear or carbonization brings that period to a close. Experience will soon show just how long it is economical to allow any particular make of engine to run between overhauls.

" When the time is ripe for overhaul the engine will be removed bodily and replaced by a reconditioned unit. There should be no question of repairs *in situ*; if an engine is out of order or has run its allotted span, it should be removed and replaced by another. In the case of light high-speed units this can be done by a couple of fitters in an hour or so, and the decarbonizing or reconditioning of the weary engine can be carried out in comfort and at leisure in a properly appointed hospital. In hospital the engine will be subjected to one or other of two treatments: either what, in aircraft parlance, is termed a top overhaul—that is, decarbonizing, cleaning the piston ring grooves and possibly replacing any faulty rings and grinding or adjusting the valves and injectors; or to a general overhaul, involving probably the fitting of new liners, bearings and other wearing parts. The former, in the case of a 100–150 h.p. engine, can be accomplished by two mechanics in one day; the latter may involve a week's work, while the cost of replace parts may amount to 10 per cent of the first cost of the engine. It is early days yet to say how frequently such overhauls will be required, but, speaking from my own experience to-day, with two high-speed sleeve-valve Diesel engines, one of 100 h.p. and the other of 300 h.p., during three years of strenuous service I find that in the present state of the art it pays to give a top overhaul—that is, one day off—every 1200 hours, and the indications are that a general overhaul will probably be desirable after 9000 hours. Taking the average service of an engine as 8 hours per day, this means a top overhaul—that is, one day off—once every six months, and a general overhaul—say, 10 days off—every three or four years. As technique and experience develop, these periods will gradually be extended. In the case of aircraft engines, for example, the untouched running period is now just three times as long as it was ten years ago, despite reduction in weight and improved performance gained during this period.

" So rapid has been the development of the light high-speed Diesel engine during the last few years that it is now competing in that most exacting and difficult of all services, the public service road vehicle. Here it is attacking the petrol engine in its securest stronghold. In this vast field it has to compete with probably the most highly developed and mechanically perfect prime mover in existence. In its competition with petrol, the Diesel engine has one trump card, its much lower fuel consumption; and, at the moment, this card has an exaggerated value because of the tax on its rival's fuel. This latter is a temporary advantage only, for it is not to be hoped that the Chancellor of the Exchequer will for long allow himself to be cheated of his revenue. Apart from the tax, the Diesel engine uses a cheaper fuel, an advantage it will probably retain for several years to come. In the long run, however, the difference in the cost of fuel will diminish almost to the vanishing point, and the Diesel engine will have to compete ultimately on its lower fuel consumption alone.

" On full load the Diesel engine can show a gain in thermal efficiency of about 25 to 30 per cent over that of a good modern petrol engine."

Any suggestion of Mr. Ricardo demands serious consideration, but it is unlikely that any step will be taken to implement this proposal unless problems of capital cost and space occupied by this type of installation can be solved to the satisfaction of the shipbuilders and shipowners.

Motor-ships completed in 1937, having total horse-powers of over 10,000, were the *Boissevain* (Sulzer), 10,800 h.p.; *Circassia* (Doxford), 11,400 h.p.; *Port Jackson* (Doxford), 10,200 h.p.; *Sussex* (Doxford), 13,200 h.p.; and *Prins Albert* (Sulzer), 15,000 h.p.

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# THE OPERATION OF OIL-ENGINES

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# The Operation of Oil-engines

The compression-ignition oil-engine, with airless injection, using "heavy" fuel oil, is now universal. "Hot-bulb" engines and engines using air-blast injection are rarely manufactured, as the modern cold-starting airless injection type is quite reliable in all sizes, having the advantages of absence of air-compressor for fuel injection and comparatively simple combustion chamber.

The main features of construction and operation of the starting and reversing gears for large marine engines are dealt with in Chapter IX of the section on Oil-engines, and need not be discussed here, and our consideration of starting-up difficulties will mainly apply to slow-speed engines of up to 50 b.h.p. per cylinder, the majority of these being of the horizontal type.

Matters with which the operator will be concerned and in which trouble may be experienced are:—

1. Starting.
2. Fuel injection nozzles.
3. Governing.
4. Lubrication.
5. Pistons, including piston rings.
6. Cooling.

Starting is usually effected by compressed air at about 250 lb. per sq. in. A small compressor is used to charge a reservoir, and it is necessary to see that this reservoir is properly charged previous to starting. In some cases this compressor is driven off the engine and charging takes place while the engine is running, but this has the disadvantage that a few false starts may reduce the pressure in the reservoir to such an extent that it is insufficient for its purpose. A separate compressor, which is highly desirable, may be belt driven when shafting is available or driven by an electric motor or small petrol engine.

A special cam-operated starting valve is fitted to the cylinder and the engine is barred round to a position just over the dead centre on the normal *expansion* stroke. The fuel injection pump is put out of action by a lever which is provided. The air stop valve on the reservoir is now opened, and when the engine is running at a sufficiently high speed the air cam is put out of action and the fuel pump is put into action. If all is well the engine should fire and continue to run.

If the engine is in good condition and ignition does not occur, then the trouble must be due to either pump or spraying nozzle. If there is an air lock in the pump chamber, then the plunger will be compressing air and the delivery valve either will not open or will deliver too small a charge. Testing with the hand lever will usually reveal the existence of an air lock, and a few strokes through the overflow valve will cure this. Obviously this test should be made *before* attempting to start the engine.

If the engine has previously been running for some time, it is possible that one or more of the fine holes in the nozzle may be choked with grit or carbon deposit. Removal of the nozzle from the cylinder and a test when connected up to the pump will reveal trouble of this kind. In some cases the nozzle valve or needle may be sticking in its guide and an unsatisfactory spray will result.

In some engines a " spill-valve " is fitted, actuated by the governor through a pivoted lever, and careful adjustment of the fulcrum is necessary, otherwise too much or too little of the fuel will be by-passed. In the first case either the engine will not start or it will " hunt " when running. In the second case combustion will be incomplete and black smoke will appear in the exhaust. It is as well to fit a small cock in the exhaust pipe near to the engine and to test the exhaust gas by impingement on a sheet of white paper, when the presence of unconsumed carbon in the exhaust can be detected and remedied.

" Diesel knock " is a phenomenon associated with compression-ignition engines. If a large portion of the fuel vaporizes in the cylinder before ignition commences, then the whole of this mixture of oil vapour and air explodes, causing a sudden rise of pressure instead of a more gradual rise due to steady burning as the fuel enters. The magnitude of the effect depends upon the boiling-point of the fuel, the amount of injection advance and the degree of turbulence in the cylinder. It sometimes occurs when the fuel is changed, and in this case a reduction of the injection advance and, where possible, some increase of the compression will at least reduce the intensity of the knock. Experimenting with different forms of spray also is desirable, but usually outside the scope of the operator.

**Lubrication.**—The object of lubrication is to maintain a film of lubricant between all rubbing surfaces. The maintenance of this film depends upon the viscosity or " body " of the oil, the speed of rubbing and the intensity of the load. It is easier to maintain this film when the surfaces are always moving in the same direction relatively to each other, e.g. a journal and its bearing, than when reciprocation occurs, e.g. trunk piston in cylinder, gudgeon pin, crosshead.

The viscosity of a lubricant decreases rapidly as the temperature rises, and a lubricant which is suitable for moderate temperatures may be quite unsuitable for comparatively high temperatures on this and other accounts.

Some lubricants have a greater tendency to " gum " owing to oxidation than others, and this takes place more rapidly at high temperatures. In the case of a trunk piston, contamination by gases which blow past the rings tends to reduce the lubricating properties of oils.

Another important property of a lubricant is what is known as " oiliness ". If a film fails and the surfaces come together, considerable heat will be generated and " seizing " will eventually occur. With certain kinds of oils, however, an exceedingly thin film sticks to (or is " adsorbed " by) each of the surfaces, preventing actual contact, and this property is extremely valuable where the lubrication is likely to be imperfect. Mineral oils have very little " oiliness ", but vegetable oils, such as rape- and castor-oil, possess this property in a high degree, so that if a little vegetable oil is blended with a mineral oil its properties as a so-called " boundary " lubricant are very much enhanced.

Generally speaking, when a liberal supply of oil is conveyed to the bearings by means of forced lubrication, a straight mineral oil of the required viscosity will be satisfactory. With drip lubrication and with reciprocating surfaces a blended oil will probably give better results.

The above short statement by no means exhausts the subject of lubrication, but the operator should know that a good oil for one purpose may be altogether unsatisfactory under other conditions. The suitability of an oil for a particular purpose can only be assessed by actual trial under working conditions, and mere appearance, colour or feel are very unsafe guides.

The efficiency and costs of running and overhauls of an engine depend largely upon satisfactory lubrication, and operators are recommended to obtain their lubricants from manufacturers of repute who, when a particular brand has been found to give satisfactory results, can guarantee consistency in quality of future supplies.

The lubrication of the piston in the case of a trunk-piston engine is of very great importance, since in this case the greater part of the friction of the engine is piston friction. The author's experience is that a considerable increase in friction horse-power may result from inadequate lubrication of the piston. A blended oil which is not subject to decomposition is the best to use in this case, and in the case of horizontal engines inspection of the piston during running gives quite a good idea of the adequacy and suitability of the lubrication. If there is any evidence of blow-by, the piston should be withdrawn at the first convenient opportunity and the rings examined for undue wear or sticking in the grooves.

In the case of an enclosed vertical engine the effect of blow-by and of carbonized oil from the inside surface of a trunk piston will be to contaminate the oil in the sump, and filtering the oil and passing it through a centrifugal separator at frequent intervals are desirable.

In large vertical engines oil is delivered under pressure to the main bearings and through the hollow crank-shaft to the crank-pin bearing. From this point it finds its way up the hollow connecting-rod to the top-end bearing, and also in crosshead engines to the guides.

The oil, after passing through the bearings, falls into the crank-case and drains to the pump suction well from which it is drawn, and after passing through a strainer and cooler is again passed through the system. Sufficient pressure is required on the system to ensure that the oil reaches the highest bearing. Oil thrown from the top and bottom end bearings in the trunk piston engine may find its way to the cylinder walls and by excess lubrication of the walls cause gumming of the piston rings. This objectionable feature is overcome to some extent by fitting baffle plates to prevent the oil thrown from the bearings reaching the cylinder walls.

In the case of the open-type engine the main bearings and crank-pins are usually lubricated by the "ring" and "drip-fed" system. The usual practice is to adopt the ordinary loose-ring lubrication for the main bearings and special drip-fed lubricators for the crank-pin. It follows that with this system the lubricating oil required may be reduced to a minimum, and little oil will be thrown on cylinder walls, the latter fact assisting in limiting carbonization of the piston rings. It must be remembered that with this system there is the danger of the attendant over-reducing the oil supply or even allowing the oil boxes to run dry, with obvious results. This, however, should not occur if care is taken.

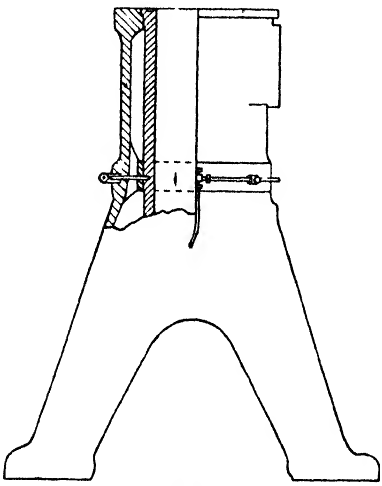

Fig. 1

The pistons and gudgeon-pins are lubricated by means of small plunger pumps which give a regular supply. The piston lubrication oil is forced from the pump through four or more points round the circumference of the cylinder (see fig. 1). These holes are not always evenly spaced on the circumference, but are often nearer together at the back and front of engine to compensate for the thrust due to obliquity of the connecting-rod.

For the gudgeon-pin, lubricating oil is forced through a hole in the cylinder, this hole registering with a groove on the piston located just above the gudgeon-pin. A hole is drilled from this groove to the hollow gudgeon-pin, after which the oil is led via small vertical holes to the small-end bearing.

**Exhaust Valves.**—These valves are arranged in pockets in the cylinder cover, each valve making its seat on a cast-iron cage or casing and so arranged that both cage and valve can be removed together for inspection or renewal without disturbing the cylinder cover.

The valve itself is of mushroom shape and opens against the compression of a strong spring. The conditions under which this valve works are rather severe, as the valve is in direct contact with the gases expelled from the cylinder, these gases being at a temperature of from 800° to 900° F. The valves in the smaller sized engines are made of special heat-resisting steel, and in larger sizes are generally made with a separate head of special cast iron secured to the steel spindle. Cast iron generally gives good service. In the larger sized engines the valve cage is water-cooled, like the cylinder cover.

A section of a typical exhaust valve is shown in fig. 2.

" Pitting " of the valve and seat is due either to the chemical action at high temperatures or to a small percentage of coking element in the fuel, small particles of which get trapped between the valve and seat. The valve and casing should be drawn periodically for regrinding, and this is generally done about every 200 working hours.

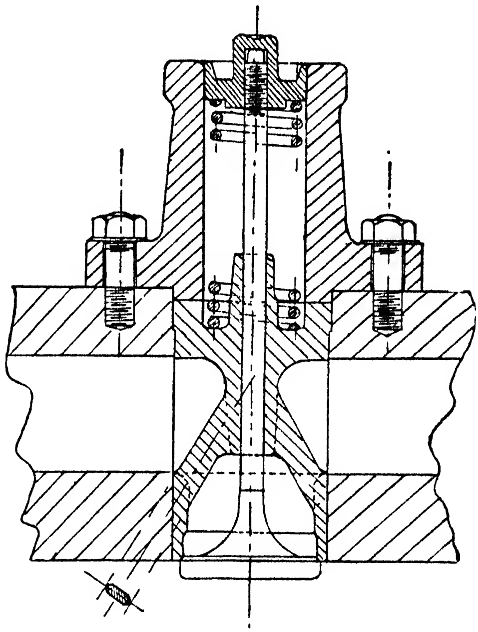

Fig 2

It is recommended that spare sets of exhaust valves should be held in readiness to substitute at short notice, and with this system the valves removed may be reground at leisure.

**Induction Valves.**—These valves are similar in design to the exhaust valve previously described, and are sometimes interchangeable. The severity of the working conditions is not so marked as with the exhaust valves, because the induction or suction valves, as they are sometimes called, are cooled by the incoming air. Under normal working conditions they should be changed after about 1500 hours' running, but in view of the possibility of distortion of the valve head, with consequent leakage at the seat, periodical examination is essential.

**Fuel-injection Valves.**—It is beyond the scope of this article to give a description of the various types of fuel valves in use, but the function of this valve is to admit fuel at, and through, a determined interval of the stroke to the combustion chamber in as finely divided a state as possible. Communication is established between the valve and cylinder by the motion of a cam operating a fulcrum lever, this lever raising the valve from its seat at the required moment. Atomization is effected in the conventional fuel valve by means of a ring pulverizer, fluted cone, and flame plate.

The following points of operation are essential to achieve smooth working conditions. The valve must be kept tight and safely withstand a pressure of upwards of 1000 lb. per square inch. It must not "stick" open; this may possibly happen if the spindle gland is screwed down too tightly; in fact, it is more safe to allow a slight leakage of compressed air at the spindle gland and feel satisfied that the spindle is free to move. The average pulverizer is prone to choke, but this may be minimized by thoroughly straining the fuel before it reaches the engine. A typical fuel-injection valve for air-injection engines is illustrated in fig. 3.

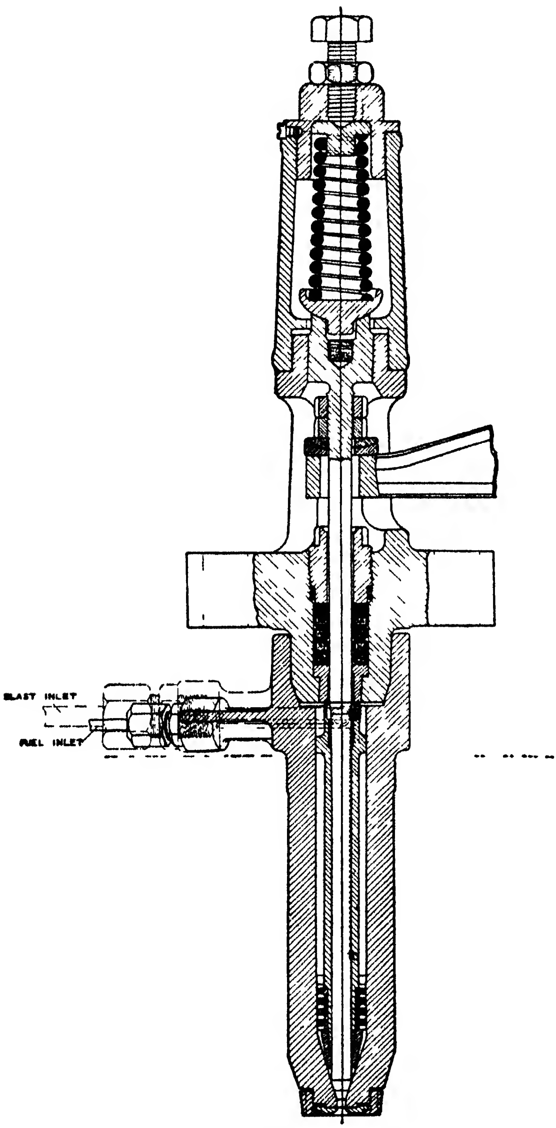

Fig 3

For airless injection, the fuel valve is usually of the automatic type, i.e. a spring-loaded valve opened by the pressure of the fuel oil supplied by the fuel pump. The fuel pump in this case is arranged to measure the quantity and time the supply of the fuel to the fuel valve. The oil pressure in the fuel system with airless injection varies from 4000 lb. to 10,000 lb. per square inch, and in consequence of these high pressures it is essential that the fuel pump and fuel valve be kept in first-class condition if satisfactory operation of the engine is to be ensured.

**Air Starting Valves.**—These valves are operated in a manner similar to that for the fuel valve, that is, by a cam and fulcrum lever. The upper portion of the valve is of plunger form; it is so made in order to balance the air pressure, which tends to open the valve, thus permitting the adoption of a weaker spring. The best arrangement is that where an external spring is provided, as in this design the spring is free from contact with the moisture which is present in the compressed air at starting. If, when starting up, it is found that one of the air starting valves is stuck open, or leaking badly, the stop valve on the starting bottle should be closed at once, otherwise the engine will surge to and fro with a consequent highly dangerous rise in compression. Fig. 4 illustrates the type of valve actuating gear usually adopted for 4-cycle vertical engines. With this arrangement it will be seen that the air starting valve spring is arranged externally.

**Valve Settings, &c.**—To ensure reliable working the engineer in charge of an oil-engine should maintain the valve settings to a close approximation of those used by the engine makers. Little alteration can be made to the settings of the exhaust, suction, and scavenge valves, as the cams operating these are keyed to the camshaft. It often happens that these valves are reseated and reground, thus altering slightly the length of spindle. It follows then that, after assembling, care should be taken to readjust the clearance between the cam and its roller, this being done by an adjustable pin situated usually on the valve end of the fulcrum lever.

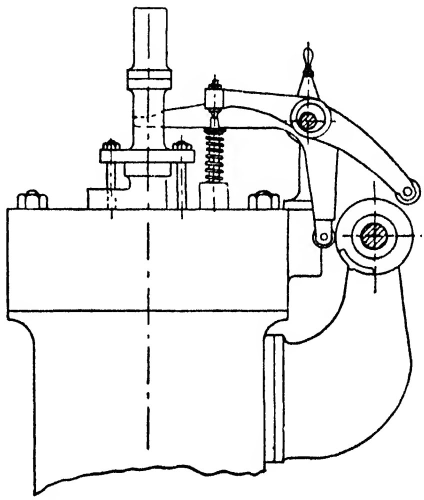

Fig. 4

In the case of the fuel-injection valve, the moment of opening may be altered by adjusting a detachable case-hardened "toe piece" secured by set-screws to the cam proper. This toe piece has slotted holes machined in and is capable of moving circumferentially to the correct position, this being determined from the indicator diagram.

This does not apply to airless injection engines where the fuel valve is of the automatic type.

**Fuel Pumps and Valves.**—The function of the fuel pump is to deliver a small quantity of oil to the fuel-injection valve, the quantity delivered being dependent on the load and controlled by governor action. The two types of pump for multi-cylinder engines, as met with in everyday practice, are the single-plunger type and the multi-plunger type, both of which possess certain advantages.

In the former type, now seldom used, a common plunger supplies fuel to all cylinders through branch pipes, equal distribution being maintained by a fitting termed the distributor. It is claimed that with this system improved governing is effected. It would seem, moreover, that inaccuracies in manufacture should be reduced to a minimum, as this pump when compared with the multi-plunger type is proportionally larger. With the multi-plunger system either a separate fuel pump is provided for each cylinder or the plungers and valves are incorporated in one casting. This system has the advantage that either cylinder may be cut out of action in the event of an accident. It is not recommended, however, that a cylinder should be cut out from the standpoint of fuel economy only, as this would produce an uneven turning moment.

The nature of the attention required depends upon the type of fuel pump used. In the Bosch type of pump, for instance, port suction is used, and a delivery valve only is required. In other types both suction and delivery valves are fitted, and as the operating pressure varies from 2000 to 10,000 lb. per sq. in., according to the size of engine, careful attention

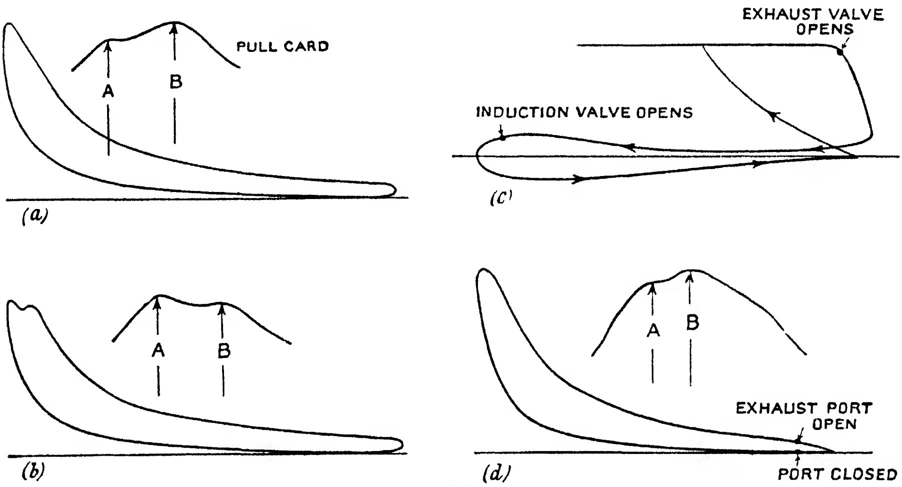


Fig. 5

must be paid to the seatings of these valves if satisfactory fuel injection is to be obtained. Oil strainers should be cleaned periodically in order to ensure that access of oil to the suction valves is not impeded.

**Indicator Cards.**—Much information may be obtained from indicator cards, and it is recommended that cards be taken periodically. There are now indicators manufactured which are specially suited for Diesel engine use, such as the Dobbie McInnes indicator and the Maihak indicator.

With these instruments, cards can be taken showing the pressure variation throughout the piston stroke.

Fig. 5 (*a*) shows a card from a four-stroke cycle engine. This may be considered a good card, timing and pressures being satisfactory.

Fig. 5 (*b*) shows a card where late firing is taking place, and indicates that improvement could be made by advancing the injection of the fuel.

It is usual to take what is termed a "pull" card, i.e. a card where the movement of the paper drum is made by hand at the moment of firing in lieu of the usual movement from the indicator mechanism. This type of card shows very clearly the compression and firing pressure in the cylinder, and is very useful when adjusting fuel-valve timing. The pull card is shown in figs. 5 (*a*) and (*b*).

Light spring cards may also be obtained by fitting the indicator with a light spring, say of 12 to 16 lb. per square inch. This gives a card which shows clearly the exhaust and induction strokes in the four-stroke cycle engine. This type of card is shown in fig. 5 (*c*).

On these cards the points of fuel injection, opening of exhaust and induction valves are indicated.

In the case of two-stroke cycle engines, the exhaust and air charging take place at the end of the stroke and are indicated in fig. 5 (*d*).

**Cooling.**—From 30 to 40 per cent of the heat of the fuel usually has to be carried away by the jacket-cooling water. For instance, with an engine of 40 b.h.p., using 0·4 lb. of fuel oil per b.h.p. hour, calorific value 19,000 B.Th.U. per lb., allowing for a rise of 50° F. the amount of cooling water to be circulated would be about 200 gallons per hour. Even a comparatively small quantity such as this, if taken from town's mains, would add appreciably to the annual cost of running, and in practice cooling tanks and in some cases cooling towers would be fitted in order to conserve water.

Obviously the quantity of cooling water required to be circulated for a given engine will depend upon the permissible rise of temperature, and this in turn will influence the cost of external cooling devices and circulating pumps.

It must be remembered that the heat received by the cooling water must all be disposed of before it re-enters the jacket. If a tank only is used the temperature will rise until the heat radiated per minute from the external surface of the tank is equal to the heat received per minute from the jacket. If the surface of the tank or tanks is insufficient to prevent excessive temperature rise, additional cooling devices must be installed.

A high jacket temperature is not objectionable; in fact it will tend to reduce piston friction by reducing oil viscosity, so long as steam locks are not formed anywhere in the jacket. Steam is an extremely poor heat conductor, and rapid overheating will occur wherever it is allowed to form. For this reason it is usually inadvisable to allow the exit temperature to exceed 160° F.

On board ship, the engines may require to run continuously for long periods, and it is essential that careful attention be given to the following:

1. Keep fuel service tank filled regularly and fuel oil strainers periodically cleaned.

2. Keep forced lubrication supply at the required steady pressure and working temperature. Investigate any undue rise in oil temperature and locate cause.

3. Examine cylinder lubrication and keep mechanical feed boxes supplied as required.

4. Examine exhaust gas discharge, which should be clear; if smoky, locate defective cylinder and find cause. Any abnormal rise in exhaust gas

temperature at cylinder outlet generally points to defective combustion in that cylinder, and should be rectified as early as possible.

5. Piston cooling, whether by oil or water, requires careful attention as to temperature and quantity of cooling medium supplied to pistons.

6. Careful observation should be regularly made of the cooling-water system, the temperature of the inlet water and discharge water from each individual cylinder, cylinder head and exhaust valve being noted. Any marked rise above the normal working temperature should be immediately investigated and the cause found.

7. All pumps supplying lubricating oil, jacket-cooling water, piston cooling, &c., should be adjusted to give a steady supply.

# OIL-ENGINES FOR ROAD VEHICLES

BY

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# Oil-engines for Road Vehicles

## CHAPTER I

## General Considerations

The success of the compression-ignition engine for land installations and marine propulsion as an economical and reliable prime mover using a fuel which, compared with motor spirit, is relatively cheap, encouraged experimental work with the object of evolving a high-speed oil-engine which could compete successfully with the petrol-engine for propulsion of motor vehicles. Its comparative simplicity due to the absence of electrical ignition devices also made a strong appeal.

The problems to be solved, however, were of a more complex nature than in the case of large engines running at more or less constant slow speed, and it has taken years of intensive research and experiment to produce reasonably satisfactory solutions of these problems. Very high revolution speeds are not yet commercially practicable, and for this reason among others applications have mainly been confined to commercial vehicles. Experiments on oil-engines running at up to 4000 r.p.m. have, however, been carried out by H. R. Ricardo and show that there is no inherent speed limitation with this type of engine.

As a preliminary to the consideration of engine types and details a summary of the general advantages and disadvantages of the compression-ignition engine as compared with the petrol-engine for road vehicles will help towards an understanding of the different lines of development adopted by various manufacturers.

**Advantages.**

1. Lower weight of fuel consumed. A well-tuned petrol-engine consumes about 0·5 lb. of fuel per b.h.p. hour, and in general this amount is exceeded. An oil-engine consumes 0·4 lb. of fuel per b.h.p. hour or less, and this amount is rarely exceeded.

2. Still lower volume of fuel consumed, owing to the higher density of fuel oil. This brings the fuel consumption in *gallons* per hour for a given b.h.p. down to 60–70 per cent of that of the petrol-engine.

3. Lower cost of fuel. The difference in cost has been reduced appre-

ciably by the tax on fuel oil, but in spite of this the cost of motor spirit is about 50 per cent higher than that of fuel oil.

4. Safer fuel, owing to the higher flash-point of fuel oil, with consequent reduced risk of fire.

5. More complete combustion of the fuel, as shown by almost complete absence of carbon monoxide in the exhaust.

6. Much greater range of mean effective pressure and consequent greater flexibility.

The limiting mixture range for petrol is from 20 per cent weak to 40 per cent rich, the use of the richer mixtures leading to considerable decrease in economy. At full throttle the smallest m.e.p. is never less than 70 per cent of the maximum. Throttling increases the pumping losses and so diminishes the economy. On the other hand, the compression-ignition engine will work satisfactorily between brake mean effective pressures of 35 and 115 per sq. in.

7. The rate of flame propagation is low for weak petrol mixtures. This is not the case with compression-ignition engines.

8. High thermal efficiencies at all loads. The curve connecting fuel consumption per b.h.p. with load is much flatter than with petrol-engines.

9. In compression-ignition engines the accelerator pedal regulates the amount of fuel injected only and the engine works at full throttle at all times (actually a throttle is not fitted). Hence the pumping losses are less and the volumetric efficiencies are high at all loads.

10. Better fuel distribution with multi-cylinder engines, since each cylinder is supplied with fuel independently. A larger induction pipe is possible, since it has to do with air only.

**Disadvantages.**

1. Higher maximum pressures. These involve heavier working parts and lead to the greater possibility of bearing troubles.

2. The ratio of maximum pressure to mean pressure is higher, and this again leads to greater weight per horse-power than for petrol-engines. Very substantial reduction in weight has, however, been made in recent years by, for instance, the use of aluminium alloys for cylinder blocks.

3. The maximum possible m.e.p. for a petrol-engine is considerably higher than for a compression-ignition engine, where it rarely exceeds 100 lb. per sq. in. This involves larger cylinders for a given mean torque.

4. Greater initial cost of engine per horse-power. At present the cost of the oil-engine is about double that of the petrol-engine, but the ratio is decreasing as the output increases. The increased initial cost must, of course, be balanced against the reduced running costs.

Two main conditions for satisfactory working are: (1) accurate metering of the fuel supply to the cylinder, and (2) rapid combustion of the fuel. At a speed of 3000 r.p.m. the time for one stroke is 0·01 sec., and if combustion is to be completed during a crank movement of 45° the available time is only $\frac{1}{400}$ sec. This involves rapid and complete mixing of the fuel with the air and a considerable degree of turbulence. Indiscriminate turbulence is, however, undesirable, as in this case much of the fuel would be deposited on the cylinder walls.

For rapid combustion the fuel and air must be given a high velocity relative to each other. The earlier alternative, still in use for large slow-speed engines, was to make the fuel spray fit the combustion chamber as far as possible and to secure adequate penetration by using high-injection pressures (up to 10,000 lb. per sq. in.). In the small high-speed engine the air itself is given a vigorous whirling motion either in the combustion chamber or in an auxiliary chamber into which the fuel is injected. The method of " air swirl " has the advantage that the velocity of swirl increases as the speed of the engine increases, and this is necessary for speeding up combustion. The fuel should be injected *across* the air-stream if satisfactory splitting up, or " atomization ", is to be effected.

Mr. C. B. Dicksee (*Proc. I. A. E.*, 1932) describes the process of combustion as follows:

1. " Atomized " fuel injected into heated air begins to absorb heat.
2. As soon as the initial boiling-point is reached vapour begins to form.
3. If the compression temperature is sufficiently above the ignition point of the vapour, the vapour will ignite as it is formed, and combustion proceeds steadily from the surface of the drops.
4. If the compression temperature is insufficient to cause immediate ignition of the vapour, this will accumulate until ignition finally takes place (ignition lag). Combustion will then be explosive and will then proceed at a much greater rate. Combustion will commence spontaneously throughout the whole mass and will be uncontrollable. The result is " Diesel knock " if an appreciable quantity has been vaporized.

With petrol-engines, " detonation " takes place in the *last* part of the mixture to be consumed. With compression-ignition engines, the knock is produced by the *first* part to be consumed, and a larger quantity is usually involved.

To avoid the knock it is obviously desirable to reduce the ignition lag, since the longer the delay the greater the quantity of fuel injected into the cylinder before ignition takes place and the greater the chance of vaporization taking place.

The conditions governing ignition lag are: (1) the compression temperature, (2) the boiling-point and ignition temperature of the fuel, (3) the amount of injection advance, (4) the amount of turbulence, (5) the type of combustion chamber, (6) the spray form.

**Types of Combustion Chamber.**—The type of combustion chamber used and the way in which the fuel is injected into this chamber are the important factors governing the satisfactory working of the engine. An engine for a commercial vehicle ordinarily is required to work at speeds ranging from 600 to 1800 r.p.m., to give a satisfactory torque at any speed within this range, and to operate with reasonable economy at all speeds and loads. Since the torque produced depends upon the amount of fuel injected per cycle, the rapid increase of torque required for acceleration of the vehicle can only be obtained by a corresponding increase in the quantity of fuel injected into the combustion chamber, and if under these circumstances there is insufficient air swirl incomplete combustion and the emission of unconsumed carbon from the exhaust will result.

Types of combustion chamber may roughly be classified as:

1. Direct injection with piston swirl, in some cases aided by partially shrouded inlet valves, as in the Gardner engine.

2. Direct injection with port swirl, as in the Ricardo sleeve-valve engine.

3. The use of a chamber on the piston, closed by a plate with a central orifice, into which the fuel is injected (Acro system), as in the Berliet engine.

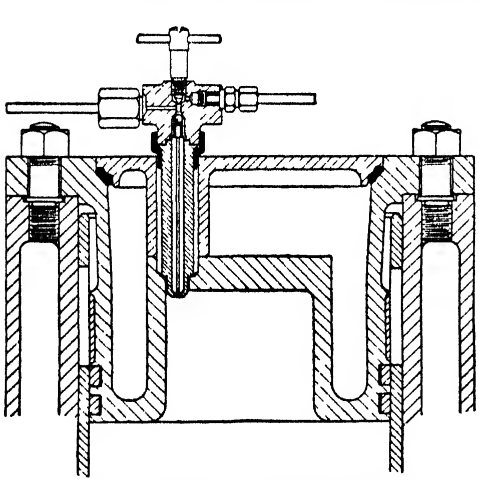

Fig. 1.—" Vortex " Combustion Chamber

4. The use of a chamber in the cylinder head, into which fuel is injected, mixing with air which has been set into violent whirling motion due to rapid compression into the chamber, by the motion of the piston. Partial combustion takes place in this chamber and is completed as the piston moves downward and the mixture enters the cylinder. There are a number of variations in shape and arrangement, but the most successful of these appears to be the Ricardo " Comet " head, which is fitted to the A.E.C., Crossley, Dorman, Paxman and Thornycroft engines.

The *shape* of the spray is important in most cases, particularly with the direct injection type, and the best shape can only be settled by experiment. It should be borne in mind that the shape and amount of penetration of the spray depends upon the pressure of air in the chamber, so that a test of a nozzle in the open air gives little or no guide to the form of the spray under working conditions.

For further information on the effect of air swirl on combustion efficiency a paper on " Air Swirl in Oil-Engines ", by J. F. Alcock (*Proc. I. Mech. E.*, December, 1934), should be consulted.

Fig. 1 shows the " Vortex " combustion chamber of the Ricardo sleeve-valve engine. Air entering through ports in the sleeve is given a rotary motion parallel to the axis of the cylinder. At the end of compression this air is forced into the cylindrical chamber in the cylinder head, the rate of swirl (r.p.m.) being increased accordingly. The fuel nozzle is set near to the circumference of the chamber and the fuel is pulverized very effectively as it leaves the nozzle.

Fig. 2 shows the chief features of the Ricardo " Comet " type of combustion chamber, which is spherical in shape. Practically the whole of the air is forced at high speed into the spherical chamber at the end of the compression stroke, and an intense whirling motion is thus set up. The oil spray is injected radially and effective combustion takes place at all loads and speeds.

A special feature of the Comet head is the heat insulation of the lower portion of the chamber, which has a very important effect in assisting combustion at high loads and speeds. The author has seen a Mirrlees-

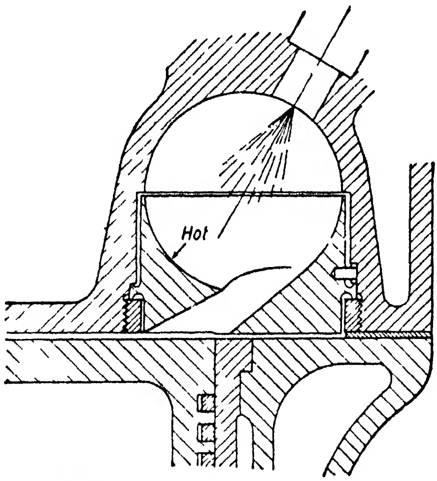


Fig 2—" Comet " Combustion Chamber

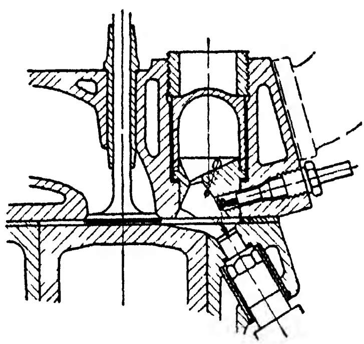

Fig 3—" Acro " Type

Ricardo engine which had been idling accelerated very rapidly to a high speed with full load without any appreciable colouring of the exhaust.

Some modifications of this type are discussed and illustrated in a paper on " Compression Swirl Oil Engines ", by H. S. Glyde and E. N. Soar (*Proc. I. A. E.*, Vol. XXXI, 1936).

Fig. 3, from a paper by H. R. Ricardo (*Proc. I. A. E.*, Vol. XXVIII, 1933), shows a modification of the " Acro " type of combustion chamber, in which the chamber is placed in the cylinder head instead of in the piston. Partial combustion of the fuel spray causes a rapid rise of pressure in the chamber, and the air rushing out past the spray completes the combustion as the piston moves downward.

# CHAPTER II

## Fuel Injection Systems

Three factors are concerned in the fuel injection system, viz., pump, piping and injection nozzle. In an ideal system the pressure would be applied instantaneously, maintained at a steady pressure until the end of injection, and released instantaneously at cut-off. Although these ideal conditions are unattainable in practice, the system which approaches most nearly to them is likely to give the best results. Uniformity of delivery to all of the cylinders is also of the utmost importance.

Since the pump plunger cannot be started instantaneously, injection should not commence until some portion of the stroke is completed and should finish in all cases while the plunger is moving at an appreciable speed. In the majority of injection systems used for high-speed engines the commencement and finish of injection are governed by the pump plunger closing or opening ports in the pump chamber. In the Beardmore system a " flash-valve " is used which operates in a similar way.

**Nozzle Flow.**—The theoretical velocity of flow through an orifice is given by:

$$v(\text{f.p. sec.}) = 8\sqrt{\frac{P}{w}}, \qquad \ldots\ldots\ldots\ldots\ldots\ldots (1)$$

where $P$ = *difference* of pressure producing flow (lb. sq. ft.),

$w$ = wt. of 1 c. ft. of the fluid (lb.).

$\therefore$ If $A$ = area of orifices (total) in sq. in.,

$c$ = coefficient of discharge,

$$\text{Flow per sec. (lb.)} = c \times 8\sqrt{\frac{P}{w}} \times \frac{A}{144} \times w,$$

$$= \frac{c}{18} \cdot A \cdot \sqrt{Pw}.$$

If $p$ is in lb. sq. in. and $s$ is the specific gravity of the fluid,

$$\text{Wt. of fuel (lb. per sec.)} = 5{\cdot}26cA\sqrt{ps}. \qquad \ldots\ldots\ldots\ldots (2)$$

If $t$ = time of injection (sec.),

$$\text{Wt. of fuel per injection} = 5{\cdot}26cA\sqrt{ps}\,t. \qquad \ldots\ldots\ldots\ldots (3)$$

If $N$ = revs. per min.,

$$\text{Wt. of fuel injected per min. (four-stroke)} = 2{\cdot}63\,cA\sqrt{ps}\,Nt. \quad (4)$$

If $\theta$ = period of ignition in degrees of crank angle,

$$\text{then } t = \frac{60}{N} \times \frac{\theta}{360} = \frac{\theta}{6N} \text{ sec.}$$

$$\therefore \text{ Wt. of fuel injected per min.} = 0{\cdot}44cA\sqrt{ps}\,\theta; \quad \ldots\ldots(5)$$

and from (3), $$\text{Wt. of fuel per injection} = 0{\cdot}88cA\sqrt{ps}\,\frac{\theta}{N}. \quad \ldots\ldots\ldots(6)$$

Thus with a given angular period of injection the weight of fuel injected per minute is *independent of the speed of revolution* if the injection pressure $p$ is constant.

The weight of fuel *per injection* is also *inversely proportional to the speed of revolution* if $\theta$ and $p$ are constant. To maintain the mean effective pressure it is therefore necessary to increase the angular period of injection as the speed increases.

It must be borne in mind that in the above investigation the pressure is assumed to be constant throughout and independent of engine speed, and these conditions are not complied with in practice. Drs. S. J. Davies and E. Giffen (*Proc. I. A. E.*, Vols. 25, 27 and 28) have made elaborate experimental investigations of this problem, and show that pressure waves of varying intensity are set up in the connecting pipe from the pump to the injection nozzle, the nature of these depending on the length and size of the pipe, the speed of revolution and the contour of the cam actuating the pump plunger. One obvious conclusion is that if the injection characteristic for each cylinder is to be the same, the length of piping from pump to injection nozzle must be the same for all of the cylinders. Drs. Giffen and Rowe (*Proc. I. Mech. E.*, March, 1939) describe a method of calculating the variations of pressure in an injection system under given conditions. A later paper by Dr. S. J. Davies (*Proc. I. Mech. E.*, October, 1939) on Recent Developments in High-speed Oil-engines discusses the injection system very fully, including the factors involved in atomization of the fuel and ignition lag.

**Fuel Pump.**—The regular and satisfactory operation of the engine depends mainly upon the satisfactory functioning of the fuel pump. An engine developing 10 b.h.p. per cylinder at 1800 r.p.m. requires about 45 cubic millimetres of fuel per cycle injected into each cylinder. With a plunger diameter of 5 mm. (area 19·63 sq. mm.) the maximum effective stroke would thus be about 2·3 mm. If the injection period is 45° of crank angle, the time occupied is $\frac{1}{240}$ sec., so that the average speed of the pump plunger during injection must be 55·2 cm. per sec. (1·8 ft. per sec.).

If injection is to be satisfactory it must commence at full pressure and speed, and cut-off must be extremely rapid with an equally rapid fall of pressure in the delivery pipe. It is therefore necessary for the plunger to be moving at speed before injection commences, for the point of commencement of injection to be constant however the load varies, and for some simple means of varying the time of opening of the spill port to be provided.

A type of pump which would operate satisfactorily with a comparatively large slow-speed engine would not necessarily do so with a small high-speed engine, although the converse of this is by no means the case. There are a number of proprietary makes of fuel pump in use and giving satisfactory results. These include the C.A.V.-Bosch, Bryce, Scintilla and Simms. The best-known and most generally used type is the C.A.V.-Bosch pump, and it is proposed to describe the construction and working of this in some detail.

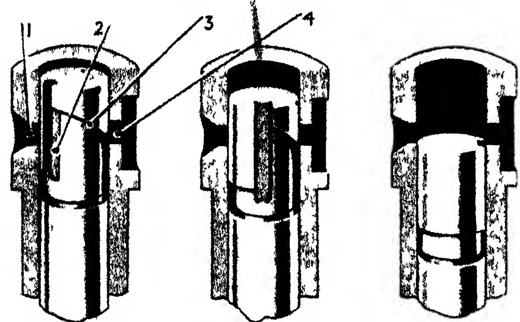

FIG 4.—C A V -Bosch Fuel Injection Pump
1, Inlet port, 2, Spill groove, 3, Helical edge, 4, Spill port

The C.A.V.-Bosch fuel injection pump is a cam-operated spring return plunger pump of the constant-stroke type. Each pump element consists of an accurately ground steel pump barrel into which is fitted an accurately ground steel pump plunger (fig. 4). The delivery valve (fig. 6), fitted at the upper end of the barrel, completes the parts comprising the pumping unit.

Fuel flows from a tank placed slightly higher than the pump to the fuel inlet connection, a suitable filter being fitted between the tank and

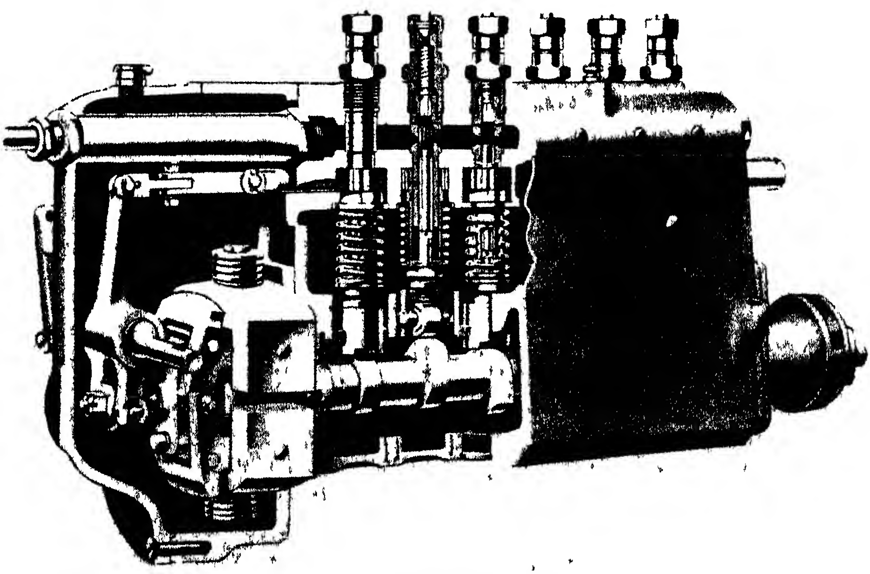
FIG. 5.—Fuel Pump and Governor

the pump, and keeps the common suction chamber in the pump casing full of clean fuel which can pass through the port 1 (fig. 4) into the upper part of the pump barrel. To enable the pump to vary the quantity of fuel delivered per stroke, the plunger is provided with a vertical groove 2 extending from its top edge to an annular groove (the upper edge 3 of which is formed as a helix) a little distance down the plunger length. There is therefore free communication between the upper chamber and the annular groove, but there is no external communication unless either the inlet 1 or the outlet 2 is open (unless the plunger is rotated to the " Stop " position, when the vertical groove permits free communication between the pump chamber and the outlet port 3). Partial rotation of the plunger is effected by means of a rack (fig. 5) operated by the accelerator pedal or the governor, as explained later, and this alters the point in the pump stroke at which pressure release occurs owing to the helical edge 3 overrunning the port 4 (fig. 4), thus enabling the enclosed fuel to escape back through this port to the common suction chamber via the vertical groove 2.

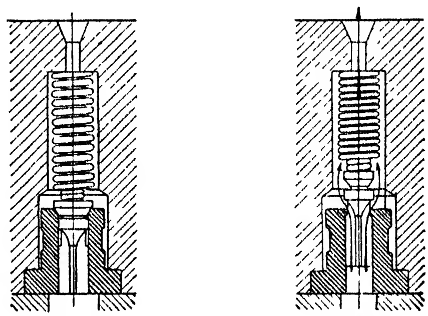

Fig. 6.—Delivery Valve

The delivery valve (shown in detail in fig. 6) acts as an effective " anti-dribble " device. Immediately the pump plunger releases the pressure in the barrel this valve, under the influence of its spring and the difference of pressures between the pump barrel and the delivery pipe, returns to its seat, causing the small piston parts of the guide to sweep down the valve seating with plunger action. The effect of the consequent increase of volume in the delivery pipe system is to reduce the pressure of the fuel therein so that the nozzle valve in the nozzle can " snap " to its seat, thus rapidly terminating the spray of fuel entirely without " dribble ".

The governor (figs. 5 and 7) is so arranged that the effective stroke of the pump can be varied by the use of the accelerator pedal, but in the event of the speed increasing beyond a predetermined figure the governor acts independently by reducing the effective stroke. The principle of the governing mechanism is shown in figs. 5 and 8.

Referring to fig. 8, the floating lever AB is pivoted on an eccentric C, so that if A remains stationary a forward movement of the pedal moves the rod BH to the right, thus increasing the effective stroke of the pump plunger. If, however, the speed becomes excessive, the governor weights D move outward and, C being now the pivot, the rod BH is moved back sufficiently to reduce the speed to normal. It will be seen that by this arrangement the free operation of the accelerator pedal is not interfered with by the governor, provided the speed does not exceed a predetermined maximum. Details of construction are shown in figs. 5 and 7.

Referring to fig. 7, each governor weight is supplied with an outer spring 5 to take charge of the idling speed, and the two strong inner springs 6 and 7 for the maximum speed. Whilst idling the spring 5 bears on the weight, but if the engine speed is increased beyond idling by the depression of the accelerator pedal, the increased centrifugal force produced causes the weight to bear against the spring plate 9 supporting the springs 6 and 7. Throughout the normal speed range of the engine the weights remain in

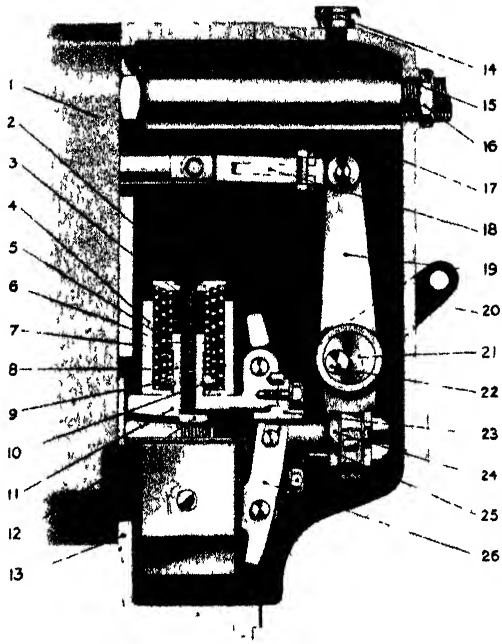


Fig 7 —Section through Governor

1. Fuel pump body
2. Governor spring collar (outer).
3. Screwed stud for governor.
4. Governor weight
5. Governor spring (outer).
6. Governor spring (inner).
7. Governor spring (centre).
8. Sleeve for weight.
9. Governor spring collar (inner).
10. Maximum-speed spring stop.
11. Driving centre
12. Locking nut on cam-shaft.
13. Governor casing (front).
14. Oil filler.
15. Governor casing (rear).
16. Feed pipe lock nut.
17. Feed pipe.
18. Operating lever coupling.
19. Operating lever
20. Control lever
21. Control shaft eccentric.
22. Control shaft.
23. Control shaft bracket
24. Operating shaft collar.
25. Operating shaft.
26. Cranked lever.

this position, the governor remaining inoperative and the engine being controlled only by the accelerator pedal.

Should the pedal be depressed too far, however, so that the pump delivers more fuel than the engine needs for the load, the speed will tend to exceed the prescribed limit, so that the weights then travel farther outward and the control rod operates as previously described. A control rod stop is provided to prevent the effective travel of the plunger becoming excessive during starting.

**Nozzles.**—The type of nozzle required depends upon the position and shape of the combustion chamber and the nature of the turbulence produced before and during injection. Fig. 9 shows the main features of the different types of nozzles used in practice. The " pintle " type is generally used in an air-cell or pre-combustion chamber engine. By vary-

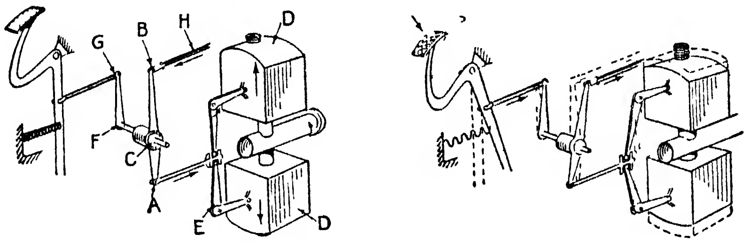


Fig 8 —Principle of the Governor

ing the size and shape of the pintle, cones of spray from 4° upward can be provided.

Certain engines, usually of the pre-combustion chamber type, require nozzles with modified spray characteristics in order that they can produce a stable performance when idling. These results are produced by a

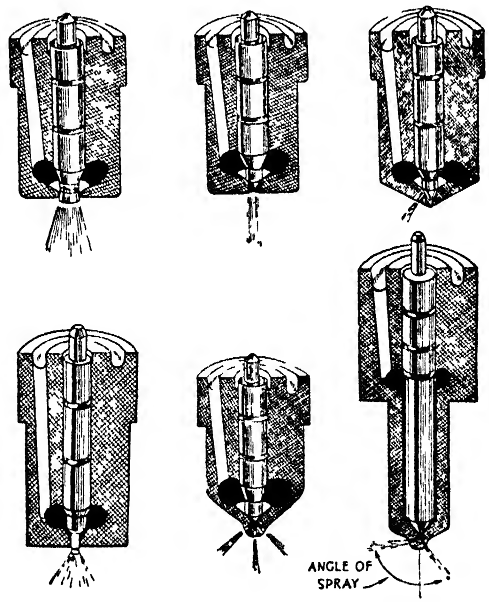


Fig 9.—C A.V.-Bosch Fuel Injection Nozzles

(Top row, left to right) Pintle, single-hole, conical end (single hole). (Bottom row, left to right) Delay, multi-hole, long stem

modification of the pintle by means of which the rate of injection increases toward the end of delivery, the effect of this being to lengthen the periods of injection at idling speeds without affecting combustion at higher speeds. This nozzle is known as a " delay " nozzle. Prolonged

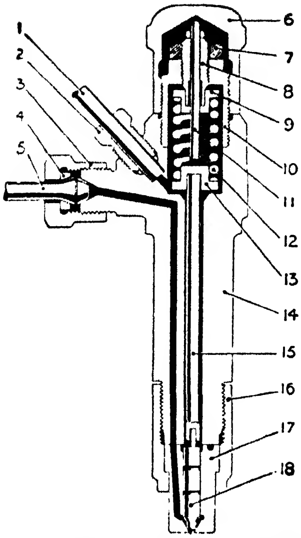


Fig. 10.—Section 7B Ante-chamber Engine

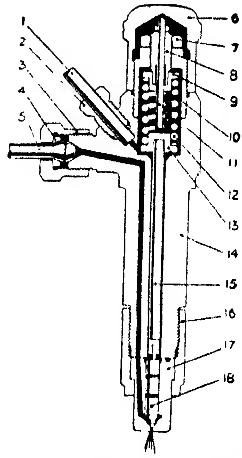


Fig. 11.—Section 7E D.I. Engine

Figs. 10 and 11.—Sections through Nozzle

1. Leak-off pipe.
2. Leak-off union nut.
3. Pressure pipe union nut.
4. Washer for union.
5. Pressure pipe.
6. Top cover.
7. Lock nut for adjusting screw.
8. Spring adjusting screw.
9. Top spring collar.
10. Spring-retaining cap.
11. Feeler pin.
12. Valve return spring.
13. Bottom spring collar.
14. Nozzle holder.
15. Valve stem.
16. Cap nut.
17. Nozzle.
18. Nozzle valve.

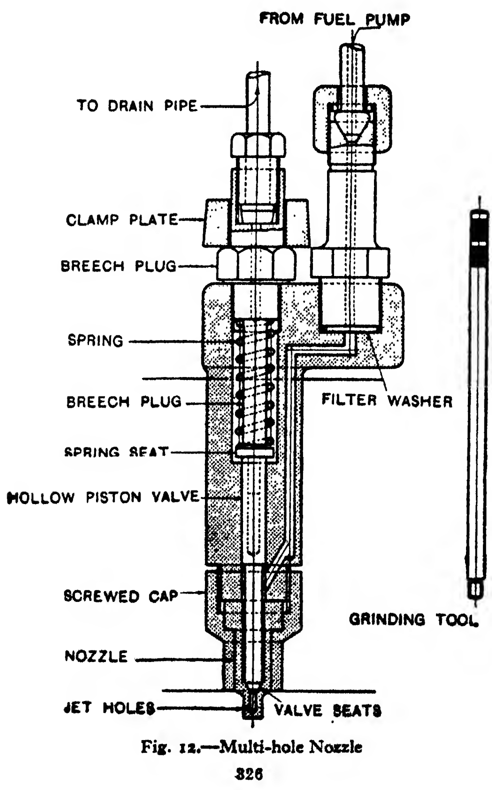


Fig. 12.—Multi-hole Nozzle

tests are usually necessary before the correct nozzle for a particular type of engine is finally decided upon.

Figs. 10 and 11 show sections of pintle type and single-hole nozzles as fitted by Leyland Motors, Ltd., to their ante-chamber and direct-injection engines respectively.

Fig. 12 is a section of the multi-hole nozzle fitted to the Gardner direct-injection engine.

Nozzles should be taken out at regular intervals for examination, testing and cleaning. The nearer the conditions of good fitting, adequate cooling and satisfactory filtering of the fuel are realized, the less attention the nozzles will need and the longer will be their efficient life. Since there is no other item of the equipment upon which the performance of the engine depends so much, regular inspection and attention will yield a handsome return for the trouble taken.

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## CHAPTER III

# Typical Engines

Over twenty British firms and a number of Continental and American firms are engaged in the manufacture of high-speed oil-engines. It is proposed in this chapter to describe the engines of three well-known British firms with large outputs, and these may be regarded as representative of the best types of engines both of the direct-injection and ante-chamber types.

It must be realized that the successful operation of a high-speed oil-engine lies largely in attention to detail in manufacture. The construction may be complex, but the details requiring attention at fairly frequent intervals should be readily accessible, easy to adjust, free from liability to go out of adjustment and reasonably fool-proof.

The success of the well-known Gardner direct-injection engine, from the points of view of reliability, easy starting from cold and economical fuel consumption, has stimulated interest in this type, and other firms, notably The Associated Equipment Co., Ltd., and Leyland Motors, Ltd., are now manufacturing engines with direct injection.

**The Gardner Engine.**—This engine, manufactured by L. Gardner & Son, Ltd., is built with four, five or six cylinders each of $4\frac{1}{4}$-in. bore and 6-in. stroke, with a swept volume of 1·4 litres per cylinder, and develops 17 b.h.p. per cylinder at 1700 r.p.m. The maximum brake m.e.p. is 102 lb. per sq. in. at 1100 r.p.m.

Fig. 13 is a part sectional elevation of the six-cylinder engine, and shows the main features of construction.

Fig. 14 is an external side view of the four-cylinder engine, showing the control mechanism.

Fig. 15 (p. 330) is a rear end view, showing the drive to cam-shaft, fuel injection pump and dynamo, with timing chain tension adjuster.

Fig 16 (p. 331) is a front end view.

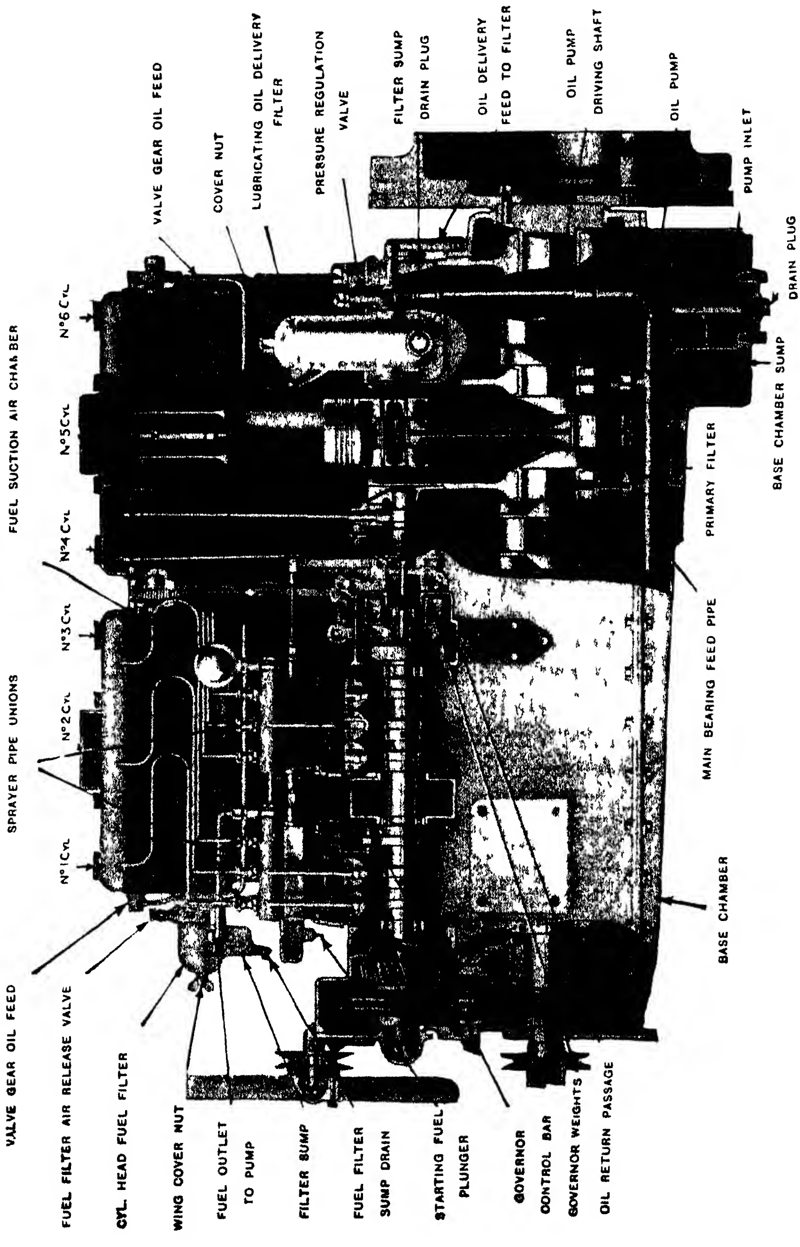


Fig 13 —Part Sectional Elevation of Gardner Six-cylinder Engine

The engine is fitted with separate crank-case, cylinder blocks and cylinder heads, and vertical overhead valves push-rod operated from cam-shaft situated in the crank-case

The cylinders are cast in an iron alloy specially developed in the makers' own foundry, and incorporate renewable dry liners of hardened material

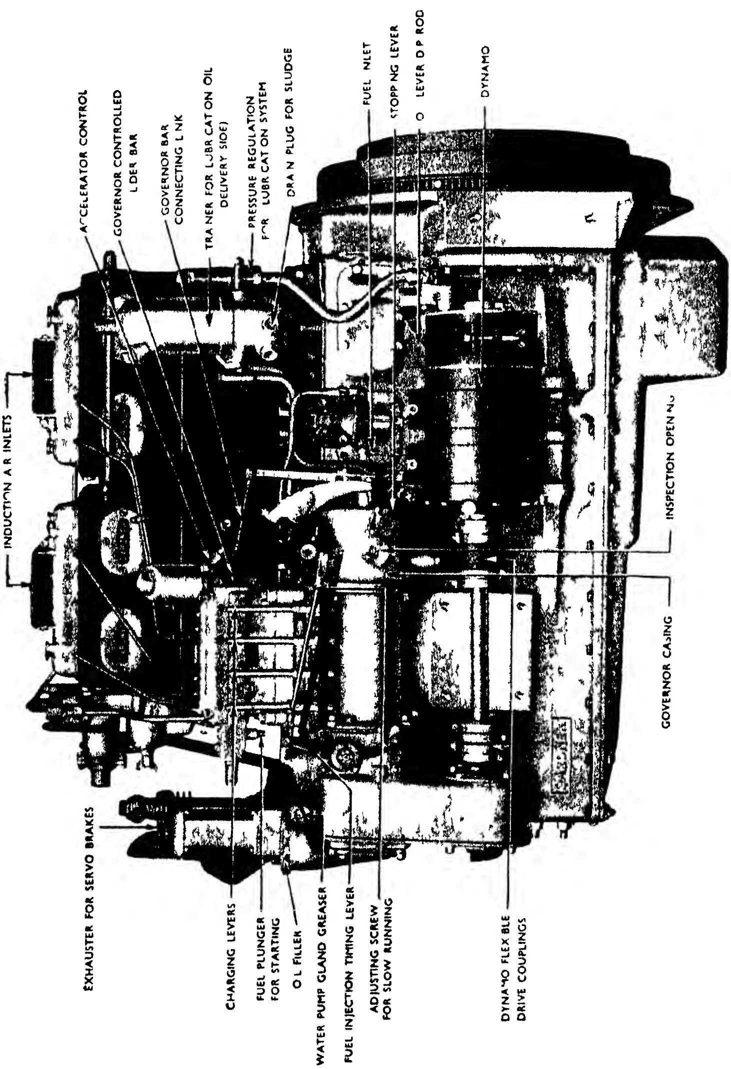


Fig 14 —Side View of Gardner Four cylinder Engine

The cylinder heads are cast of the same material as the cylinders, in blocks to correspond with the cylinder blocks. A closed chamber is formed upon the upper part of the head in which are housed the vertical overhead valves, valve levers and sprayers, also the patented devices for facilitating hand starting and for lubricating the valves and guides. The whole is completely enclosed by aluminium covers which also form air-intake passages.

The crank-case is cast in aluminium alloy or the lighter "elektron" metal, according to requirements. It is of rigid design, further supplemented by through bolts extending from below the crank-shaft centre line up to the cylinder feet. Large area oil passages integral with the casting transmit the pressure lubrication from the gear-type oil pump to the bearings.

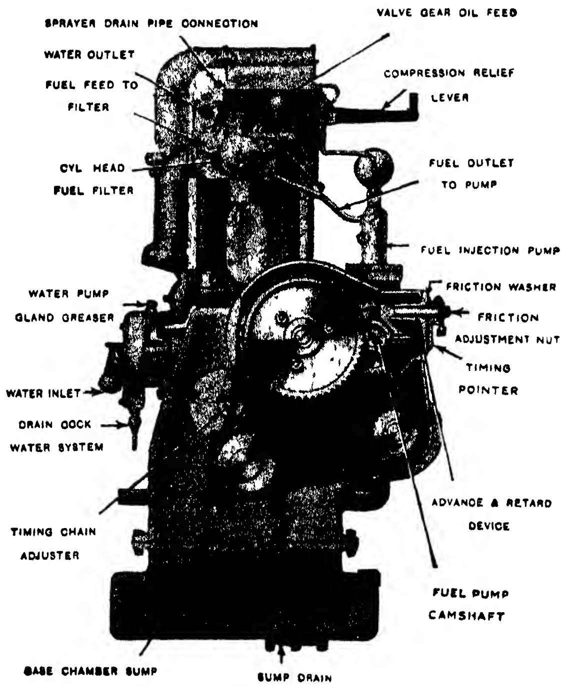


Fig 15.—Rear End View of Gardner Four-cylinder Engine

The crank-shafts are of nickel alloy steel machined all over from specially treated forgings. The journal bearings are 3½ in. diameter, the six-throw crank-shaft being supported by 7 of these.

The connecting-rods are of H section forged alloy steel with central duct leading lubricating oil under pressure from the crank-pin to the small end bearing. The big ends are fitted with detachable bearing shells lined with special purpose white metal, and the small end bearing is a bronze bush. The gudgeon-pin, of unusually large diameter, is of the full floating type.

The pistons are of heat-treated special purpose aluminium alloy, carrying four pressure rings and one scraper ring fitted just above the gudgeon-pin, and having the partly spherical open combustion chamber formed in the crown. The uppermost rings are of specially hardened material.

Pressure lubrication is provided by a gear-type oil pump of large capacity driven by helical gears from the main cam-shaft. The sump is protected by a primary gauze filter of large area, and the oil is delivered from the pump through a passage formed in the crank-case and thence by an external pipe to the delivery filter and pressure regulator, from which it is distributed to the various lubrication points. The surplus oil by-passed by the pressure regulator is separately circulated through the governor unit, the fuel pump cams, tappet mechanism and main timing drive.

The temperature of the cooling water is regulated by thermostatic control.

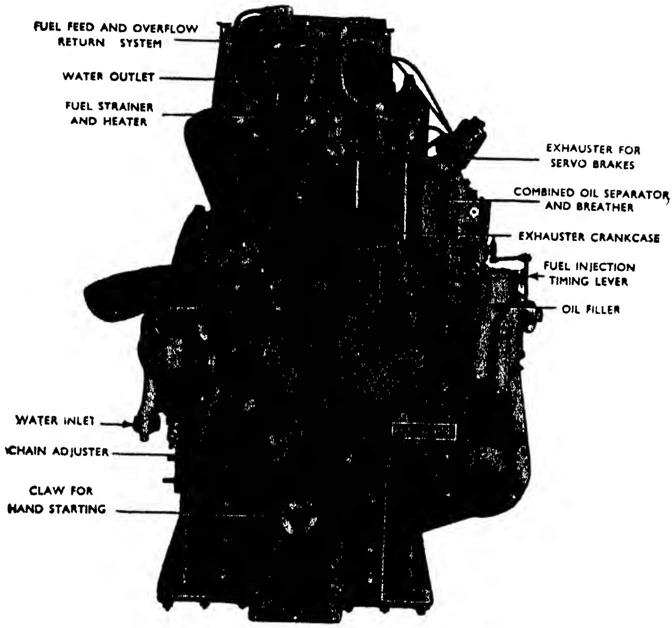


Fig. 16.—Front End View of Gardner Two-cylinder Engine

Fig. 17 shows the timing diagram for the engine. In this connection it should be noted that a crank angle of 54° represents a time interval of 0·005 sec. (approx.) only at 1700 r.p.m.

Fig. 18 (p. 333) shows typical performance curves for the six-cylinder Gardner engine. The figures from which the graphs are plotted are obtained during ordinary routine testing and can be reproduced at any time by an engine maintained in good condition. Two important conclusions can be drawn from an examination of these curves. (1) Since there is no very decided droop in the torque-speed curve at 1700 r.p.m., it would be practicable to maintain a high torque at much higher speeds, and only considerations of life and maintenance limit the speed to the above figure.* (2) The specific fuel consumption varies very little with speed.

* Actually a four-cylinder Gardner engine of $3\frac{3}{4}$-in. bore and $5\frac{1}{2}$-in. stroke, with a maximum speed of 3000 r.p.m., was fitted to a Lagonda car in 1935, and after some severe tests on Welsh hills, all of which gave complete satisfaction, a speed of 83 m.p.h. was attained on a level road. (*The Autocar*, 23rd August, 1935.)

This is evidence that the combustion efficiency varies very little with speed. Since the spray formation and air circulation are all liable to variation with variation of speed, this must be regarded as a very satisfactory achievement.

A decompression device operated by a lever is used when the engine has to be started from cold by hand. If an electric starter is fitted the use of this device is unnecessary.

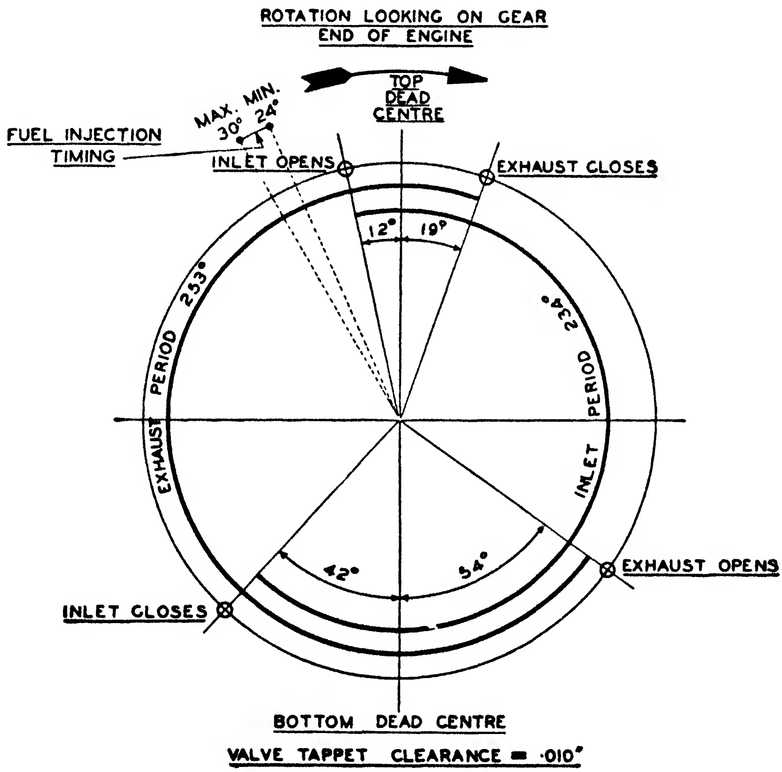


Fig. 17.—Timing Diagram

**The A.E.C. Engine.**—The Associated Equipment Co., Ltd., placed the first A.E.C. high-speed oil-engine on the road in 1928, and since that time experimental work has been carried on continuously with a view to obtaining steady improvement in results.

The improvement which had been made by the A.E.C. in the power that can be developed by a high-speed oil-engine for a given cylinder volume had brought the output obtainable from the original A.E.C. 8·8-litre engine up to a figure greater than that which is really necessary for even the heaviest type of vehicle, and considerably more than is required for double-deck passenger vehicles running in city services. Advantage was therefore taken of this fact to produce a six-cylinder 7·7-litre oil-engine which while having an output adequate for all purposes is at the same time smaller and lighter than the original engine, thus saving both weight and space, with a corresponding advantage in carrying capacity.

The combustion chamber of this engine is the A.E.C.-Ricardo swirl-chamber design referred to in Chapter I, which ensures that the combustion reactions keep pace with the engine speed and thus prevent the engine speed from being limited by the speed of combustion.

Fig. 19 is a section of the cylinder showing the essential details of com-

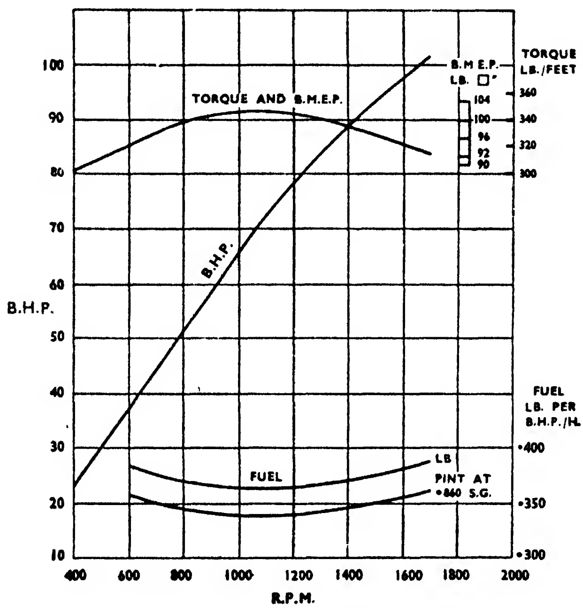


Fig. 18.—Performance Curves, Six-cylinder Gardner Engine

bustion chamber, injection nozzle and valve gear, and fig. 20 is an external view of the engine.

The unit is built upon a cast-iron crank-case of rigid construction. The cylinder block is secured to the crank-case by means of the main bearing bolts, which are carried upward to the top of the crank-case for this purpose, and the crank-case is thus relieved of all strain due to gas pressures.

Two cast-iron cylinder heads are fitted, each covering three cylinders.

The valves are operated by short push rods from the cam-shaft, which is situated in a tunnel bored in the cylinder block. Flat-faced tappets transmit the cam motion to the valves. The drive to the cam-shaft includes a pair of gears in which is incorporated a special spring arrangement for eliminating all backlash. A light cover fitted over the cylinder heads also incorporates the air intake.

The pistons, which are of aluminium alloy, carry four pressure rings and three oil scrapers, two of which are below the gudgeon-pin. The gudgeon-pins are fully floating and are secured against end movement by special circlips.

Lead bronze is used for lining the connecting-rod bearings, the main bearings being of white metal.

The crank-shaft is carried on seven main bearings and is designed to have ample stiffness. It is bored through its length to reduce the weight.

Lubrication is by a gear-type pump having a patented device for providing a supply of oil at low pressure to the valve gear.

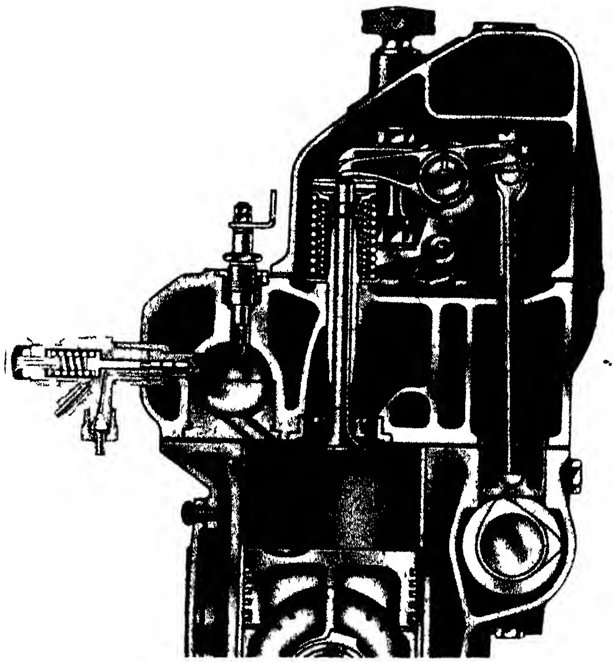

Fig. 19.—Section of Cylinder, A E C. Engine

C.A.V.-Bosch fuel injection equipment is used.

All auxiliaries with the exception of the starter are mounted on the near side of the engine. Accessibility is thus provided to those parts which may occasionally require attention.

Fig. 21 illustrates the variation of specific fuel consumption with load. Actually the fuel consumption per *indicated* horse-power hour decreases continuously as the load decreases, mainly owing to reduced heat losses with weaker mixtures, but since the mechanical efficiency decreases as the load is reduced the curve rises somewhat at the smaller loads. (1 pint of fuel oil = about 1·08 lb.)

The A.E.C.-Ricardo power units, described above, have achieved outstanding success, under the most arduous conditions, in every branch of the transport industry. Under certain conditions, however, where the fuel consumption is of paramount importance and vehicles are not called upon to operate for long periods at full torque and high speed, the direct-injection engine has much to recommend it. Easier starting from cold

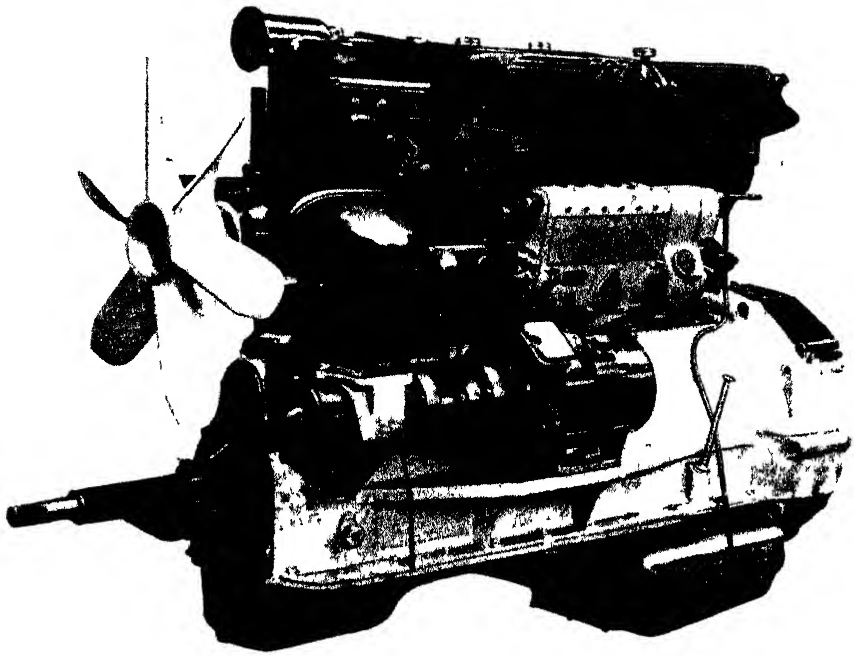

Fig 20 — A E C Engine External View

and lower specific fuel consumption at all loads are recognized to be features of a well-designed direct-injection engine, and for several years A.E.C. engineers have been engaged upon research work with a view to production

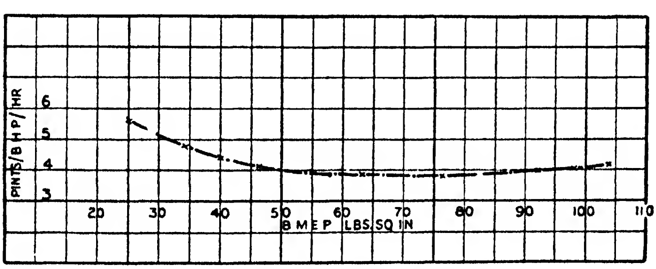


Fig 21.—Part-load Consumption Curve

of a direct-injection oil-engine which will be free from the disadvantages usually associated with some units of this type.

Before the company felt justified in placing the engine which was finally evolved and is described below in the range of engines offered for commercial vehicles, engines had been tested over more than a million miles of actual operating conditions by selected operators.

The main features of the engine, the dimensions of which remain at 105-mm. bore and 146-mm. stroke, are the same as those of the Ricardo type described above. The differences lie in the open combustion chamber, partially masked inlet valve, position of injection nozzle and cavity in piston head.

Fig. 22, which is a cross-section of the cylinder head, shows clearly the new features of the engine, and should be compared with fig. 19.

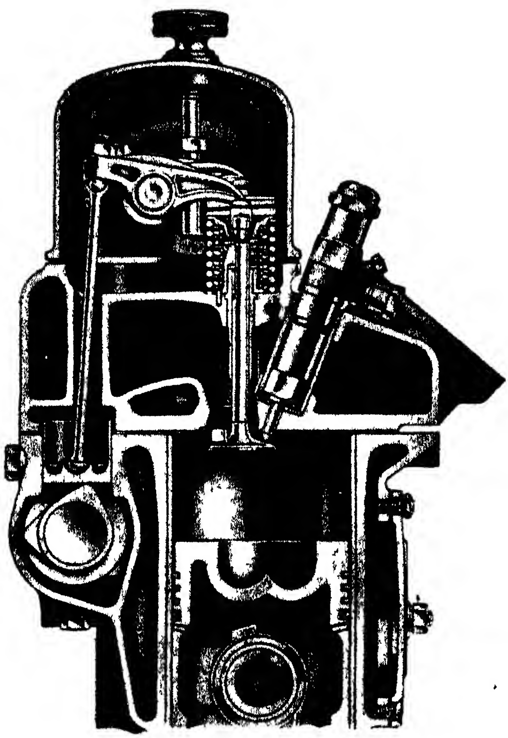

Fig 22 —Cylinder Head of the A E C 7 7-litre 6-cyl. Direct-ignition Oil-engine

Figs. 23 and 24 show in elevation and plan respectively the nature of the turbulence produced by the combination of piston cavity, mask on inlet valve and fuel spray. The effectiveness of the system of combustion is borne out by the low fuel consumption (see fig. 28) and the absence of exhaust smoke, particularly at the top end of the speed range.

Figs. 25, 26 and 27 are external views of the engine, showing clearly the positions of the accessories.

Fig. 28 (p. 339) shows performance curves of the A.E.C. direct-injection engine. It will be noted that the minimum fuel consumption per b.h.p. hour is 0·36 lb. at 1100 r.p.m., while with the Ricardo type the minimum fuel consumption is 0.405 lb. at 1100 r.p.m., i.e. there is a saving of 9 per cent with the direct-injection type.

**Leyland High-speed Engines.**—Messrs. Leyland Motors, Ltd., manufacture two types of oil-engine: (1) the "Light-Six" ante-chamber oil-engine, bore $3\frac{1}{2}$ in., stroke 5 in., and (2) the six-cylinder direct-injection oil-engine for heavy duty, bore $4\frac{1}{2}$ in., stroke $5\frac{1}{2}$ in.

**The "Light-Six" Ante-chamber Oil-engine.**—Fig. 29 (p. 340) shows a longitudinal section of the engine, which develops 34 b.h.p. at 1000 r.p.m. with a fuel consumption of 0·41 pint per b.h.p. hour, and fig. 30 is a cross-section showing the ante-chamber and spraying nozzle. The ante-chamber is similar to the Ricardo type in principle, but differs in construction, as will be seen by comparison with figs. 2 and 19. The chamber is simple in construction and has no loose parts. A useful feature is that when the cylinder head is removed the ante-chamber is easily accessible for cleaning.

The cylinder block is cast integral with the crank-case in cast iron,

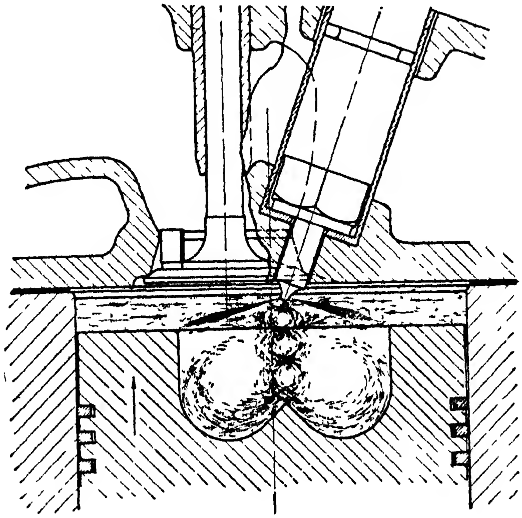

Fig. 23

thus forming a housing for the crank-shaft, the seven main bearings, which are dowelled in position, and the cam-shaft, which is located on the near side. Dry liners of centrifugally cast iron are fitted.

The crank-shaft is machined from a nickel-steel forging and is balanced

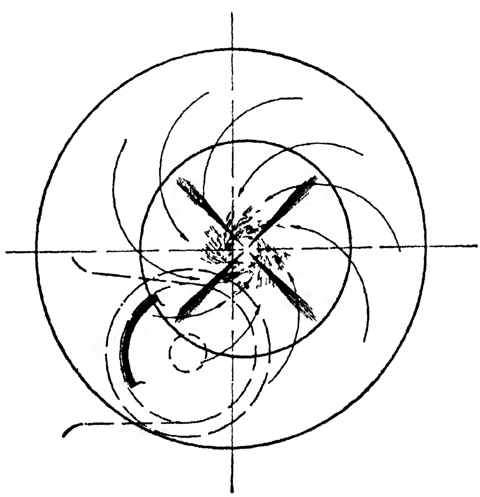

Fig. 24

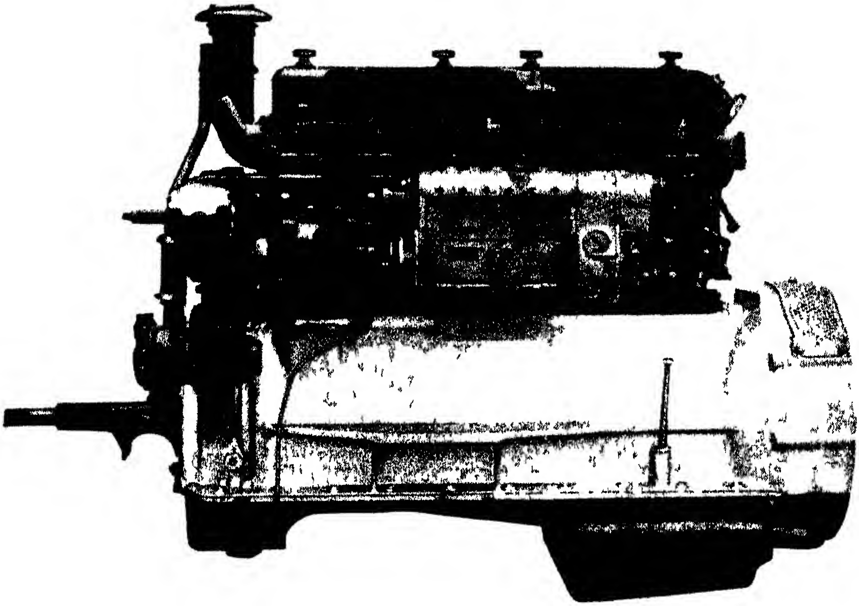

Fig 25 —A E C 7 7-litre 6-cyl Direct-injection Oil-engine

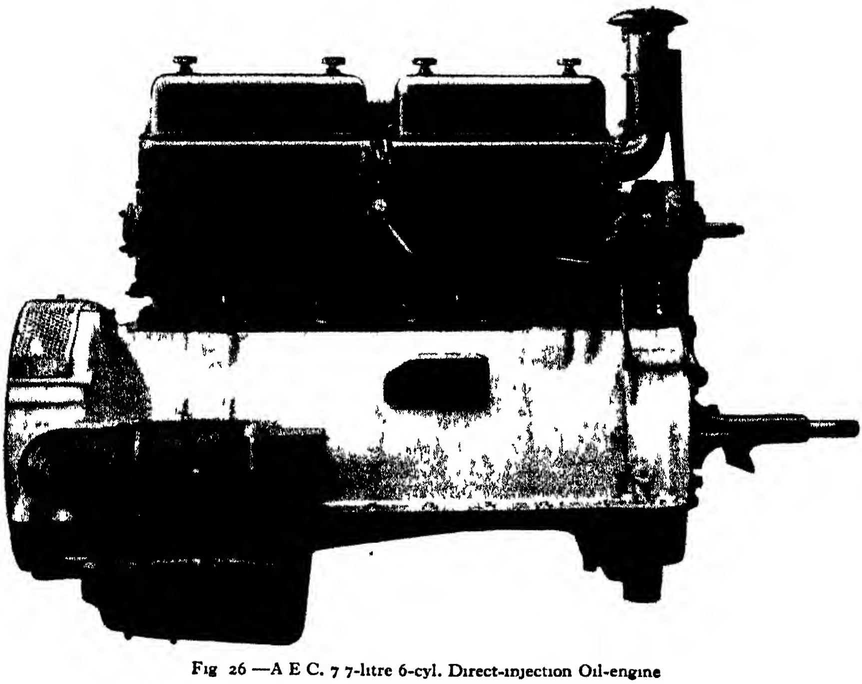

Fig 26 —A E C. 7 7-litre 6-cyl. Direct-injection Oil-engine

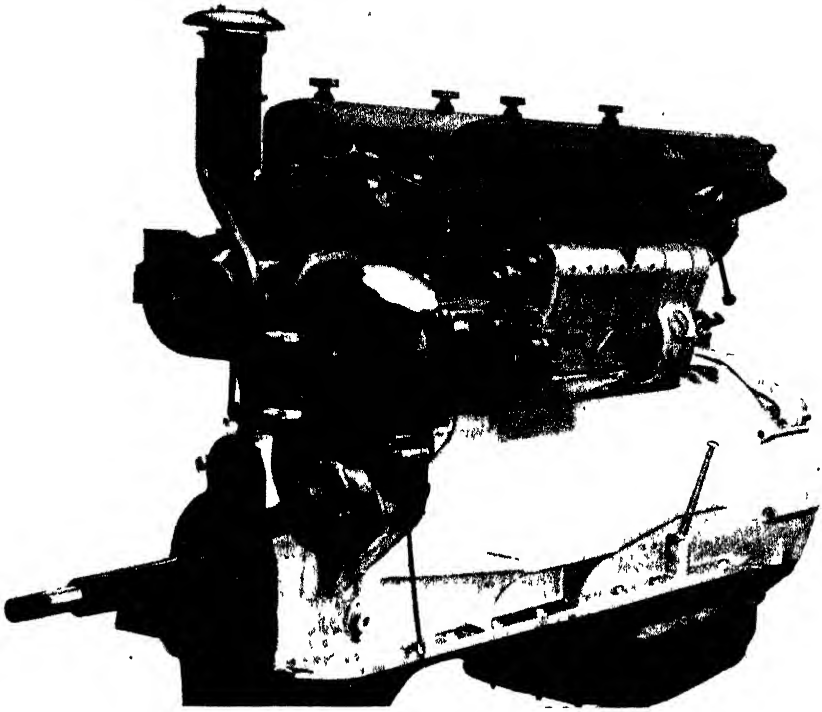

Fig. 27.—A E.C. 7·7-litre 6-cyl. Direct-injection Oil-engine

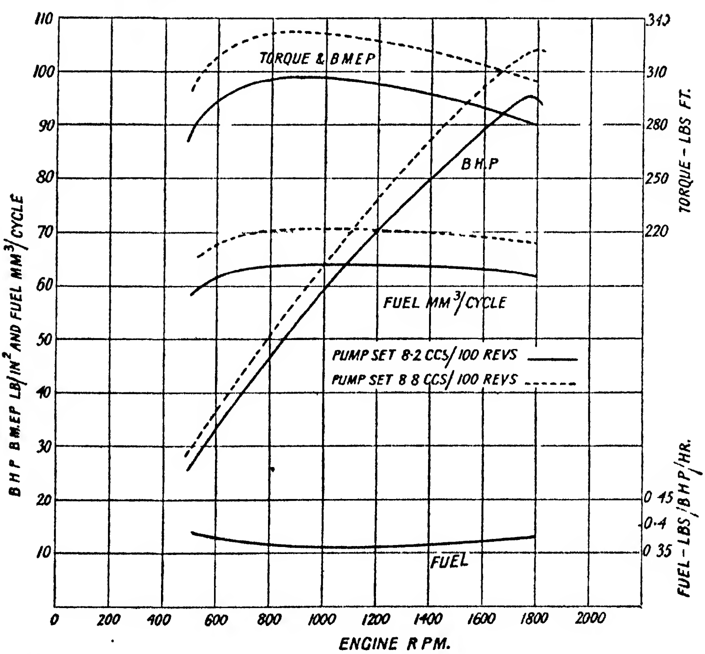


Fig. 28.—Performance Curves of A.E.C. 105-mm. Bore × 146-mm. Stroke 6-cyl. Toroidal D.I. Engine

statically and dynamically, an oscillation damper being fitted at the forward end.

The connecting-rods are of alloy steel and the big end bearings are of copper-lead alloy.

The pistons are of heat-treated aluminium alloy and are fitted with four compression rings and one scraper ring below the gudgeon-pin. The gudgeon-pins are fully floating.

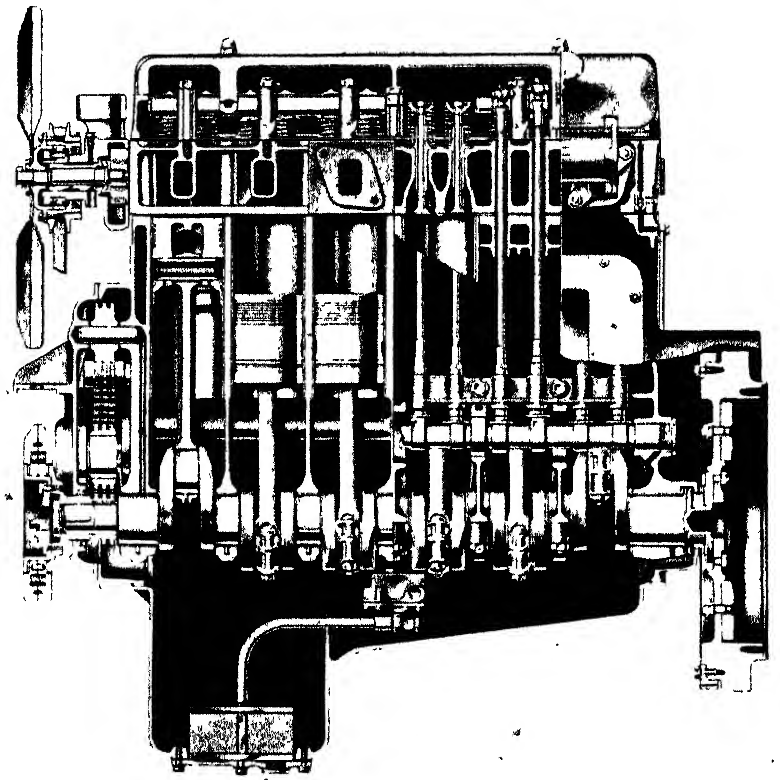

Fig. 29.—Leyland " Light-Six " Oil-engine

The inlet valves are of nickel steel and the exhaust valves of heat-resisting steel. The valve seats are screwed in and replaceable, the exhaust seats being faced with stellite. The valves are operated by push-rods and rocker levers.

C.A.V.-Bosch fuel-injection equipment with fixed timing is fitted. The governor (figs. 7 and 8) gives an idling speed of 300 r.p.m. and a maximum engine speed of 1900 r.p.m.

Fig. 31 shows the valve timing and fig. 32 is a typical performance curve.

**Direct-injection Leyland Oil-engine.**—This six-cylinder engine develops 58 b.h.p. at a speed of 1000 r.p.m. with a fuel consumption of

0·37 pint per b.h.p. hour, an increase of 71 per cent in power for an increase of 82 per cent in cylinder capacity and an ease of 10 per cent in specific fuel consumption as compared with the "Light-Six" engine.

Fig. 33 is an external view of the engine, the general construction of which differs from that of the ante-chamber engine.

The cylinder block is a monobloc casting of chromium iron with large water spaces, and is a simple clean casting. Dry liners, centrifugally cast, are fitted.

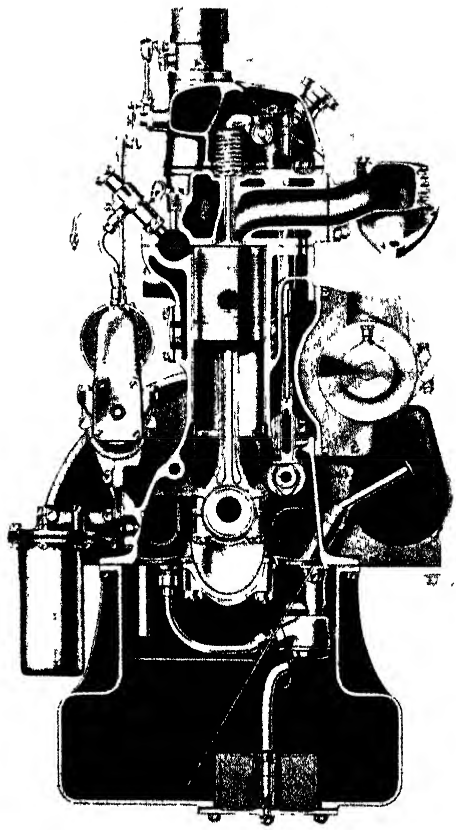

Fig. 30.—Leyland "Light-Six" Engine

The crank-case is an aluminium casting providing a rigid foundation for the crank-shaft bearings. It is well ribbed internally, has wide flanges on each side to increase the torsional rigidity, and is carried below the crank-shaft centre to resist bending. The main bearings are dowelled and held by bolts which pass through to the base of the cylinder block, thus relieving the crank-case of all tensile loading.

The cylinder head, shown in section in fig. 34, is a single casting into which stellite-faced valve seats are screwed. The cam-shaft, driven by skew gears, is carried in the head in four bronze bushes and rotates in a trough of oil.

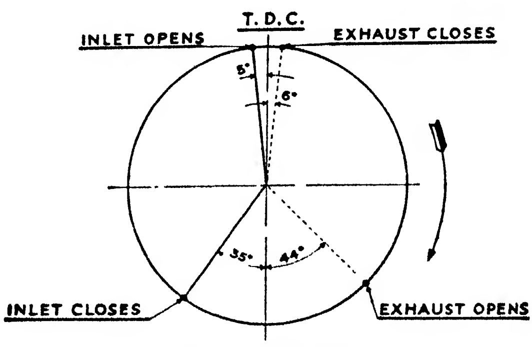


Fig. 31.—Valve Timing

The exhaust valves are of conventional design, but the inlet valves are masked on one side (4, fig. 35). For this reason the valves are keyed (2, fig. 35) in their guides (3, fig. 35), and to prevent wrong replacement the guides are located by pegs in the cylinder head. The keys 2 are retained by circlips (1, fig. 35).

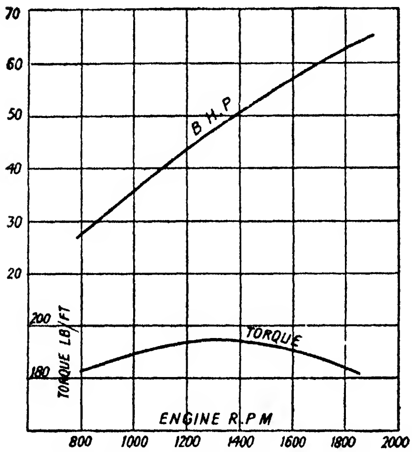


Fig 32 —Typical Output Curve

The pistons (fig. 34) are of heat-treated aluminium alloy and have an open cup offset from the centre of the crown. Fuel injection takes place tangentially into this cup as shown. The air as it enters the cylinder is encouraged to rotate in an orderly swirl around the cylinder axis, and the fuel mixes with the swirling air, being kept well away from the cylinder walls, where it would have a detrimental effect on piston lubrication.

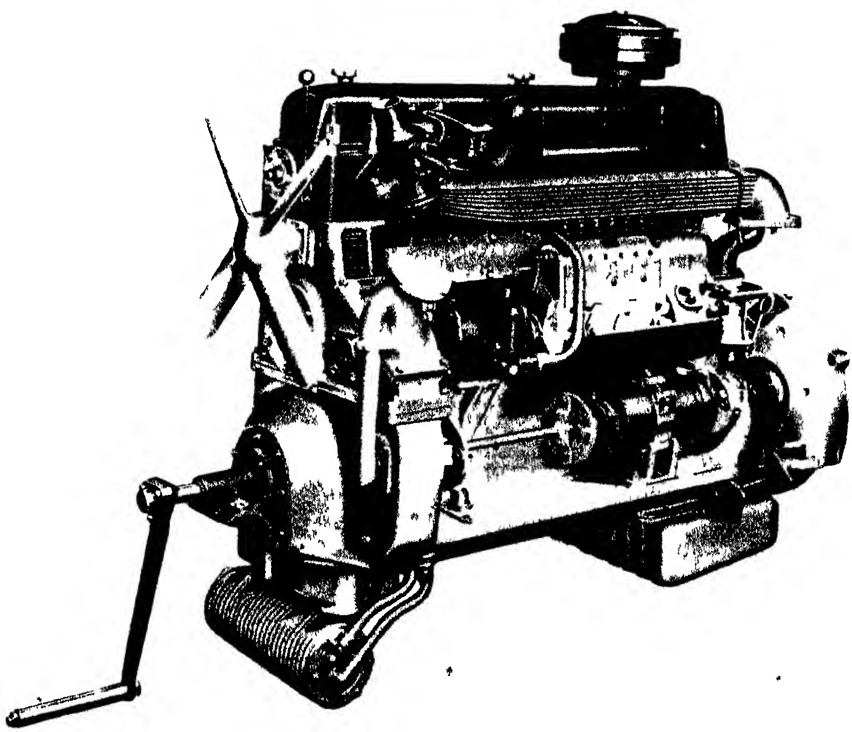

Fig 33 —D -I. Leyland Engine

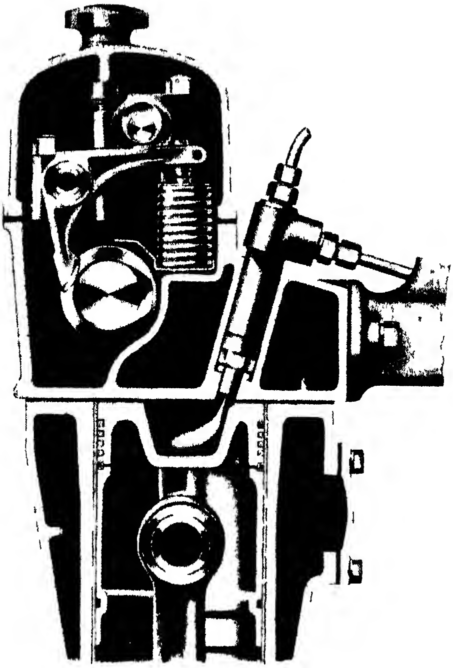

Fig 34 —Cylinder Head

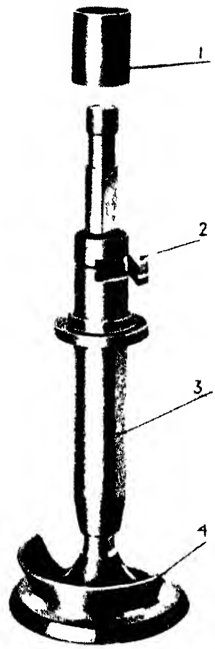


Fig 35

1, Circlip 3 Valve guide.
2, Key 4 Masking

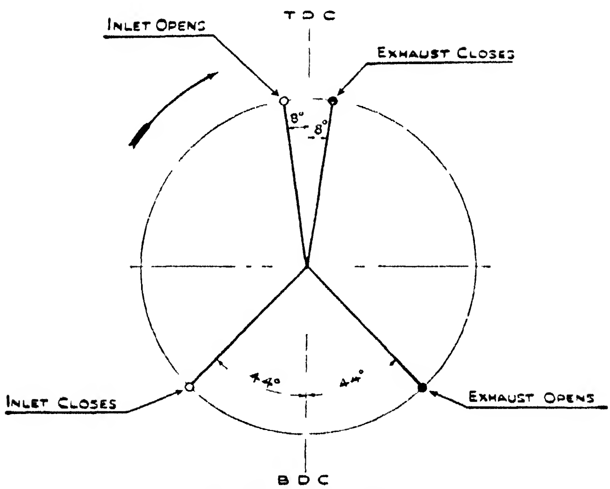


Fig 36 —Valve Timing

The injection pump and filter are manufactured by C.A.V.-Bosch, Ltd., the injection nozzles (fig. 11, p. 326) being of Leyland design.

Fig. 36 shows the valve timing, and fig. 37 is a typical performance curve.

(Acknowledgments are due to the firms mentioned above for the supply of illustrations and technical information.)

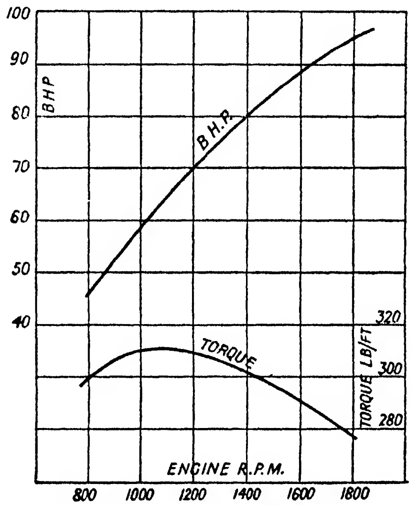


Fig. 37.—Typical Output Curve

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## CHAPTER IV

# Calculations

**Cylinder Sizes.**—Assuming a temperature of 80° C. and an absolute pressure of 13 lb. per sq. in. at the end of the induction stroke, the weight of air admitted per stroke is 0·056V, where V = stroke volume in cubic feet.

Minimum weight W (lb.) of air required for 1 lb. of fuel (for an average fuel oil) = 14·7.

∴ Maximum weight of fuel that can be burned per working stroke is given by

$$w = \frac{0{\cdot}056V}{14{\cdot}7} = 0{\cdot}0038V \text{ lb.} \quad \dots\dots\dots\dots\dots(1)$$

If $d$ = diameter of cylinder (in.), $s$ = stroke (in.),

$$V \text{ (c. ft.)} = \frac{\cdot785 d^2 s}{1728};$$

$$\therefore\ w = \frac{\cdot0038 \times \cdot785}{1728} d^2 s = 1{\cdot}73 \times 10^{-6} \times d^2 s. \quad \dots(2)$$

Assuming that not more than 85 per cent of the air can be consumed, if the speed is N r.p.m. (four-stroke engine) the maximum practicable fuel consumption per hour (lb.) is

$$F = 1{\cdot}73 \times 10^{-6} \times d^2s \times 30N \times 0{\cdot}85 = 4{\cdot}4 \times 10^{-5} \times d^2sN. \quad (3)$$

Let $x$ = fuel consumption per b.h.p. hr. at maximum load (lb.),
$B$ = maximum b.h.p. per cylinder,
$\eta p$ = brake m.e.p. at load B,

then, for a four-stroke engine,

$$\eta p = \frac{B \times 10^6}{d^2sN}, \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (4)$$

and from (3), $$xB = 4{\cdot}4 \times 10^{-5} \times d^2sN;$$

$$\therefore\ d^2s = 22{,}800\,\frac{xB}{N}; \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (5)$$

and from (4), $$\eta p = \frac{44}{x}. \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (6)$$

Thus if the fuel consumption is 0·4 lb. per b.h.p. hr. at fuel load the engine must be designed for a brake m.e.p. of 110 lb. per sq. in. From equation (5), taking $x = 0{\cdot}4$, we see that

Maximum practicable brake horse-power per cylinder is given by

$$B = \frac{d^2sN}{9100}; \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (7)$$

If $d$ and $s$ are in *millimetres*, (7) becomes

$$B = \frac{\left(\frac{d}{10}\right)^2\left(\frac{s}{10}\right)\left(\frac{N}{1000}\right)}{150}. \quad \ldots\ldots\ldots\ldots\ldots (7a)$$

**Relation between brake m.e.p., volumetric efficiency and thermal efficiency.**

Let $V$ = stroke volume (c. ft.) for cylinder,

$e_v$ = volumetric efficiency

$$= \frac{\text{actual weight of air taken in per stroke}}{\text{weight of fill at } 15^\circ \text{ C. and } 14{\cdot}7 \text{ lb. per sq. in.}},$$

$W$ = minimum wt. of air required per lb. of fuel
= 14·7 lb. for average fuel,

$r$ = air ratio

$$= \frac{\text{actual wt. of air taken in per lb. of fuel}}{W},$$

$H$ = calorific value of fuel (B.Th.U. per lb.),
$w$ = wt. of fuel used per b.h.p. hour (lb.),

$e_t$ = brake thermal efficiency $= \dfrac{2545}{wH}$.

Then, Weight of air taken in per stroke $= 0{\cdot}077Ve_v$. ........(8)

$$\text{Weight of fuel burned per stroke} = \frac{{\cdot}077Ve_v}{Wr}. \quad \text{........(9)}$$

$$\text{Heat generated per stroke} = \frac{{\cdot}077Ve_vH}{Wr}. \quad \text{..........(10)}$$

$$\text{Heat converted to useful work per stroke} = \frac{{\cdot}077Ve_ve_tH}{Wr}. \quad \text{..(11)}$$

$$\text{Useful work done per stroke (ft.-lb.)} = \frac{{\cdot}077Ve_ve_tH}{Wr} \times 778,$$

$$= \frac{59{\cdot}9Ve_ve_tH}{Wr}. \quad \text{......(12)}$$

But useful work done per stroke (ft.-lb.) $= 144\eta pV$; ......(13)

$$\therefore\ 144\eta pV = \frac{59{\cdot}9Ve_ve_tH}{Wr},$$

$$\therefore\ e_v = \frac{\eta pWr}{{\cdot}415e_tH}. \quad \text{..................(14)}$$

Substituting for $e_t$, we have

$$e_v = \frac{rW\eta pw}{1060}. \quad \text{....................(15)}$$

By the use of equation (15) it becomes possible to estimate the volumetric efficiency of a four-stroke engine from test results and an exhaust-gas analysis. The air-box and throttle-plate method (Watson and Schofield, *Proc. I. Mech. E.*, 1912) is more direct, but less convenient and takes longer.

$\eta p$ can be calculated from (4), and $rWw$ is the actual weight of air taken in per b.h.p. hour. The author's formula (evolved in 1920), given below, enables the air supply per lb. of fuel to be calculated easily from the exhaust-gas analysis (assuming that there are no unburned hydrocarbons in the exhaust).

Let $W_A = rW$ = actual weight of air per lb. of fuel,

$h$ = wt. of hydrogen in 1 lb. of fuel,

$c$ = wt. of carbon in 1 lb. of fuel,

$p_1$ = percentage of $CO_2$ in exhaust gases,

$p_2$ = percentage of CO in exhaust gases;

then,

$$W_A = 1{\cdot}2\left\{6h + \frac{200 - p_2}{p_1 + p_2}c\right\}. \quad \text{....................(16)}$$

*Example.*—Given $c = 0{\cdot}85$; $h = 0{\cdot}14$; $p_1 = 11$; $p_2 = 1$; $w = 0{\cdot}38$; $d = 4$ in.; $s = 5$ in.; B = 14 b.h.p. per cylinder at 1600 r.p.m.

From (4), $$\eta p = \frac{14 \times 10^6}{80 \times 1600} = 109 \text{ lb. per sq. in.}$$

From (16), $$rW = 1{\cdot}2\left\{{\cdot}84 + \frac{199 \times {\cdot}85}{12}\right\} = 17{\cdot}9.$$

$\therefore$ from (15), $$e_v = \frac{17{\cdot}9 \times 109 \times {\cdot}38}{1060} = {\cdot}70 \text{ (70 per cent).}$$

As a check on the above we may proceed as follows:

Actual wt. of air per hour $= 17{\cdot}9 \times {\cdot}38 \times 14 = 95{\cdot}3$ lb.

Wt. per hour at 100 per cent volumetric efficiency

$$= \frac{63}{1728} \times 48{,}000 \times {\cdot}077 = 135 \text{ lb.};$$

$$\therefore e_v = \frac{95{\cdot}3}{135} = {\cdot}705 \text{ (70}{\cdot}\text{5 per cent).}$$

The volumetric efficiency depends upon the pressure and temperature of the cylinder contents at the end of the induction stroke. The pressure in the cylinder depends upon the difference of pressure required to force the air through the inlet-valve opening, and this in turn depends upon the size and lift of the inlet valve and the speed of revolution. The final induction temperature depends upon the temperature of the air at entry and the weight and temperature of residual gases in the clearance space, the latter depending upon the temperature of the exhaust and the former upon the compression ratio.

**Starting Conditions.**—To secure reliable ignition the temperature at end of compression should not be less than 350° C. (662° F.). The final temperature depends upon the initial temperature, the compression ratio and the index of the compression curve. On starting with a cold cylinder the initial temperature and index of compression will be appreciably lower than when the engine is working, and in order to ensure starting with a few turns only of the crank either the compression ratio must be higher than necessary for running conditions or (in the case of antechamber engines) a heater plug must be used.

Efficient spraying of the fuel is obviously a vital factor. Preliminary heating of the water in the jackets is also a useful aid to quick starting, and suitable lubricant, which has a reasonably low viscosity when cold, must be used.

**Conclusion.**—The history of the oil-engined road vehicle in Britain dates from 1929, when a Mercedes-Benz tank wagon, imported into England, was put into regular service. The first omnibus with an all-British oil-engine, manufactured by L. Gardner and Sons, was put into service in

Leeds in 1930 and gave very satisfactory service. In the comparatively short period which has elapsed since then, the oil-engine has practically ousted the petrol-engine as a motive power unit for commercial vehicles, and leading manufacturers now feature oil-engined chassis as standard, petrol-engines being fitted only where directly specified. All doubts regarding reliability and upkeep costs have practically disappeared, and intensive research and experiment will undoubtedly result in further reductions of weight and substantial increases in revolution speeds in the near future. Improved types of combustion chamber and governing devices have gone far towards solving the problem of smoke emission during acceleration, with consequent abatement of fuel consumption, reduction of internal carbon deposit, and improvement of external atmosphere.

# MOTOR-CARS

BY

H. D. TEAGE AND MONTAGUE TOMBS

Of the Technical Staff of *The Autocar*

# Motor-cars

**Automobile Design of To-day.**—During recent years a great change has taken place in the effectiveness of the private car, a change so gradual that the extent of it is only to be realized by comparing the road behaviour of the modern car with that of one built ten years ago. The difference is not so apparent in the general design or specification, but in the performance, and again not so much in the extent thereof as in the manner of it. Above all things, the modern car is extremely quiet, most comfortable, exceedingly easy to handle, possessed of a good acceleration and of powerful, reliable brakes. Moreover, it will retain these characteristics for a considerable period without need of serious attention. Yet another remarkable fact is that such characteristics are to be found to a marked degree in the average modern car irrespective of size and of price.

To a certain extent standardization of mechanism has taken place, not so much in the cars as a whole, but chiefly in regard of component parts. Despite that fact and despite the generally high standard, variations in the recipe for building the car are easily able to secure a definite individuality for different makes. Also, cars made in different countries have very different characteristics, as is only natural, yet the paper specifications of most cars read very much alike.

It may be said that the current specification of the time would read much as follows:

Six-cylinder engine, with push-rod operated overhead valves in specially shaped anti-detonation combustion chambers, compression ratio between 6 and 7 to 1. In order to obtain the increase of power available from modern fuels, large size balanced crank-shaft carried in steel-backed bearings, H-section connecting-rods of steel, aluminium alloy pistons with some form of expansion regulation and a specially treated surface to resist wear, three or four piston-rings with at least one special kind to secure oil control, sparking plugs of 14 mm. size, coil ignition with automatic advance, down-draught carburettor with air silencer and cleaner, pump water cooling, pressure lubrication with renewable filter element and voltage-controlled ventilated dynamo driven by belt.

In unit with the engine a four-speed gear-box with helical-toothed gears and synchromesh on second, third, and top for easy gear changing, and dipstick for testing oil level. Between engine and gear-box a venti-

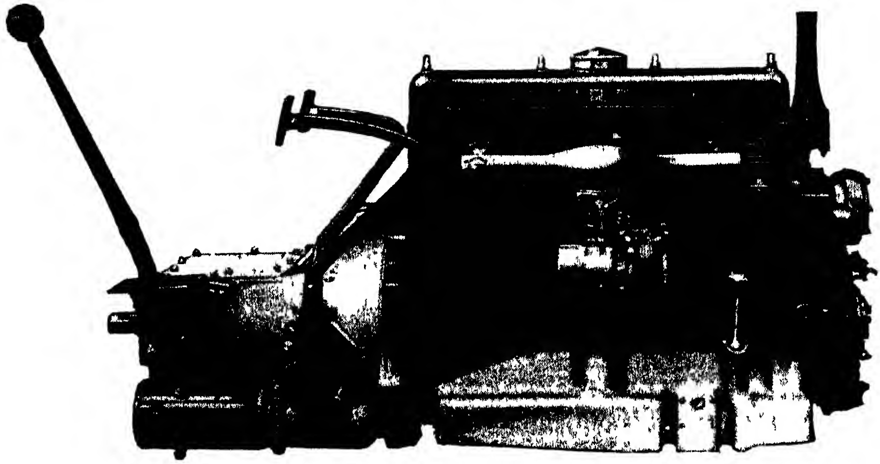

Fig. 1 —16-h p. Six-cylinder Sunbeam Engine, showing Pump Type Carburettor, and the Extra Large Sump-wheel holding about 3 gallons of Oil

lated casing containing light flywheel, electric starter-ring, and single dry-plate clutch with a spring disc and a cushioned hub.

The unit mounted in the frame of the car on thick and soft rubber

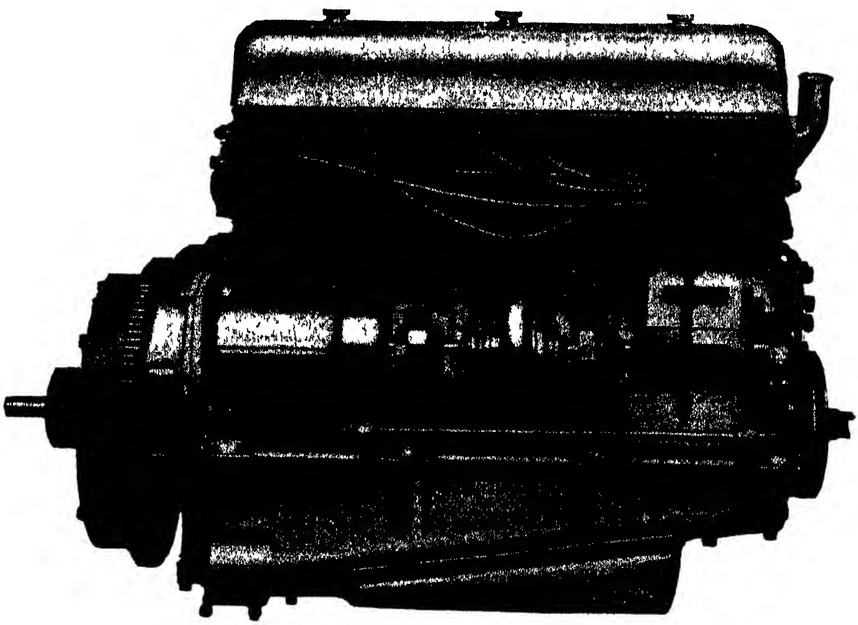

Fig. 2.—Six-cylinder Alvis Engine

blocks specially disposed so as to provide a flexible cradle which prevents the transmission to the rest of the car of engine vibration and noise.

Transmission by an open tubular propeller-shaft with needle-bearing

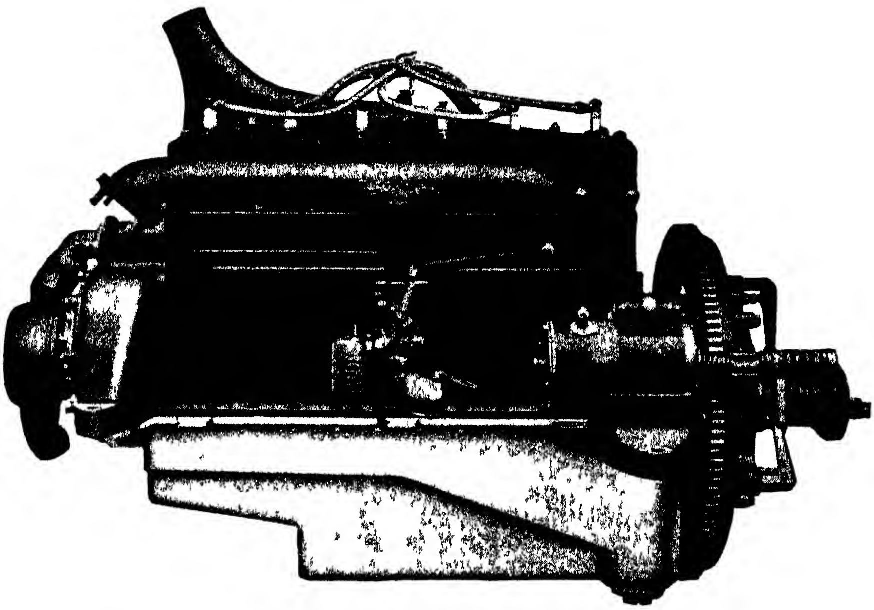

Fig 3—12-h p Armstrong Siddeley Engine One of the smallest Six-cylinder Engines

universal joints and a splined telescopic joint, to a spiral bevel final drive-gear contained in the centre of a pressed steel or " banjo " rear-axle casing having three-quarter floating axle-shafts.

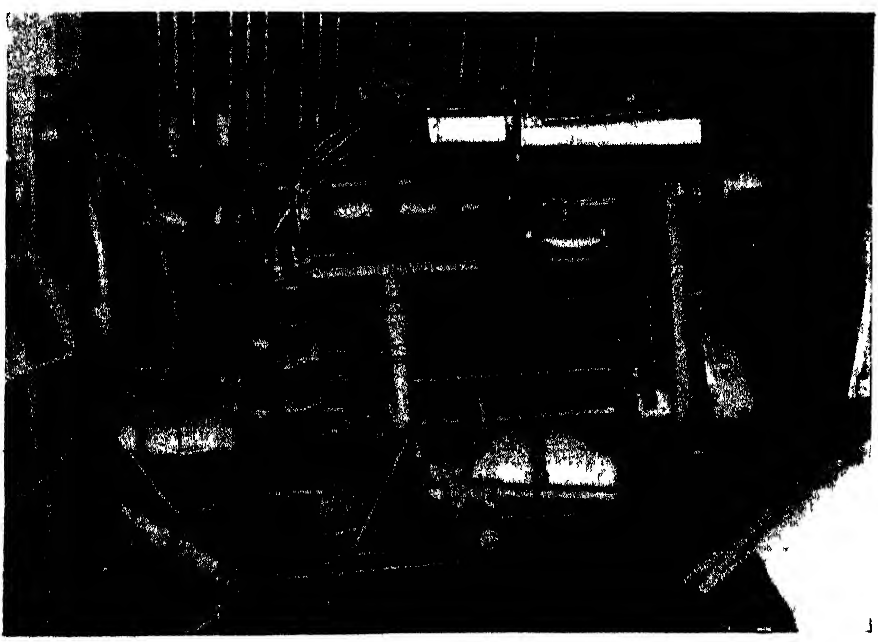

Fig 4—The Studebaker " Director " Engine—a Typical American Design

Main frame of box section throughout, with or without a cruciform centre. Suspension by half-elliptic springs back and front, or with independent front springing. Steering-gear by worm or cam and lever, brakes operated by direct pull-rods, by cable, or hydraulically, brake shoes provided with easy adjustment, wheels of pressed steel either disc or spoke, and shod with low-pressure tyres. Saloon body of pressed steel, with flush-fitting steel sliding roof. Easy jacking system provided, and bumper bars fore-and-aft.

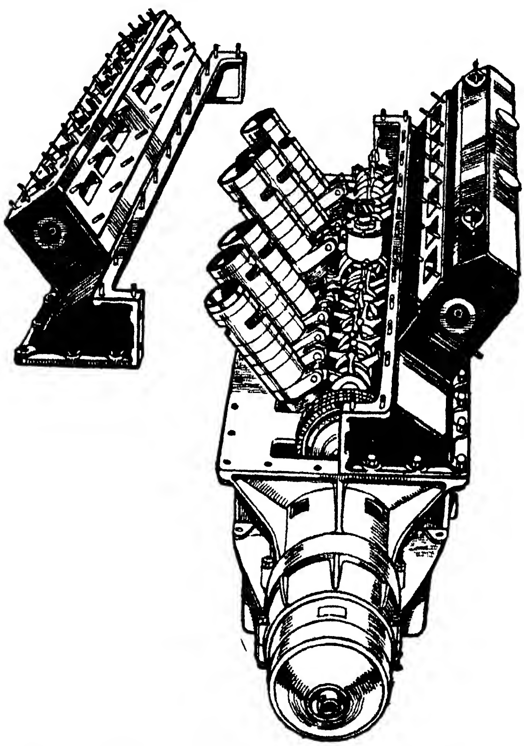

Fig. 5.—A Twelve-cylinder Vee Sleeve-valve Engine, the Voisin

Although this specification may be a fair average, the application of it may produce very different varieties of cars, for it is the blending of the components which decides eventual characteristics, and success for the recipe largely depends upon the technique developed by the manufacturer. The endeavour to describe the thousands of details of design and production which together bring about the completion of the car would involve the writing of a number of large volumes, and is altogether outside the scope of the present work. Nevertheless, it is possible to give a brief outline of current design and many of the reasons for the adoption of certain features.

**Engines.**—Six-cylinder designs are more numerous than any other type on the world markets, but in this country more four-cylinder engines of the smaller size are actually sold. The reasons in Britain are, of course, the horse-power tax and the suitability of smaller cars for our roads and traffic conditions. In medium and large sizes the six-cylinder engine does seem to offer the best all-round compromise, but for cars up to 12 or 14 h.p. the modern four-cylinder is admirably suited, better, in fact, than a small six would be. A good four-cylinder will pull very well at slow speeds, prove economical to run, and at the higher speeds be in effect as smooth-running as the equivalent six if a good system of flexible engine mounting is employed. Also, there is the lesser complication in favour of the four.

In respect of the design of engines of all types it may be stated that the following should be incorporated to obtain the best results:

(*a*) A rigid construction, which is obtained by the method of casting the cylinders in a single block of special iron, integral with the top half of the crank-case.

(*b*) A stout counter-weighted crank-shaft carried on rigidly supported journal bearings and carefully regulated for static and rotational balance.

(*c*) Connecting-rods of minimum weight and maximum stiffness.

(*d*) Pistons which are a compromise between reduction in weight and the use of a metal of a quantity and quality which will assist towards the rapid conduction of heat away to the cylinder wall and lubricating oil.

The relative weights of each piston and connecting-rod assembly should be carefully adjusted so that they are all alike, and the compression space of each of the cylinders machined or die-cast to an equal volume.

Further than this, to obtain smooth running it is important that (*a*) the thickness of metal in the cylinder bores should be as regular as possible; (*b*) masses of metal at certain points in the cylinder block should be avoided owing to their liability to cause distortion under working temperatures; (*c*) the manifolds or conduit-pipes from carburettor to inlet valves should be so arranged (1) that each cylinder gets an equal supply of mixture, and (2) that the pipe is suitably warmed at certain points to avoid the deposition of liquid petrol under varying conditions of weather, temperature and throttle opening. In most modern cars the engine is freely mounted on rubber bearers, so that torque reaction is smoothed by the inertia of the unit.

**Cylinder and Piston Materials.**—It is usual to employ cast iron as the material for the cylinders. Steel cylinders with aluminium water-jackets, or constructions of a similar kind, are sometimes employed, but they are too expensive to be worth while on the car for the average market. Of recent years there has been a noticeable change in engine design, in that the older method of employing a cast-iron multi-cylinder block and an aluminium crank-case has been largely superseded by casting the multi-cylinder and the greater part of the crank-case in one piece, employing a shallow sump of aluminium or pressed steel to close in the bottom. This construction has manifold advantages; it gives an exceedingly stiff engine, and allows a really rigid support for the crank-shaft bearings, a matter of great importance for highly efficient high-speed engines intended to run smoothly. The increase in weight is not so much as might be expected, but is far more than counterbalanced by the advantages gained, one of which is reduced manufacturing costs. Pistons constructed of aluminium alloys instead of cast iron, on the other hand, have become universal, because they enable the weight of the reciprocating parts to be reduced and at the same time offer advantages in heat conductivity.

Much of the difficulty originally encountered by the adoption of aluminium alloy pistons has been overcome, chiefly by the design and disposition of the metal, by provision for expansion at certain critical points, and by the selection of more suitable alloys. It is becoming customary also to treat the surface of the piston by a tinning or an anodizing process in order to produce a harder outer skin. Although the average engine may be expected to run for a considerable time, or rather distance, before cylinder bore wear takes place, yet an occasional sample may occur in which bore wear is not good wear, in spite of the precautions taken to prevent it. The advances made in these precautions are manifold, and start with the cylinder block itself. Modern foundry methods have established a more complete control of the quality of the special alloy cast iron employed, and also over the degrees of hardness or "chill" which shall be allowed in certain areas. Then design has progressed; cylinder bores are kept separated with water spacing in between, water-jackets are carried right to the foot of the cylinder barrel, bosses and lugs for the cylinder-head studs are more carefully disposed to avoid distortion under tightening down or under heat expansion; in short, the metal is better disposed for even casting and for the avoidance of unequal expansion. In many modern overhead valve engines matters are carried further, inasmuch as the water-cooling system of the engine is arranged to grade the temperature evenly, and thus a thermo-siphon circulation is maintained in the cylinder-jackets whilst the valve seats and sparking-plugs have cool water directly impinged upon them by the pump circulation system.

Pistons made of two metals are coming into use. Aluminium, with its good heat conductivity, is used for the head and the gudgeon-pin bosses, and steel or cast-iron sleeves, suitably attached, form the skirt. In some cases the cast-iron portions carry the rings. Such pistons weigh very little more than aluminium ones, and on some cars are running with a clearance of two-thousandths of an inch.

**Some General Data.**—From a purely design point of view it will be interesting to record that for a six-cylinder water-cooled engine of normal design the stroke-bore ratio lies in the neighbourhood of 1·5 to 1, although modern practice tends towards a shortened stroke: the compression ratio is from 6 to 7, the ratio of crank-pin diameter to cylinder bore 0·5 to 0·7, the ratio of connecting-rod or length to crank radius 4 or 6 to 1, the maximum speed of revolution on which the engine will do useful work, from 4000 to 5000 r.p.m., and the best brake mean effective pressure from 110 to 125 lb. per square inch. These are figures which apply to normal engines only, and do not by any means represent the limits of what is regularly obtained in racing practice to-day. Roughly speaking, a good touring-car engine should be capable of developing as a maximum at least 3 b.h.p. per 100 c. c. capacity.

Whilst on the subject of cylinder capacity, it is worth remarking that four-cylinder engines vary in size from 2500 c. c. down to so small as 628 c. c. Owing to the growing number of small cars in this country, the most popular sizes of four-cylinder engines lie between 750 c. c. and 1500 c. c., the latter engine having proved itself capable of pulling with satisfaction a fairly roomy four-seater body at a considerable speed. It used to be quite common to find four-cylinder engines in the neighbourhood of 1000 c. c. capacity having the crank-shaft carried on two bearings only. The engine, however, was kept short and stiff, particularly in regard to the mounting of the bearings, and the crank-shaft was carefully balanced. Modern designs now have three bearings.

**Detail Work.**—Modern engines show very little departure from accepted principles, but there is a steady process of cleaning up and improving the detail work. In most cases it has become necessary to endeavour to obtain a greater output of power from an engine of given size, and this has resulted in a gradual increase of compression pressure. The higher octane petrols now in general use permit high compressions to be used without producing noticeable roughness in the engine or any marked tendency towards pinking.

In order to take advantage of the new fuels, considerable changes have been made in cylinder-head and combustion-chamber design. Chief amongst them is a general adoption of the die-cast aluminium alloy detachable cylinder head on side-valve engines. These heads permit a more rapid conduction of heat. There is a wide variation in the design of these heads, according to the size and type of engine upon which they are used. To allow for the different rates of expansion under heat of aluminium and cast iron, the stud holes in the aluminium head are given clearance, and a comparatively hard gasket is employed, which allows for the small movement.

**Overhead Valves.**—The overhead valve has made such progress that in point of popularity it has surpassed the side-valve design and come into general use. The difficulties which hindered its progress in the past have been overcome, and to-day it is reliable and trouble-free. Success has been attained by taking care of the detail work, using light rockers with properly hardened faces, and pressure-feed lubrication to the fulcrum points as well as oil leads to the ball ends and contacting faces. There are two main methods of operating overhead valves: one is to employ a cam-shaft, or cam-shafts, in the crank-case, in conjunction with tappets and push rods which engage rockers carried over the top of the cylinder. The other is to mount a cam-shaft, or two cam-shafts, over the cylinder head and operate the valves through fingers or rockers. When the cam-shaft is at the side it is driven either by spur gears from the crank-shaft, by helical toothed spur gear, by inverted teeth chain, or by a twin roller chain. This last drive, the roller chain, has been widely adopted of late; it used to be arranged triangular fashion to include the dynamo, but nowadays dynamos are mostly driven by endless rubber belts. Properly installed, it runs with very little noise. The drives for overhead cam-shafts are more varied in type. One method is to employ a long twin roller chain reaching up to the overhead shaft and kept at the correct tension by a long spring blade bearing along one side. In order to render cylinder head removal easy, on some designs an extra spur gear is provided between the upper sprocket of the chain and the cam-shaft end, so that the cam-shaft can be lifted bodily away with the head without disturbing the chain, the timing being marked on the spur gears. Another means of detachment employed is to use a flange to couple the cam-shaft end to the sprocket, and to provide in the casing of the

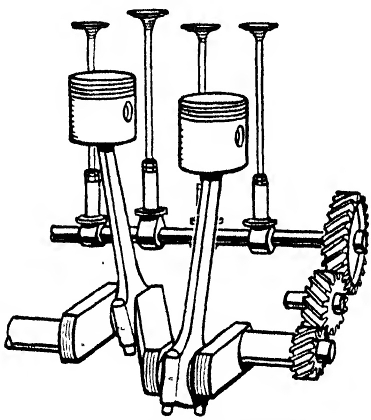

Fig. 6.—Side Valve Gear Drive

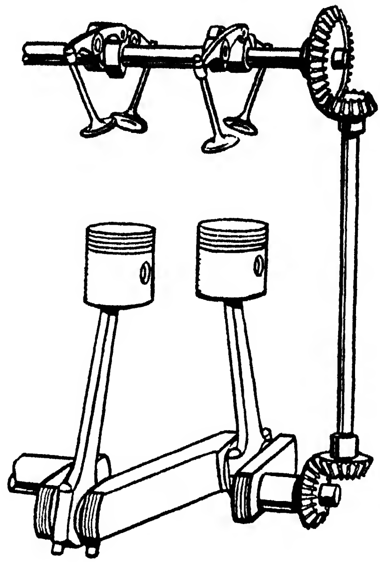

Fig. 7.—Overhead Cam-shaft driven by Bevel Gears and Vertical Shaft

timing chain a projection upon which the sprocket can rest when detached, so that the chain cannot free itself from its lower sprocket.

In most cases, however, overhead valves are operated by the side cam-shaft and push-rod principle, which has the advantages of a compact drive for the cam-shaft, and the obviating of any need to disturb the

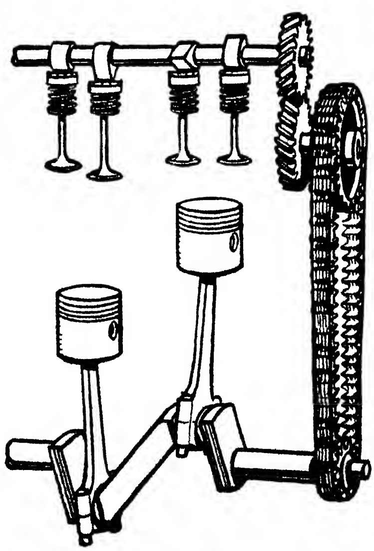

Fig. 8.—Overhead Cam-shaft driven by Chain, with Gear Intermediate to allow Easy Removal of Cylinder Head

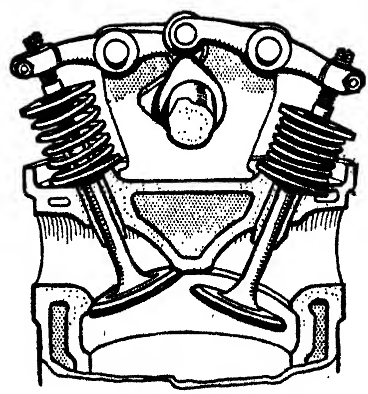

Fig. 9.—Central Overhead Cam-shaft operating through Rockers. Valves placed at an Angle in a Hemispherical Combustion Chamber

timing when the cylinder head has to be removed for decarbonization.

An ingenious and also very practical arrangement of inclined overhead valves is to be found in the Riley engine. Cylinder block and crank-case form one casting, and in the case there is a cam-shaft on each side, carried as high up as possible, and driven by helical toothed gears. Overhead valves are set at 90° in hemispherical machined combustion chambers in a detachable cylinder head, and over this are two aluminium cover boxes containing light rockers which are engaged by fine push rods, like stout knitting needles, running only a few

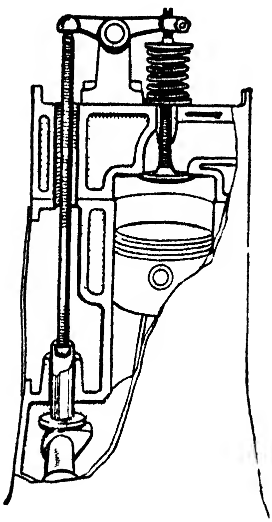

Fig. 10—Push Rod and Rocker Operation of Overhead Valve

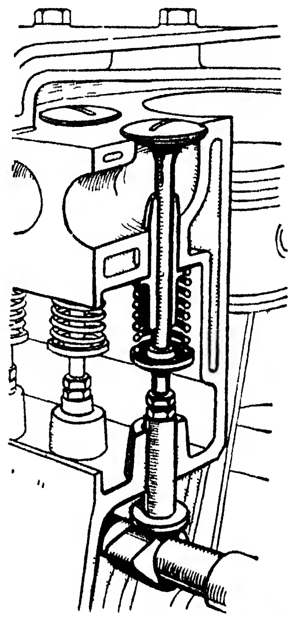

Fig 11—Side Valve operated through a Mushroom-ended Tappet

inches downwards to meet the tappets above the cams. This design combines efficiency with ease of manufacture, and also accessibility.

On engines of larger capacity it is usual to employ a single side cam-shaft, push rods of steel or duralumin tube, and rockers of compact size carried on a common spindle which is hollow, and is used also as a channel for oil to be led under pressure to the rocker bearings, and afterwards through drillings in the rockers to the ends. (See fig. 13.)

On side-valve engines, and on those that have push rods and overhead valves, it is usual to interpose a tappet between the cam and the valve extremity, or the push-rod extremity. In the case of side valves these tappets are made adjustable so as to set the correct clearance; on push-rod designs the adjustment is usually provided on the rocker, where it is easy to reach. Modern practice in regard to the cam follower or tappet is to employ a hollow drum of chilled cast iron, which is very durable. Cam-shafts are either left open to the interior of the crank-case to obtain lubrication for the cams, or they are surrounded by oil-filled tunnels. Their bearings are lubricated under direct pressure from the main oil pump of the engine.

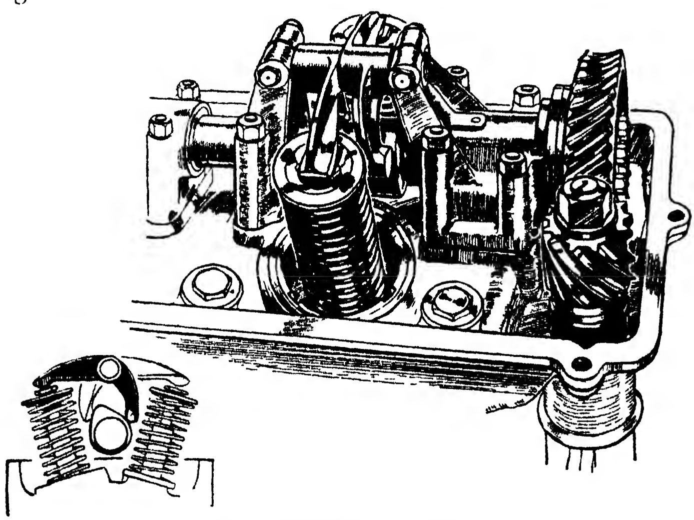

Fig 12.—Skew-gear Cam-shaft drive

Whether overhead or side-by-side valves be employed, the shape of the combustion chamber has a considerable effect on the power produced, especially at high speeds. In the case of overhead-valve motors the hemi-

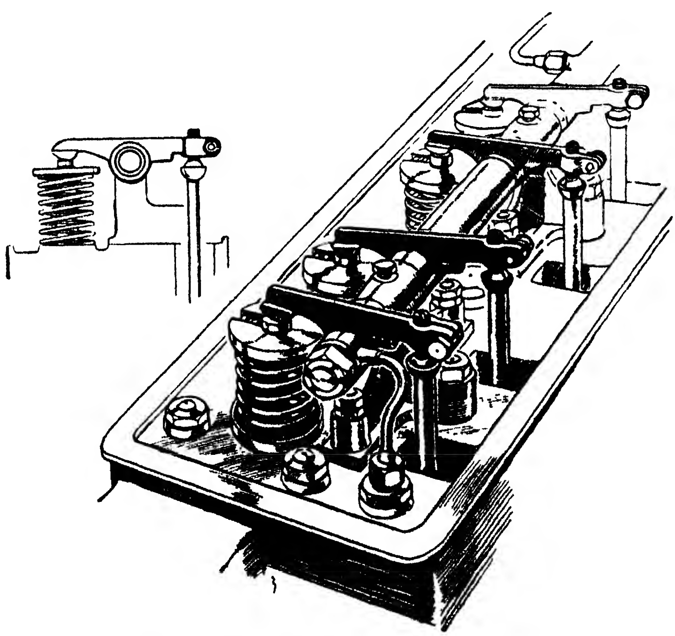

Fig. 13.—Push-rod Valve Gear with light rockers

spherical (or approximately hemispherical) head usually proves to be more efficient than the flat-headed type, but it involves inclined valves which, though excellent in many ways, often necessitate a more expensive type of valve gear and almost invariably preclude the use of push rods.

As regards side-by-side valve engines, it has become usual to arrange the head so that the greater part of the compression space lies directly above the valves, with the idea of increasing turbulence and therefore the rate of flame propagation. Also it has been found that smoothness of running and the power output are largely influenced by the proportioning of the combustion chamber and the position of the sparking-plug, which should be close to the hottest point in the chamber so as to prevent pre-ignition knock.

**Detachable Cylinder Heads.**—For the purpose of facilitating the removal of carbon deposit from the combustion space and piston head, it has become standard practice to make the cylinder head detachable. The detachable head gives the advantage of providing a water-cooled socket for the sparking-plug; moreover, it simplifies casting operations in manufacture. When the cylinder head is detachable, it is possible to machine completely the combustion chambers, and thus ensure that the compression spaces of different cylinders are of exactly equal volume, which is important if smooth running is desired.

**Water Cooling.**—Although water cooling by thermo-siphon circulation was at one time employed generally, the pump system has largely superseded it, for the latter is really the more efficient. The thermo-siphon system if properly carried out in every respect can be perfectly satisfactory, but its use necessitates a larger radiator and a considerable volume of water, the warming of which entails waste of fuel, particularly when the car is used on short journeys.

On high-efficiency engines a water pump is undoubtedly necessary; it possesses the advantage of making the cooling system more efficient and allowing the car to climb long gradients without boiling occurring. The disability of the pump system at one time used to be the difficulty of keeping water from leaking out of the bearing in which the pump-shaft was rotated. Modern design of glands, however, appears to have overcome the defect.

In thermo-siphon systems it is important that there should be a large header tank at the top of the radiator, and that the water-pipe running upwards from the cylinder block should enter this water-tank at the bottom, so that the pipe is always submerged. If this precaution is not taken, surging in the radiator may result in loss of water through the vent-pipe, and, if the level becomes too low, the circulation will cease to function and the engine incur serious damage.

One of the additions to the cooling system of the modern car is a thermostatic control, automatically actuated, which regulates the temperature of the water flow according to the needs of the moment. Recently the thermostat by-pass system has become practically universal in use.

This method of controlling the temperature has considerable bearing upon the efficiency and fuel consumption of the vehicle, besides rendering it possible for the engine to pull smoothly and strongly very shortly after it has been started up from " dead cold ".

**Systems of Lubricating.**—One of the most important features of the modern petrol-engine is the lubricating system, for without the introduction of clean cool oil in regular quantities at various points, running would become practically impossible. Lubricating systems have been developed to a fine degree of excellence so far as carrying out their actual duties is concerned; that is to say, as lubricating systems they work with a minimum of attention. When cars are becoming worn and cylinder

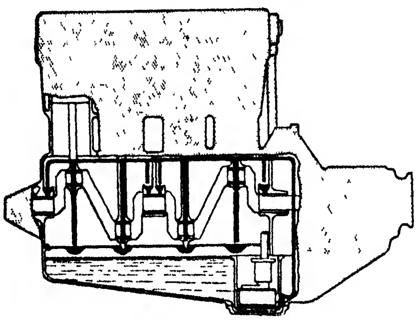

Fig. 14.—Forced-feed Lubrication

A single pump draws oil from a well in the lower part of the crank-case and delivers it to the main bearings.

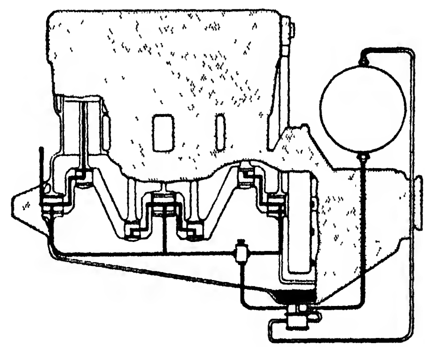

Fig. 15.—Dry-sump Lubrication

Two pumps are employed. The smaller draws oil from the tank and forces it through the bearings and to the timing gear. The oil then falls to the bottom of the engine, where it collects, and is returned to the oil-tank by a larger pump.

bores and pistons no longer fit well, too much oil escapes past the piston rings into the combustion chamber, rapid carbonization takes place, and there is a recurring necessity for stripping a large part of the cylinder assembly for cleaning purposes. Leakage of the oil into the combustion chamber is largely controlled (*a*) by correct design of the cylinder barrels to avoid warping under heat, (*b*) by using reasonably fine clearance between piston and cylinder, and (*c*) by the use of suitable piston rings, and notably a scraper ring which, as it were, peels surplus oil from the cylinder wall on the downstroke. Carbonization due to excess of oil is best avoided by careful design of the detail work of the lubricating system, so that the quantity of oil flung up into the cylinders is kept under strict control.

Two main systems of lubrication are at present in vogue, the more usual one for touring cars being that in which a pump driven from the cam-shaft picks up oil through a filter in the sump, and forces it under pressure through a second filter, which is arranged so that it can be removed easily and cleaned without loss of oil, to the main crank-shaft bearings, through a hollow crank-shaft to the big ends, and also to the cam-shaft bearings. Sometimes the oil is passed up pipes on the connecting-rods to the gudgeon-pin bearings. With overhead cam-shaft engines a separate lead takes oil through a metered orifice and delivers

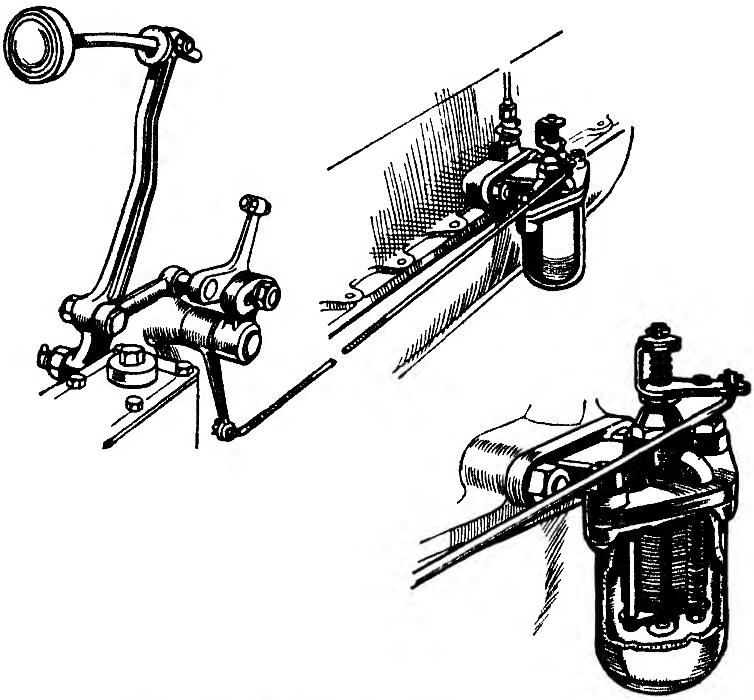

Fig. 16.—Self-cleaning Oil Filter operated from the Brake Pedal

it to the bearings of the cam-shaft and the rockers, or with push-rod operated valves to the rocker shaft, surplus oil falling back into the sump through passages provided for the purpose. In most cases the oil flung from the big-end bearings is used to lubricate the cylinder walls, pistons, and gudgeon-pins. The old system of using troughs into which the big ends dipped for their oil has become obsolete, being insufficient for modern high-speed engine requirements. The pump normally has a relief valve to regulate the pressure, which, on touring-car engines, may be anything from 15 to 60 lb. per square inch. It is noticeable that separate oil-pipes in an engine are kept to an absolute minimum, the conduits being drilled in the cylinder-cum-crank-case castings wherever possible. The second system is similar to the first, except that a larger

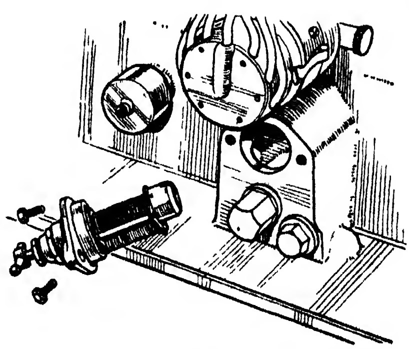

Fig. 17.—An Oil Filter which is automatically cleaned by turning an exterior handle

quantity of oil is contained in a separate cooled tank, and there are two pumps, one to draw oil from the tank and deliver round the engine system, the second to retrieve the overflow from the sump and return it to the main tank. This is known as the " dry-sump " system, and is employed chiefly on racing or sports cars. (See figs. 14 and 15.)

Engine lubrication has come in for a good deal of overhaul, and there is no doubt that some very important improvements are being incorporated. Chief amongst them is an increase in the size and delivery volume of the oil pump, in conjunction with a by-pass valve which, when the engine bearings are relatively new and unworn, short-circuits a comparatively large amount of oil, and delivers only as much as is necessary to the bearings. When the bearings are worn, however, the pump is able to make good, with a considerable margin, the wastage due to increased clearances. Another improvement is the use of an intake filter to the pump, which is arranged to float just below the level of oil in the sump, so that sludge is not picked up. In filters on the pressure side of the pump some considerable improvements have been effected. The filter takes the form of a pack of material which can be renewed easily from time to time. From the point of view of the car owner, possibly the most satisfactory improvement is the evolution of a form of oil filter which is not only particularly efficient, but possesses the great virtue that it can be made to clean itself by some automatic means, or can be cleaned by the owner through merely turning a handle (see figs. 16 and 17). Another good feature is the tendency to increase the size of oil sumps so that a much larger bulk of oil can be carried in the engine. Another point that is being taken care of is the supply of extra oil to the walls of the cylinders when an engine is first started up from cold.

**Oil Pumps and Drives.**—In the detail work of lubricating systems a great deal of interest is to be found. Of the actual pumps employed there are three main types. The one most commonly in use consists of a pair of spur-gear wheels, the gaps between the teeth of which serve as little reservoirs, and as the wheels rotate the oil is forced out through a lead pipe. The second type is a simple plunger operated either by a cam or by an eccentric, whilst the third pattern consists of a vane or plunger rotating in an eccentric casing. It is very important on all these types of pumps to arrange that they are submerged in oil, and that the oil cannot drain from them if the engine is left standing for a long period. Usually the pump is placed low down in the crank-case, so that it is below the level of the oil in the sump; but in some cases the pump is carried at cam-shaft level, and in this position it is important that loops be provided in the intake and delivery pipes so that the pump is situated in a trap full of oil.

**Carburettors.**—After passing through many stages of development, modern carburettor design has settled down to an instrument which comprises a float-controlled fuel level and a jet or jets through which the fuel is forced by the difference in pressure existing between the interior of the inlet pipe and the outer atmosphere. It might appear to be a simple proposition to supply a mixture of petrol and air in constant proportions, but this is by no means the case, for the laws which govern the flow of petrol and air differ, and at high speeds the flow of spirit increases disproportionately. If, also, the mixture is travelling at a fairly high speed and the throttle is suddenly closed, since the petrol is heavier than air it will continue to flow at too great a rate. The converse also applies, and at very low speeds it is necessary to facilitate the flow of petrol in order to obtain a mixture of sufficient strength for acceleration purposes.

In order to compensate for these various conditions, hundreds of car

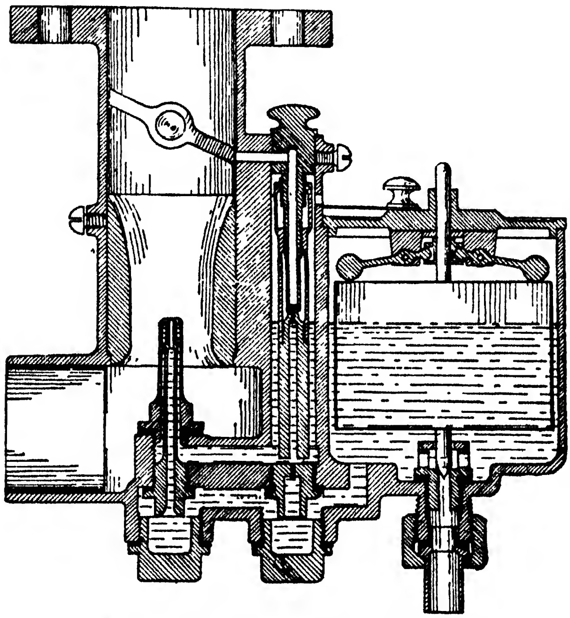

Fig. 18.—An example of the Well-type Carburettor

burettor designs have been produced; in fact, it is safe to say that this instrument has received more attention from designers, amateur and professional, than any other part of the car. Under these circumstances it is obviously impossible to describe all the various methods which are employed. Take, however, examples of carburettors commonly in use. In the Zenith type (fig. 18) two jets are employed, the flow of fuel from the float chamber to one of these being restricted by a compensating jet. At low speeds both jets are in normal operation, but as the air velocity through the choke is increased above a certain amount, the restriction in the flow to the compensating jet comes into action. Thus the main jet, being normal, allows the fuel to increase its rate of flow, whilst the compensating jet restricts its supply, so that a correct proportion of total fuel and air is maintained throughout the range. It will be seen that when the throttle is closed a well of petrol is formed above the restricting jet which supplies the necessarily strong mixture for acceleration, but as the throttle is opened this well is emptied and the petrol has to be drawn from the low level of the compensating jet.

Another type of carburettor employs the induction-pipe vacuum to control the jet and the choke size, as in the case of the S. U. carburettor. Here we find a movable taper jet needle attached to a piston moving in a cylinder which is in communication with the induction pipe on the carburettor side of the throttle. Any opening of the throttle causes a depression in the cylinder and consequently the needle and piston are raised, thus increasing the jet size and removing the choke restriction. There is a certain lag in the movement of the plunger, so that the increased air velocity past the jet during this period of lag enables a quick pick-up to be obtained. (See fig. 19.)

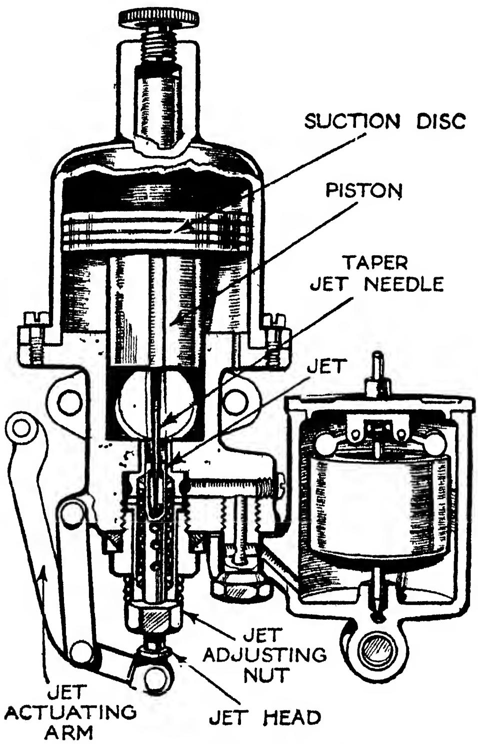

Fig. 19.—The S. U. Carburettor

A very successful carburation system has been evolved in connection with the well-known Claudel-Hobson instrument, which depends on an ingenious system of submerged jet and well in conjunction with a specially shaped throttle barrel. The well consists of a series of concentric diffuser tubes through which, under certain circumstances, a limited amount of air is allowed to pass, owing to the unsealing of a number of small holes by a drop in the fuel level in the diffuser. By means of the throttle ports the air flow is more or less concentrated on the jet as required by driving conditions, and thus the desired end is attained. These three carburettors are typical of the most usual systems, though wide variations are to be found in their application.

The driver of a car who expects to go everywhere on top gear demands that he shall be able to snap the accelerator pedal flat down on to the floor-boards at any time and expect the car to accelerate smoothly away from whatever speed happens to obtain at that moment inside the maximum range of the car. To enable him to achieve this the pump-type carburettor has come into being, and in this device a small auxiliary pump is set in action by the opening of the throttle, and causes an extra supply of petrol to fly into the inlet pipe in order to give the temporarily richer mixture necessary for clean acceleration. (See fig. 20.) There is a tendency towards down-draught carburettors, in which the fuel has a downward path from the jet to the inlet port. This scheme provides an increase in power, and a large throttle opening, but introduces problems of its own in the matter of design.

**Carburation.**—Although the main principles of carburation are unchanged, there has been much improvement in detail.

Perhaps the most important feature of recent years is the adoption of special devices to ensure easy starting, and the maintenance of a reasonable engine speed during the warming-up period.

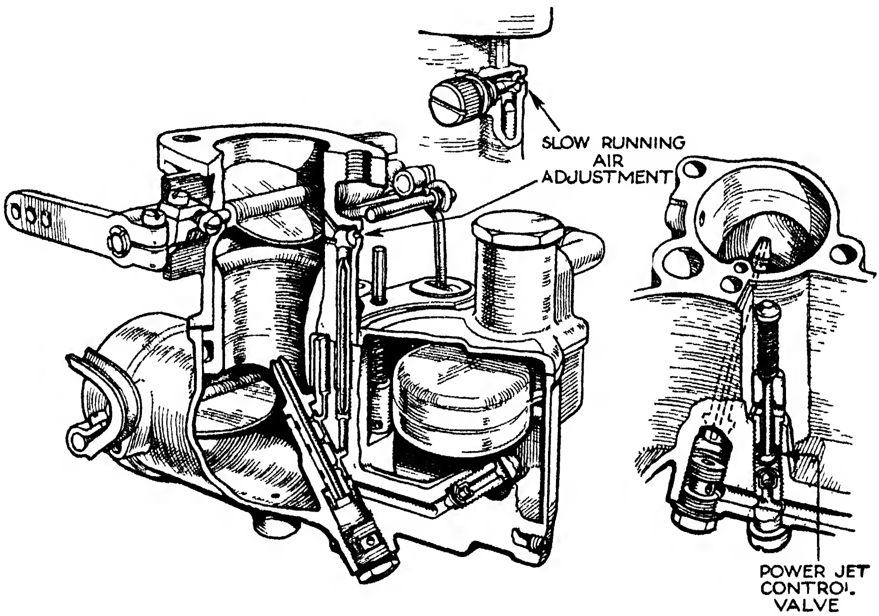


Fig 20—An example of the Pump-type Carburettor

In most cases these ends are attained by small auxiliary carburettors built into the main instrument, and operated in various ways ranging from a dashboard control to an entirely automatic arrangement over which the driver has no control.

Two very simple forms are used in conjunction with the Zenith " V " type carburettor. In both types a dashboard control serves to raise a small plunger from its seating, and thus to connect a small auxiliary carburettor with the engine through a hole on the engine side of the throttle valve. In both it is essential that the throttle shall be closed to the normal idling position.

At starting speeds the small carburettor provides a rich mixture, but as soon as the engine fires the depression is increased, with the effect, in one case, that it operates on the back of a stepped air valve, and by increasing the air flow reduces the depression on the jet. Thus a suitable mixture, both in quality and quantity, is provided for warming up.

The second type is of the dip-tube variety, in which fuel for starting is drawn from a small container. Increased depression tends to exhaust the container, and to permit extra air to flow through the dip-tube with the fuel, and thus the correct results are obtained.

A more elaborate arrangement is to be found in the S. U. thermostatically controlled carburettor. In this device, again, a small additional carburettor is used, consisting this time of a spring-loaded taper needle and disc valve operating in a body fed from the main float chamber. The operation, however, is entirely automatic, since the duct leading to the inlet pipe can be opened or closed by a solenoid, controlled in its turn by a thermostatic switch.

The method of operation is most ingenious, since when the ignition switch key is turned, a connection is made with one terminal of the solenoid, the other terminal being earthed through a thermostatic switch placed in the water-circulating system.

With these connections made, the solenoid rises, and with it a flat valve which uncovers a duct to the inlet pipe, thus bringing the auxiliary carburettor into action.

As soon as the water reaches a predetermined temperature, the contact in the thermostatic switch is broken, and current ceases to flow. Thus the core of the solenoid, and with it the flat valve, falls, closing the duct and putting the auxiliary carburettor out of action.

Another form of automatic thermostatic control of an auxiliary carburettor is to be found on certain Solex models. In this design the thermostatic element, which consists of a bi-metal strip carrying a valve on one end, is mounted on the exhaust pipe. The auxiliary carburettor is brought into action by the opening (by depression) of a small valve, the stem of which is attached to a membrane of larger diameter than the valve head. This membrane works in a separate chamber, and the air passages are so connected that the membrane will only permit the small valve to open when the thermostatic valve is open also, that is, when the engine is cold.

As soon, therefore, as the thermostatic element heats up to a predetermined figure, the bi-metal strip assumes a curvature, and the valve at its end closes the connection to the membrane chamber, and thus puts the auxiliary carburettor out of action.

**Ignition Systems.**—*Magneto ignition* at one time was generally fitted, but has now been almost entirely superseded by battery ignition. Although the magneto itself occasionally gives trouble, it is on the whole very reliable. Such defects that it may develop are, by experience, easy to trace and easy to remedy; moreover, the methods of mounting being standardized, it is always possible in the case of complete failure to obtain temporarily a spare instrument. The troubles most often encountered, even with the modern magneto, are usually traceable to the contact-breaker, of which either the bush on the rocker arm sticks or seizes, or else the controlling spring of the rocker arm may break.

*Battery Ignition.*—On most of the cars produced to-day battery ignition is being fitted, for if properly installed it gives advantages in the matter of easy starting and regularity of running at low speeds. The system includes using the lighting dynamo in conjunction with the lighting battery as a means of exciting an induction coil, part of the device consisting of a unit with a contact-breaker and a distributor accessibly mounted and driven from any convenient place on the engine. The modern battery-ignition system undoubtedly owes its inception to America. At first it did not meet with an entirely favourable reception in Great Britain, owing to the prejudices then existing in the minds of the pioneer motorists against the old-fashioned coil and accumulator set. It came at first into popularity for six-cylinder engines, in which the magneto has to be driven at 1½ times engine speed. In the modern sets the detail work is now so good that reliability is assured. A centrifugal or a vacuum-controlled advance and retard governor is incorporated. Most of the installations cause a light to shine on the dashboard if the ignition set is left switched on when the engine is not running, and the driver has no excuse for carelessness about running batteries down overnight. In order to give the batteries themselves a better chance of life, most modern cars have an automatic control of their charging rate which ensures that the input is able rapidly to make good the output.

**Starting and Lighting Equipment.**—One of the most important parts of the electrical equipment of the car is the starting motor. There are two main systems in use for lighting and starting sets: in one there is a starting motor with a separate dynamo to attend to the generation of current for the battery; in the other single-unit system a combined dynamo and motor—usually termed dynamotor—is arranged to serve the double purpose of starting the engine, and thereafter running as a dynamo for the generation of current. Owing, however, to the relatively large size of the single-unit dynamotor, the two-unit system is more favoured for use on small cars. The single-unit arrangement, however, has the advantage, when well carried out, of operating quite silently, whereas on many of the two-unit systems, where a small pinion on the starting motor is arranged to mesh with a gear ring cut on the fly-wheel, the act of starting the engine creates more noise than is evinced by the average modern car in any other respect. It is noticeable, however, that considerable improvement has been effected of late in the matter of starters, chiefly by mounting the starting motor very much more rigidly than hitherto was the case, and by taking far more care of the accuracy of the gears and of their meshing. Moreover, these drives are now normally enclosed in a suitable form of casing, which not only keeps dirt off the gears but retains the lubricant, and to a certain extent deadens sound.

It is not proposed in these notes to describe the motor, dynamo, or batteries, but rather to indicate the functions of these parts and their normal uses and abuses. As already indicated, the most usual system requires, in addition to the batteries, two separate instruments—a dynamo which is either gear-, chain-, or belt-driven from the engine and which supplies the current to the batteries; and a motor capable of producing considerable power for short periods. The latter is usually connected to the engine through a Bendix spring drive. On the armature shaft is formed a quick-thread screw which passes through a heavy gear pinion. As soon as the motor is switched on, the shaft revolves at high speed, and since the gear wheel tends to lag, it is drawn along the thread into engagement with a large gear wheel on the fly-wheel of the engine, and thus held stationary till the end of the thread is reached and it is forced to revolve with the shaft. A sudden stop would cause great shock and probably break the teeth, so a coil spring is interposed between the wheel and the end of the spindle, which allows the drive to be taken up more easily. As soon as the engine fires, the small gear overruns the armature spindle, and consequently it is forced out of engagement. The Bendix drive is reliable and seldom requires attention, but should it fail to engage, the trouble is likely to be due to thick oil congealing on the thread, and a few drops of paraffin will effect an immediate cure.

The storage batteries or accumulators require occasional attention, and should always be well charged. The correct rate of charge is usually printed on the accumulator case, and though this is regulated by the special devices on the dynamo, it is as well to take occasional readings of the ammeter to make sure that all is well with the system. The various regulating devices differ very widely and would require a special treatise for description. Most of them are of an electrical nature, though there are a few instances of mechanical ones. If accumulators are kept well charged, and the acid is not allowed to sink below the top of the plates, they will give satisfactory results for years.

If a large number of short journeys are undertaken, without any long runs, it is obvious that the batteries are likely to become exhausted, since the heavy discharge each time the engine is started is not counterbalanced by the charging time between engine stops. For this reason it is as well (in the case of some small cars) to utilize the handle for starting up the engine from the cold, particularly in winter.

Since so many slight variations are possible in the electrical installation it is not feasible to provide wiring diagrams to suit all cases.

The function of the fuse or fuses which may be interposed in the field or lighting circuits is purely that of a safety valve, for since the fuse wire is made of low melting-point alloys, only just capable of carrying the maximum normal current, any overloading causes the fuse wire to melt and thus save damage to the more costly and important parts. In replacing a blown fuse the correct wire should be employed, or there will be a risk of damage. A strand of copper wire may be employed as a purely temporary repair, but it is equivalent to screwing down the safety valve.

**Types of Friction Clutches.**—Of the clutch on the up-to-date car a great deal is demanded. It must be capable of transmitting the drive from the engine to the gear-box with absolute rigidity of torque, but at the same time it must be sensitive enough to engage from rest with complete gentleness; it has indeed to be so proportioned that, with the engine running and the car at a standstill in first speed, if the foot is released suddenly from the clutch the latter should be capable of taking up the drive perfectly smoothly and without jerk or jar. Besides this, the driven member of the clutch must be as light as possible, in order that the operation of changing gear may be rendered easy and quick, since with a heavy spinning member it takes some time for the gears to slow down sufficiently for a higher ratio

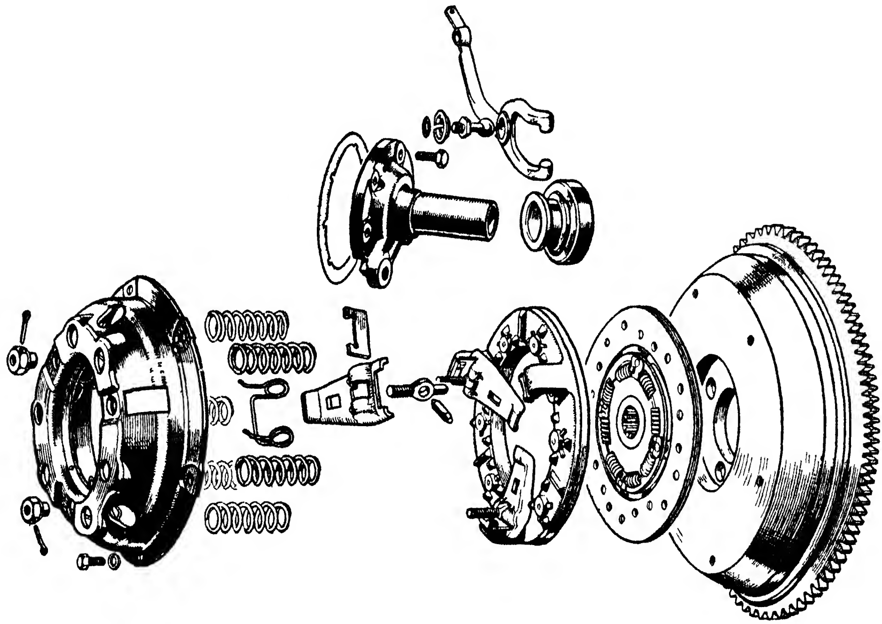

Fig 21 —Borg and Beck Clutch

to be engaged quietly. At one time the cone clutch was practically in universal use, but of recent years it has entirely disappeared, and its place has been taken by the single-disc or single-plate clutch. The reason for the change has been the adoption of unit construction and synchromesh. Also, the modern type of clutch is surprisingly trouble-free, and, having a light driven member, permits an easy and rapid gear change on cars with well-designed gear-boxes. Most of these clutches run dry, that is, without oil, but on some types the surfaces are faced not with friction linings but with cork, and these clutches are usually made to run in oil.

The multi-disc clutch, on the other hand, consists of a series of discs of relatively small diameter mounted on splines on the driven shaft; alternating between these discs is a second set attached by their outer edges to an elongation of the fly-wheel boss. Both sets of discs are free to slide, but cannot rotate without the respective members to which they are attached, and the whole assembly runs in oil. This is a very satisfactory form of clutch, but it has one drawback; that is, in very cold weather when the car is first taken out of the garage, the oil in the clutch being congealed prevents the discs from separating readily from one another, with the result that gear changing is decidedly awkward to perform without noise until the clutch is warmed up. Modern single-plate clutches are remarkably free from trouble.

**Unit Construction of Engine and Gear-box.**—The unit construction—that is, attachment of the gear-box to the engine, together with the clutch between, and the mounting of the installation thus formed as a single unit which carries also the clutch-pedal linkage and the change-speed lever—has become a general practice which again owes its inception to America. From a manufacturing point of view, unit construction presents a gain, though it is necessary that the design be properly carried out to ensure that the alignment between the plain bearings of the crank-shaft and the ball-races of the gear-box is maintained accurately after wear has taken place. It is possible, however, to arrange unit construction so that each of the three components can be reached without the necessity of dismantling the others.

Since unit construction became universal another far-reaching improvement accrued, for it became possible to mount the unit on floating rubber pads so that vibration and noise were damped out before reaching the occupants of the car. This has been developed to an excellent pitch of efficiency, and is so effective that it has become difficult to tell the difference between a four- and a six-cylinder engine, except at the very slowest speeds.

**Three- and Four-speed Gear-boxes.**—Statistics of all cars show four-speed gear-boxes are on the increase. In a manner of speaking, this is curious but understandable. It might be thought that with the steadily improving flexibility of engines, the increase in engine size and the reduction in car weight, four-speed gear-boxes were less of a necessity than they were in the days when engine speeds only ranged from 300 to 2000 r.p.m. To-day, however, an enthusiastic car driver demands not only flexibility on top gear but a hurricane acceleration as well, for this is necessary to him when endeavouring to thread a way through traffic. Given a really well-calculated ratio, a third-speed gear is of immense value in this respect, and it also comes in exceedingly useful for climbing mild hills quickly.

This has come about concurrently with the adoption of synchromesh mechanism, which is referred to later. There has also been considerable development of the mechanism for changing gear in order to make the lever lighter to handle, and to mount the lever in some position where it does not obstruct entry to the driver's seat from the left side of the car. Another refinement is the provision of a dipstick to show the level of oil in the gear-box. From America has come a widespread use of the " over drive ". This is used with a three-speed gear-box, and is similar in effect to the older-fashioned " geared up fourth ". The over drive, however, is usually contained in the rear axle, and thus may give a selection of six speeds, three normal, and three over drive. The actual changing is carried out semi-automatically; a finger-tip control is moved and the gear changes when the accelerator is released and the clutch pedal depressed. There are various other controls in existence. Hydraulic gear controls are also in use.

Gear-box design has undergone a complete revision in the last few years, and the average box of the day has four speeds, with herring-bone or plain helical teeth, always in mesh, the changing being effected by moving dog clutches. A considerable advance has been secured in quietness of running on all gears.

**Synchromesh and Free Wheels.**—There are various forms of synchromesh mechanism, but the underlying principle is the same in almost every case. Briefly, it may be explained as the synchronization of the speeds of the gear-box parts to be engaged before engagement is possible. The system is only employed in conjunction with constant mesh gears, and is normally employed for second, third, and top gears, but occasionally on first gear as well. In action, the first movement of the gear-selector lever causes small friction-clutch elements on the two gear parts to be engaged, thus tending to equalize their speed of rotation. A floating gate prevents engagement of the dog clutches or internally toothed gears until the speeds are accurately synchronized, but thereafter it is pushed aside to permit perfect and silent engagement.

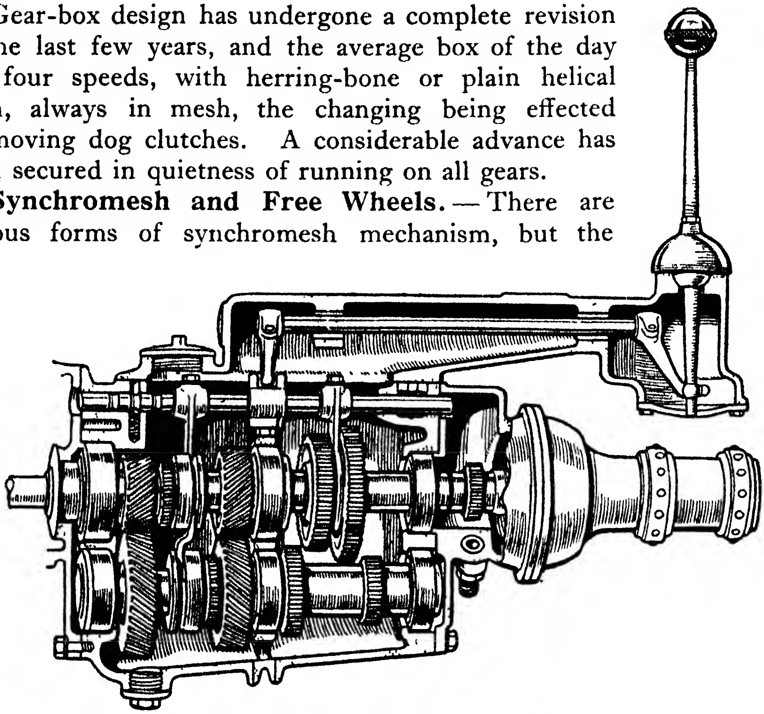

Fig. 22.—An example of a Gear-box with a " Silent Third "

One result of the general adoption of synchromesh has been the disappearance of the free wheel. From being nearly universal the free wheel has become the exception, though a large number of motorists appear to regret the fact. Synchromesh, especially the more highly developed forms of it, has rendered gear changing so easy that the free wheel from that point of view has become redundant.

**Drives to the Rear Axle.**—Two main methods are in use for conveying the drive from the gear-box to the back axle. The most common consists of an open propeller shaft having a universal joint at each end, either a fabric disc or some form of metallic or rubber-cushioned metallic joint. The open shaft saves unsprung weight, and was mostly to be found on sporting types of chassis. Of recent years it has become general, and takes the form of a tubular construction, with needle-roller bearings in the two universal joints.

In the other form of drive the torque tube extends from the centre

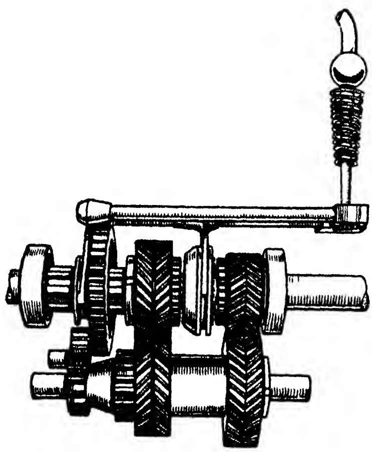

Fig. 23.—" Silent Second " Herring-bone gear Gear-box

Fig. 24.—Spiral Bevel Final Drive

of the rear axle to a crutch fork or hollow ball-joint swinging from the rear of the gear-box; down the centre of the torque tube the propeller shaft passes. When an open propeller shaft is used sometimes a separate torque member is fitted, but more often both torque and drive are arranged to be taken through the rear springs. The enclosed propeller shaft is provided with a central steady bearing within the torque tube, to prevent whirling and avoid vibration at high speeds.

**Final Drives.**—Of the forms of final drive in use the most common is the spiral-bevel gear (fig. 24). At one time straight-tooth bevels were generally employed, but they have now fallen completely out of favour on account of the more silent running of the helical-tooth variety. Worm drives are seldom used.

The form of construction used for the casing of the rear axle most generally employed is the banjo type, consisting of a drum-shaped centre running into taper-tubular ends, the casing being made from steel pressings welded together along the length. Castings are used for the front drum cover, which houses the bevel pinion and carries as well the housings for the bearings of the crown wheel. The rear cover is usually of steel, welded in place. The construction is such that when the axle shafts are withdrawn the crown wheel and differential can be withdrawn from the front. It is customary to use axle shafts of the floating type, that is to say, those with which dogs are provided at the outer extremities and engage with the hubs of the wheels, the latter being mounted on bearings placed outside the ends of the axle tube, so that the shafts take none of the axle load. In another form of construction largely used, semi-floating shafts are employed. In this case the end bearings are outside the axle-casing ends, but the central bearings carrying the final-drive gear take their part in locating the wheels and thus providing a broad bearing base for the latter. A banjo type of rear axle is illustrated in fig. 25, showing the accessibility of the final drive and differential unit.

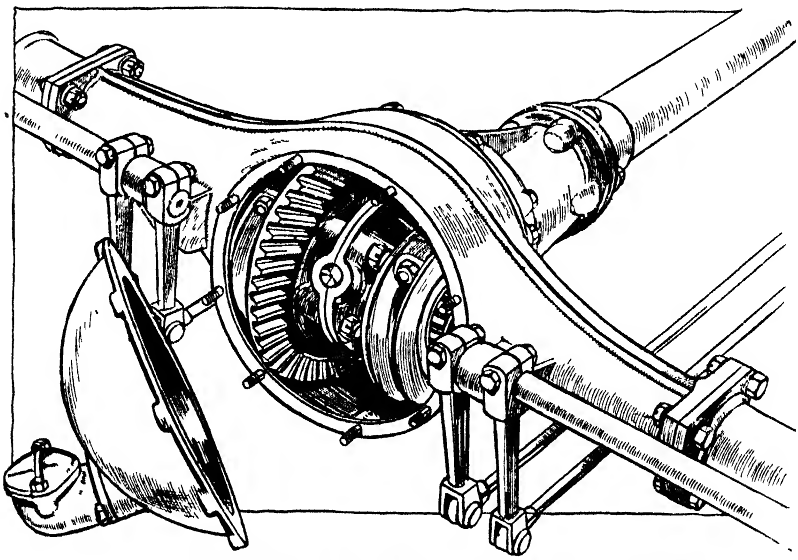

Fig. 25.—A Banjo Rear Axle showing Accessibility of the Final Drive and Differential Unit

**Recent Developments in Transmission.**—In the matter of transmission, developments of considerable importance have taken place during the past few years. There is a marked tendency towards the adoption of fluid transmissions, often, as in the case of the Daimler-Lanchester group, in conjunction with epicyclic gearing having pre-selector mechanism.

Dealing first with the fluid fly-wheel, the broad principle is that of the transfer of energy from one set of vanes to another, through the medium of some such fluid as oil.

In practice, an annular ring is formed half in the driver and half in the driven member, a small clearance being allowed between the two.

Radial vanes are disposed in each half of the annulus. Oil contained in the space is flung outwards by the driver, and on striking the blades of the driven member a part of the energy is transferred thereto, while the fluid tends to fall owing to loss of momentum. Thus there is formed a spiral column of oil passing through the two sets of vanes in a circular path.

At low engine speeds the driving force is negligible, but at a given speed, determined by design, the drive becomes to all intents and purposes solid, there being not more than 2 per cent slip.

With this device it is impossible to stall an engine on a hill, for as soon as the crank revolutions fall below the predetermined figure, slip occurs until a suitable gear ratio is engaged and the engine is enabled to accelerate again.

The fluid fly-wheel as employed by the Daimler, B.S.A. and Lanchester cars is not in itself a clutch, though it acts as such to a certain extent, in that there is no drive at low crank revolutions. Thus it is possible to stop the car by lifting the foot from the accelerator pedal and applying brake pressure, and to glide away again merely by depressing the accelerator.

The fluid fly-wheel is used in conjunction with the Wilson epicyclic gear, in which is incorporated an ingenious system of pre-selector mechanism.

By this means any of the four forward ratios, neutral or reverse, may be selected at any time by means of a small control lever, but will come into action only when a control pedal is depressed and released.

Three of the forward ratios and reverse are brought into action by means of band brakes, which are contracted on to the annuli of their respective epicyclic trains by the action of a single powerful spring, the adjustment of the band mechanism being maintained automatically. Top gear, however, is obtained by the engagement of a separate clutch.

Another form of clutch used with good results with the pre-selective epicyclic gear is the Newton centrifugal device, which also gives an automatic action.

**Four-wheel Brakes.**—Four-wheel brakes are now universally adopted, and of recent years they have been vastly improved in so many respects that they are very easy to adjust, seldom require attention, and are extremely powerful.

As a general rule four-wheel brakes are arranged so that the stopping effort between front and back is about equal, although on the better classes of car and those which are handled by expert drivers it is customary to contrive upwards of sixty per cent of braking on the front wheels and forty per cent on the rear.

It is possible to lay out a four-wheel braking set so that on a medium sized car a stop in minimum distance can be obtained without the exertion of any abnormal force on the pedal. In fact, on some cars the effort necessary to apply the brakes for anything short of an emergency is little more than the effort on the clutch or accelerator pedal. Recent improvements in cast-iron brake drums and in regularity of efficiency in brake shoe-lining materials have considerably altered the situation as regards brakes. Brake "hook-ups" have come into general use which embody straight pull rods without rocking levers, and friction-reducing wedge cams to expand self-aligning shoes. These new brakes are very efficient, fully compensated, and easy to adjust without even jacking up the wheels.

Another development on the way is differential braking, which can employ an even loading on front and rear wheels to ensure equal wear under normal light braking, but automatically increases the loading to perhaps 75 per cent front and 25 per cent rear during emergency use when the maximum stopping power is needed.

Improvements in brake gear in this direction have been such as to render unnecessary any extraneous methods of increasing pressure or applying the brakes through the agency of mechanical or other forms of relay. The servo system, as it is named, is now found on large and heavy cars. Other improvements also have affected this matter, for the efficiency of brake shoes has been enormously increased, and there are various types of shoe in use which work on the self-servo principle, meaning that the act of applying the brake causes the brake to apply itself. The efficiency of these modern shoes has been greatly increased.

The hydraulic system of brake application has come into widespread popularity, and it is capable of giving considerable braking power with very light pedal application, inasmuch as the mechanical losses are decidedly small. With the advent of independent front-wheel suspension, hydraulic operation of brakes became a further advantage, as it avoids complication of brake rods. Many detail improvements have been made in hydraulic mechanism, and also in the methods of operating the brake shoes.

Brake drums are steadily increasing in diameter and also in rigidity, the provision of stiffening ribs on the outside of the drum having become a general practice. (The ribs also assist in the dissipation of heat.) With this increased rigidity the tendency of brakes to squeak has been reduced.

**Steering-gears.**—Whereas at one time the steering-gear most commonly in use on cars consisted of a worm engaging a worm wheel or a segment thereof, nowadays there is an extraordinary variety of steering-gears to choose from. The worm-gear remains in use, but on many cars a worm and nut is employed, or a double worm and split nut, or there may be a cam-gear with a considerable degree of irreversibility, or else with an action which varies the gear ratio between the centre position and the extreme locks. Modern steering-gears have been brought to a very fine pitch of excellence, and this has been necessary because the demand is for ultra-light finger-tip steering, partly because modern congestion of parking places makes it imperative for a car to be easily manœuvred at very slow speeds, and partly because of modern high speeds and of the difficulty that has been experienced with regard to wheel wobble.

Wheel wobble has been one of the most elusive problems that the automobile engineer has had to tackle, and it came into prominence after the adoption of front-wheel brakes and of low-pressure tyres. There are many causes of wheel wobble, but the chief one is gyroscopic reaction due to changes of angle in the plane of rotation of the front wheels during passage over uneven surfaces. With the adoption of more flexible front springs and the greater amplitude of movement, the difficulty is increased, but may be countered by correct steering geometry and the use of torsion-bar couplings or of independent suspension.

**Spring Suspension.**—Suspension is still in a state of flux. Although independent suspension especially for the front has made great strides, there are still many manufacturers who pin their faith to the more orthodox solid front axle and half-elliptic springs, partly because the last named is a well-tried system which can be made extremely good if developed on modern lines. Independent front suspension usually takes the form of a transverse spring either above or below the frame level, and oppositely mounted " wishbones " or transverse links. Each stub axle and steering swivel is coupled to a spring end, and to the wishbone either above or below, so that the swivel rides vertically on approximately a parallel motion. There are of course many other types of independent springing, some of which employ large coil springs in place of laminated half-ellipties. Besides giving improved riding over really bad surfaces, a well-designed system should also give greatly improved steering.

Suspension generally is made much " softer " than it was, that is, the periodicity is slower and the amplitude of movement rather greater. Most half-elliptic springs to-day have some special form of bearing in the spring eye and shackle; either rubber bushings are used, or self-lubricating bushes, or screwed bushes which resist lateral wear. It has been found important to insulate the rear axle from the rest of the car as much as is possible, in order to prevent the transmission of road noise into the body. For this purpose the rubber bushes are used, and also in many cases rubber pads are interposed between the spring and the anchorage to the rear axle casing.

Modern springs, however, are flat, long, and wide, and practically every car from the most expensive to the cheapest has four shock absorbers fitted as part of its standard equipment. The object of the shock absorber is to damp the rebound movement of the spring, and there are many different types in use, but the hydraulic shock absorber is the most common. It has been greatly improved, and in the present type pistons are employed in place of vanes.

In the newer forms of hydraulic shock absorber considerable progress has been made in retaining working efficiency under wide changes of temperature. The shock absorber now has to do more work than formerly owing to the larger amplitude of slow-rate springs.

Another improvement of some moment in relation to suspension is the torsion-bar stabilizer. This takes various forms, but has for its object the damping of side roll or sway. In some examples a torsion bar is used to connect the shock absorbers on opposite sides of the car together. If both wheels rise or fall together the torsion bar is inoperative, but if one wheel is moved by the road surface without the other, the torsion bar causes both shock absorbers to resist the movement, and thus damps the side roll. In another form the torsion bar consists of a spring steel rod carried in rubber bushes transversely across the car. The ends of the rod are brought round at right angles parallel to the length of the car, and have their extremities coupled to the axle ends. If one wheel rises without the other, the movement is resisted by torsion in the bar. Torsion bars fitted back and front are particularly effective. The last described system has the further advantage that the ends of the torsion bar may be also used as radius rods to control the movements of the front axle under heavy braking, and to improve steering.

Rubber is coming more into use on the modern car, not only on the spring eyes, but on the shock-absorber anchorages, and also as a flexible oil-retaining cover material to protect various exposed joints in the steering-gear. (See fig. 26.)

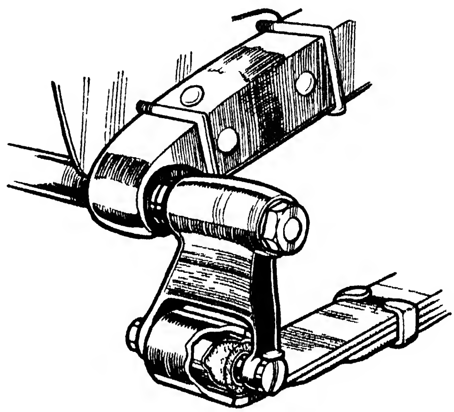

Fig. 26.—Rubber "Silent Bloc" Bushes for Spring Shackles

Incidentally, in connection with the frames and suspensions of cars, it is noticeable that a form of spring bumper is now fitted to the front and back of almost every car. These bumpers are infinite in type, and are valuable in protecting wings and relieving the severity of minor collisions.

**Wheels.**—Because they are the strongest construction, and in the eyes of most people the neatest to view, wire wheels once attained a complete ascendancy over any other type, but of late the steel disc wheel with pressed stiffening ribs has come into general use, despite the fact that the introduction of the more hardy cellulose paint, and also of high-pressure water-spraying plants for washing, has helped to remove the objection which many car owners had to the wire wheel—namely, that it was difficult to keep clean. Originally wire wheels were very expensive, but of late years their construction, and particularly the means of detachment, has been simplified and the cost reduced. One of the reasons for the general adoption of the pressed spoke or the disc wheel is that since cars had their engines moved farther forward and received larger front wings as well as a front overhang, the disc wheel became necessary to balance the appearance.

**Frames.**—The frame is a feature of the modern car which has undergone an extensive redesign, to secure the extra rigidity necessary to deal with the flexible engine mounting (as the engine is no longer a tie for the frame), with the new distribution of strength which is necessary if independent front suspension is employed, and with the increase of performance due to increased engine power. Thus the modern frame is usually of a box section built up from channels with the fourth face welded into place. It has a deep section, is plentifully provided with box or tube cross members, and may also have a cruciform central member, sometimes of box section. Great use is made of welding in the modern frame, and the old-fashioned riveted or bolted frame brackets for spring anchorages, engine mountings, brake tackle and so forth have disappeared in favour of built-up pressed-steel parts welded permanently into place. This has resulted in weight saving, in accuracy, and in ease of production.

Of recent years there has been a development of a type of construction in which the pressed-steel body is extensively reinforced, and provides the greater part, if not the whole, of the frame. This principle has variations, and in some cases there is a frame as well as that provided by the steel body, body and frame combining to provide the desired stiffness.

**Conclusion.**—During the last few years the outlook upon design of the motor-car has undergone a complete revision. At one time the engineer was hard at work trying to produce a reliable mechanism, and at the same time to make it so accessible that, if a breakdown occurred, repairs or adjustments were easily made.

Reliability can now be taken for granted with any reputable car, and the engineer is no longer busy striving after reliability and accessibility, for accessibility ceases to be imperatively necessary when reliability is assured, but is engaged in arranging and rearranging the mechanical parts of the car in such a way as to give the greatest scope to the coachwork and the equipment. For example, the modern trend is to get the largest possible size of body on a wheel-base limited to keep the car as light as possible, and to arrange so that all the seats are within the wheelbase to ensure comfortable riding. This has meant that engines, larger in size in direct relationship to the reduction in the horse-power tax, are rubber-mounted much farther forward in the chassis, whereby an improved distribution weight is secured and springs back and front are more closely matched in periodicity, so that level riding is obtained. Pressed-steel coachwork has become universal on quantity-produced cars, and all types of British-built cars have sliding roofs, control of ventilation more or less effective, luggage accommodation under cover, and a most comprehensive kind of electrical equipment which includes dipping headlamps, reversing lights, pass lights, traffic indicators, silent-running twin-bladed electric screen wipers, electric fuel pumps, and other fittings. Frames are built much closer to the ground than they were, box sections are largely used for side and cross members, the rear end of the frame is often underslung so as to give a low body platform and extra leg room without the use of sunk wells. Much care is expended on silencing coachwork, and on preventing heat and fumes from reaching the occupants by interposing rubber or composition mats completely over the flooring and around any upwardly protruding mechanism. So great has been the improvement in detail during the last few years that to-day the least expensive small car of good make is as quiet running and comfortable as the larger and more imposing vehicle, notwithstanding the fact that the road performance of the small car is of a much higher order than formerly.

*(All the illustrations in this article are reproduced by courtesy of the proprietors of "The Autocar")*

# AERO ENGINES

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# Aero Engines

## CHAPTER I

## General Considerations

The horse-power required to drive an aeroplane at a given speed and under given atmospheric conditions is proportional (*a*) to the total weight of the machine and its load, (*b*) to the total head resistance due to friction and air displacement, usually known as *drag*. The total weight includes the weight of engines and accessories, and in the case of a flight of long duration the weight of the necessary fuel forms a considerable proportion of the total weight. Some types of engine offer a higher head resistance than others, although this resistance can in all cases be reduced appreciably by suitable cowling.

It will thus be seen that a reduction in weight of engine per horse-power required by reducing the total load to be carried, and a reduction in fuel consumption per hour per horse-power, will have a similar effect, so that for long flights a heavier engine with a low fuel consumption may have an advantage over a lighter engine with a higher fuel consumption. For instance, an aeroplane with engines of 1000 b.h.p., using 0·6 lb. of fuel per b.h.p. hour, would require to carry 3000 lb. of fuel for a flight of 5 hours' duration. A reduction of 10 per cent in fuel consumption would reduce the load to be carried by 300 lb. or alternatively permit 300 lb. of additional load to be carried. Other advantages of improved engine efficiency will be dealt with later.

Many types of engines have been tried out in past years, but the increasing severity of modern conditions of flight and drawbacks such as obstruction of the pilot's view and undue space requirements have eliminated some of the earlier types. The rotary engine, for instance, which gave such excellent service in earlier years, has proved to be unsuitable for the large powers required by modern high-speed aircraft. Considerable power was absorbed in overcoming the resistance of the air to the rapidly revolving cylinders; difficulty was experienced in providing for the uniform cooling of the cylinders, the leading surfaces tending to keep cooler than the following surfaces, so that distortion resulted, necessitating the use of "obturator" rings in the pistons; difficulty was also experienced in supplying uniform quantities of mixture to the cylinders. It was practically impossible to devise any means of silencing the exhaust, and the consumption of lubricant was in general excessive. Their comparatively high weight per horse-power and higher fuel consumption than more modern types were also adverse factors.

The term "weight per horse-power" may be misleading unless the conditions of measurement of weight are specified. No real comparison is possible unless the weight "in running order" is given, which includes all *necessary* accessories together with pipes and manifolds and (in the case of liquid-cooled engines) weight of radiator and cooling liquid.

The *net dry weight* excludes exhaust branch pipes and manifolds or stub pipes, radiator connections, air intakes, oil and cooling liquid. Reduction of weight per horse-power is obtained (*a*) by increase of mean effective pressure in the cylinders; (*b*) by increase of speed of revolution; (*c*) by use of materials of low specific gravity, such as aluminium alloys, for parts not subjected to severe stresses; (*d*) by use of materials possessing increased ultimate strength and resistance to fluctuating stresses, combined possibly with low specific gravity; (*e*) by change of design of moving parts, such as push rods and valves, so as to reduce inertia stresses.

For a four-stroke single-acting engine such as is generally used, the brake horse-power is calculated as follows:

Let $d$ = bore in inches,
$s$ = stroke in inches,
$n$ = revolutions per minute,
N = number of cylinders,
$\eta$ = mechanical efficiency $= \dfrac{\text{B.H.P.}}{\text{I.H.P.}}$,
$p$ = indicated mean effective pressure (m.e.p.) (lb./sq. in.),
$\eta p$ = brake mean effective pressure (lb./sq. in.).

Then
$$\text{B.H.P.} = \frac{\cdot 785 d^2 \eta p s n \text{N}}{12 \times 2 \times 33000} = \frac{1}{1010} d^2 \eta p s \frac{n}{1000} \text{N}. \quad \ldots (1)$$

Thus if $d = 5\frac{1}{2}$ in.; $s = 6\frac{1}{2}$ in.; $n = 2800$; N = 14; $\eta p = 120$,

$$\text{B.H.P.} = \frac{30{\cdot}3 \times 120 \times 6{\cdot}5 \times 2{\cdot}8 \times 14}{1010} = 920.$$

Again, from (1),
$$\eta p = \frac{1010 \text{ B.H.P.}}{d^2 s \dfrac{n}{1000} \text{N}}. \quad \ldots\ldots\ldots\ldots\ldots\ldots (2)$$

The total stroke volume (c. in.) $= \cdot 785 d^2 s\text{N}$.

$$\therefore \text{ B.H.P. per cubic inch} = \frac{\eta p}{792} \cdot \frac{n}{1000}. \quad \ldots\ldots\ldots\ldots (3)$$

Since 1 litre = 61 c. in., we have

$$\text{B.H.P. per litre} = \frac{\eta p}{13} \cdot \frac{n}{1000}. \quad \ldots\ldots\ldots\ldots\ldots(4)$$

The brake horse-power per litre of total stroke volume can therefore be increased by increase in m.e.p., increase in mechanical efficiency, or increase in speed of revolution.

Increase in possible m.e.p. on the piston depends on (*a*) the *weight* of air taken in per induction stroke, (*b*) the fraction of this air which can be completely consumed with the aid of the fuel which mixes with it, (*c*) the fraction of the heat of combustion which is converted to work in the cylinder. It must be borne in mind that the air, which is not paid for, is as important in the generation of power as the hydrocarbon fuel, which has to be paid for, although, for reasons of cost, the tendency is to use no more fuel than can be burned completely, thus producing maximum heat *per pound of fuel*. It is, however, possible to burn the whole of the air with excess of fuel and thus obtain a greater m.e.p. For obvious reasons this is only done on occasions when extra power is required for short periods. It is necessary that combustion should not only be complete but should be complete at or near the commencement of the stroke, if maximum m.e.p. is to be obtained. This can only be effected with the aid of considerable "turbulence" in the mixture, causing rapid spreading of the flame initiated at the sparking plug.

The weight of air taken in per induction stroke depends on its pressure and temperature at the end of this stroke. The pressure can be increased by supercharging (Chap. VII). The temperature depends on the weight of residual exhaust gas left in the clearance space and the temperature of this residual gas at the beginning of the induction stroke. The quantity and temperature of the residual exhaust gases are reduced by increase of compression ratio. At high altitudes the pressure of the external air is low (the pressure decreases from 14·7 lb./sq. in. at sea-level to 8·3 lb./sq. in. at an altitude of 15,000 ft.), and although the drag at a given speed is reduced owing to the reduced density of the air, the maximum power is reduced in a much greater proportion unless the pressure inside the cylinder is increased artificially by supercharging.

The mechanical efficiency of the engine depends to some extent upon the type, in most cases being from 85 to 90 per cent. It is improbable that any appreciable increase in this value will be practicable in the near future.

Increase in revolution speed reduces the size of the engine for a given power, but for maximum efficiency air propellers should be of large diameter and run at a comparatively slow speed. The diameter is usually as great as the necessary ground clearance permits and ranges from about 7·5 to 11·5 ft., the best speeds for this range being from about 1400 to 750 r.p.m. Speed-reducing gear is therefore necessary, and the increase in weight due to this must be taken into account in estimating the value of increased engine revolutions.

Factors which limit revolution speed are inertia forces and difficulties in getting the gases into and out of the cylinder. The maximum load due to inertia forces is given by

$$P\ (\text{lb.}) = \frac{Wr}{3{\cdot}53}\left(\frac{n}{100}\right)^2\left(1 + \frac{1}{k}\right), \quad \ldots\ldots\ldots\ldots\ldots(5)$$

where W = wt. of one line of reciprocating parts (piston plus portion of connecting-rod) in lb.,

$r$ = crank radius (in.),

$n$ = r.p.m.,

$k = \dfrac{\text{connecting-rod length}}{\text{crank radius}}.$

For example, if W = 3 lb.; $r$ = 2·6 in.; $n$ = 2800; $k$ = 3·5,

then
$$P = \frac{7{\cdot}8 \times 784 \times 1{\cdot}285}{3{\cdot}53} = 2220 \text{ lb.}$$

If the cylinder diameter is 4½ in., its area is 15·9 sq. in., and 2220 lb. represents a load of 140 lb. per sq. in. of piston. An increase of 20 per cent in speed will correspond to an increase of 44 per cent in inertia load.

In order that gas may flow into the cylinder through the inlet valve or port the pressure inside the cylinder must be less than the external pressure. This difference of pressure is proportional to the square of the speed of flow, which again depends upon the size and lift of the valve and the piston speed. The size of the valve is obviously limited by the size of the cylinder, and the lift is limited by inertia considerations, so that at high speeds the *weight* of the charge is so reduced that the m.e.p. falls off appreciably under natural aspiration. Further, during exhaust the increase of back pressure necessary to discharge the gases may become very large. With supercharging a smaller inlet valve can be used, which permits of the use of a larger exhaust valve in some cases.

From 1927 to 1937 the brake m.e.p. has increased steadily from 125 to 170 lb./sq. in., and the revolution speed has increased from 1700 to 2800 r.p.m. The horse-power per litre has thus increased from 18 to 35, so that the horse-power obtainable from the same size of engine has practically doubled during the ten years. In the same time the weight per h.p. " in running order " has decreased from 1·65 to 1·25 lb. These are average results and have been exceeded in special cases. Values of over 200 for brake m.e.p. were in use on production engines in 1938.

The reason why the total weight has not decreased in proportion to the increase in horse-power is that the increased maximum pressures and increased speeds necessitate stronger and stiffer moving parts, which involve some increase in weight even with materials of higher strength. Additional accessories, such as a supercharger, also account for some of the additional weight.

**Engine Torque.**—The torque (or twisting moment) exerted by the engine on the crank-shaft is balanced by an equal and opposite torque on the engine frame and its connections to the structure, which, of course, must be designed to resist this torque.

Let $B$ = brake horse-power of engine,
$n$ = r.p.m.,
$T$ = torque (lb. ft.) due to B.

Then
$$B = \frac{2\pi n T}{33{,}000}$$
$$\therefore \quad T = \frac{33{,}000B}{6{\cdot}28n} = 5250\,\frac{B}{n}. \quad \ldots\ldots\ldots\ldots\ldots (6)$$

The torque is thus proportional to $B/n$, so that if as a result of a series of full throttle tests on an engine values of B are plotted against values of $n$,

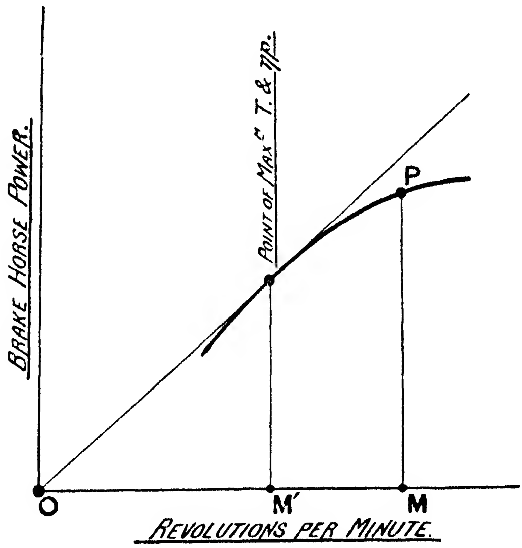


Fig. 1.—Diagram of Power-speed Curve

the torque for any speed is found from the slope of the line joining the origin to the corresponding point on the curve (fig. 1). The speed giving maximum torque is found by drawing a line from the origin to touch the curve, since this is the line of maximum slope.

Torque may be found in terms of mean effective pressure as follows:

From equation (1),
$$\frac{1000B}{n} = \frac{1}{1010}\, d^2 \eta p s N.$$

Hence, by (6),
$$T = \frac{d^2 s N}{192} \,.\, \eta p. \quad \ldots\ldots\ldots\ldots\ldots\ldots (7)$$

Equations (6) and (7) give the value of the *mean* torque, and the maximum torque will be greater than this, but in practice the inertia of the engine and flexibility of the structure prevent transmission of more than a fraction of the torque variations to the airplane structure. One method of measurement of the brake horse-power of an engine in flight involves measurement of the torque reaction on the engine frame and the use of equation (6).

**Air Resistance.**—The actual propeller thrust required to overcome the air resistance of the engines and accessories is known as the " power plant drag ", and this depends upon the projected area of the plant in the direction of flight and the arrangement and spacing of fins (for an air-cooled engine) or the effect of the radiator (for a liquid-cooled engine). It is important to note that a reduction of fuel consumption per b.h.p. hour, if accompanied by an increase in power plant drag due to rearrangement of the elements of the plant, may be of little value in reducing the total fuel consumption for a long flight. The air resistance can be reduced by suitable cowling and in some cases by placing the engine either partly or wholly inside the wing.

**Fuel Consumption.**—The chief requirements for fuel economy are: (1) complete combustion of the fuel; (2) completion of combustion at or near the beginning of the power stroke; (3) conversion of as much as possible of the heat of combustion into work.

The necessity for turbulence in ensuring complete and rapid combustion has already been referred to. Increasing the compression ratio (and consequently also the expansion ratio) will increase the thermal efficiency, but at the same time will increase the maximum pressure. According to H. R. Ricardo, an increase of compression ratio from 5 to 8 in a well-designed engine will increase the brake m.e.p. from 136 lb. sq. in. to 164 lb. sq. in., while the maximum pressure will increase from 515 lb. sq. in. to 960 lb. sq. in. At the same time the fuel consumption per b.h.p. hour will decrease from 0·485 lb. to 0·34 lb. We thus see that with an unsupercharged engine the effect of increasing the compression ratio from 5 to 8 is (*a*) 20 per cent increase in m.e.p., (*b*) 80 per cent increase in maximum pressure, (*c*) 30 per cent decrease in fuel consumption. Other advantages of increasing the compression ratio are a lower exhaust temperature and reduction of the heat to be dissipated from the walls of the cylinder.

One disadvantage of increasing the compression ratio, however, is the increased possibility of " detonation ". Detonation must not be confused with pre-ignition. If the compression pressure is increased, the temperature of the mixture at the end of compression is also increased, but even with the highest compression ratio used in practice this temperature is not high enough to cause spontaneous ignition. If, however, there is a hot spot in the cylinder, such as overheated sparking-plug points or carbon deposit, its presence in conjunction with the increased compression temperature may cause ignition to take place *before* the spark passes, and this is true pre-ignition.

Detonation, on the other hand, takes place *after* the spark passes.

The full explanation of the phenomenon is probably a complex one, but a general idea of what happens may be gathered from the following. Combustion is initiated at the sparking-plug, and as a result of the heat generated the gas ahead of the flame front is compressed, this effect being accentuated by pressure waves which are generated. If this compression raises the temperature of the unignited gas to ignition point, instantaneous ignition occurs and a pressure wave of high intensity and frequency strikes the cylinder walls, producing the familiar " pinking " sound. Detonation if allowed to continue will cause overheating of the plug points and eventually lead to pre-ignition.

If high compression ratios have to be used, the onset of detonation can be delayed (*a*) by correct design of cylinder heads, (*b*) by avoidance of hot spots in the combustion chamber, (*c*) by the use of suitable " dopes " (usually small additions of tetra-ethyl lead), (*d*) by the use of fuel having a high " octane number ".

The *octane number* of a fuel is defined as the percentage by volume of iso-octane ($C_8H_{18}$) in a mixture of iso-octane and heptane ($C_7H_{16}$) which has the same anti-detonating value as the fuel, the test being carried out in a specified engine under given conditions. A fuel having an octane value of not less than 87 is generally specified for modern aero engines.

As the altitude at which the engine is working increases the density of the air decreases, and the ordinary carburettor will deliver a mixture of gradually increasing richness. Methods of automatic control of mixture strength with altitude are discussed in Chap. VIII.

When supercharging, or " boost ", is used, as is almost invariably the case in modern aircraft, some automatic means of controlling the amount of boost according to altitude is necessary, since excessive induction pressure at ground level may result in maximum pressures sufficient to wreck the engine if maintained for any appreciable length of time. Automatic boost control is dealt with in Chap. VII.

The nature of the unbalanced forces and couples due to reciprocating parts and the problem of torsional vibrations are matters of considerable importance, but lack of space precludes discussion of these. Reference may be made to *Theory of Machines*, by Toft and Kersey, and *Handbook of Aeronautics*, Vol. II, both published by Sir Isaac Pitman & Sons, Ltd., for information on these points.

The compression-ignition engine has undoubted possibilities for large powers and long flights. Its chief advantages are: (*a*) low fuel consumption, (*b*) absence of complicated ignition apparatus, (*c*) it is easier for the pilot to maintain an economical cruising mixture at high altitudes without complicated control devices, (*d*) decreased fire risk, (*e*) for high-altitude cruising the fact that high pressure boost also involves high temperature is of small importance, and an intercooler, with its accompanying weight and drag, is not required, (*f*) cheaper fuel.

Its main disadvantage for aero engines is its high weight per horse-power, this being about 30 per cent higher than for petrol-engines. On a long flight, however, this would be counterbalanced by the smaller weight of fuel to be carried. This type of engine is more fully dealt with in the section on Oil-engines for Road Vehicles (p. 315).

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## CHAPTER II

# Engine Types

Aero engine types may be classified as follows:

(*a*) **Radial Type,** in which the cylinders are spaced at equal angular intervals round the crank-shaft, each piston and connecting-rod transmitting its power through the same crank. The power may be doubled by arranging for two sets of cylinders as above, actuating two cranks at 180°.

British representatives of this type are the engines manufactured by Messrs. Armstrong-Siddeley Motors, Ltd., and The Bristol Aeroplane Co., Ltd.

(*b*) **V Type,** in which two banks of cylinders in line are arranged at an angle of 60° with 12 cylinders, or 90° with 8 cylinders, each pair of cylinders operating on the same crank.

All of the engines manufactured by Rolls-Royce Co., Ltd., are of the 12-cylinder V type, as are a number of engines of American, French, German and Italian manufacture. The De Havilland " Gipsy Twelve " is of the *inverted* V type.

(*c*) **Broad Arrow Type,** in which a vertical bank of cylinders in line is added to the two banks constituting the V, the total angle of the V usually being 120°. This involves fewer cranks and shorter overall length for the same number of cylinders.

The Napier " Lion " engines (no longer in production) are of this type, with 12 cylinders, and so are a number of engines of French and Italian manufacture.

(*d*) **Single bank of cylinders in line.**—In this type the cylinders are usually inverted, in order that the view of the pilot may not be unduly obstructed.

The " Gipsy " engines of the De Havilland Co. are the chief British representatives of this type.

(*e*) **H Type,** in which each vertical of the H represents a line of pairs of vertical cylinders operating a separate crank-shaft, the two crank-shafts being geared together and operating a single propeller shaft.

The Napier " Rapier " and " Dagger " engines are of this type, the Rapier having 16 cylinders and the Dagger 24.

(*f*) **Opposed piston two-stroke Type.**—This is similar in principle to the Oechelhauser engine (see Gas-engines, p. 40), but in the aero engine as manufactured by the Junkers Co. each of a pair of opposed pistons operates a separate crank. There are two crank-shafts, one at each end of the cylinder, the two crank-shafts being geared to the same propeller shaft.

With the exception of type (*f*), in which the pistons act as inlet and exhaust valves, the majority of the above engines are operated by the ordinary type of poppet valve, the exhaust valves being sodium-cooled in many cases. An important development, however, is the adoption of the sleeve valve by The Bristol Aeroplane Co. for their latest types. Reasons for this will be discussed later.

Some of the factors which influence the choice of engine to be installed are:

(1) Reliability in operation for continuous periods at full load.
(2) Convenience of attachment to aeroplane structure.
(3) Accessibility for overhaul or minor adjustments.
(4) Total flying time between overhauls.

Each of the types referred to above has disadvantages and advantages in one or more of the above respects as compared with the other types, but reference to flight records shows that in respect of reliability there is little to choose between the engine types of the different manufacturers.

## CHAPTER III

# Radial Engines

The demand for ever-increasing lightness in relation to power output in the early stages of aero-engine development led to the exercise of an immense amount of ingenuity in design, and great attention was naturally devoted to the air-cooled type, owing partly to its presumed lower weight per horse-power and partly to the lessened complication and increased safety resulting from the absence of a water-cooling system, together with the diminished head resistance due to the absence of a radiator.

Although experience has shown that these considerations have very little effect on the decision to adopt radial engines for modern aircraft, the starwise disposition of the cylinders is found to be convenient for installation in the aeroplane and to give ready accessibility and ease and rapidity in engine replacement.

The number of cylinders acting on each crank is usually 5, 7 or 9, and the larger engines have two cranks with either 14 or 18 cylinders, the second set being staggered with regard to the first so that air has reasonably free access to its cooling fins. The reason for using an odd number of cylinders per crank is that a satisfactory sequence of power strokes is not easily obtained with an even number of cylinders. With 7 cylinders the order of firing would be 1, 3, 5, 7, 2, 4, 6.

The valves of these engines are actuated by multi-lobed cams revolving slowly either in the same direction as, or in the opposite direction to, that of the crank-shaft. The number of lobes and speeds of revolution may be determined as follows:*

Referring to fig. 2, let there be N cylinders, and suppose the crank-shaft to revolve clockwise, and consider firstly the left-hand diagram *a*, illustrating the case in which the multi-lobed cam is driven in the *same* direction as the crank-shaft. The black circles indicate the valve tappet rollers.

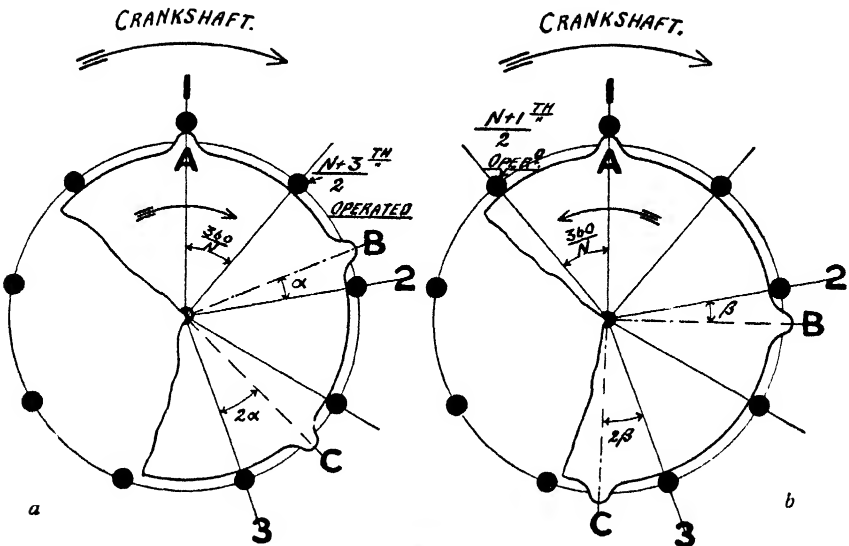


Fig. 2

Suppose valve 1 being operated by a lobe A of the cam; the next to be operated is 2, and this will require a second lobe B at an angular distance $a^\circ$ behind 2; similarly 3 will need a third lobe C; and so on.

Now the valve immediately to the right of valve No. 1 is the $\left(\frac{N+3}{2}\right)$th to be operated, and this valve is also the first to lift during the second revolution of the crank-shaft; it must therefore be operated by the lobe A. Hence while the crank-shaft turns through $360 + \frac{360}{N}$ degrees, the cam must turn through $\frac{360}{N}$ degrees; thus we have

$$\frac{\text{speed of cam}}{\text{speed of crank}} = \frac{\frac{360}{N}}{360 + \frac{360}{N}},$$

* Burls, *Aero Engines* (Griffin), p. 71, and Walker, *The Automobile Engineer*, June, 1920.

and this reduces to

$$\frac{\text{speed of cam}}{\text{speed of crank}} = \frac{1}{N + 1}. \quad\ldots\ldots\ldots\ldots\ldots\ldots (8)$$

*Number of Lobes.*—Counting valve No. 1 in the first crank-shaft revolution, $\left(1 + \frac{N-1}{2}\right)$, i.e. $\frac{N+1}{2}$ valves are lifted and each of these requires a lobe; thus the number of lobes must be $\frac{N+1}{2}$. Hence with a single-

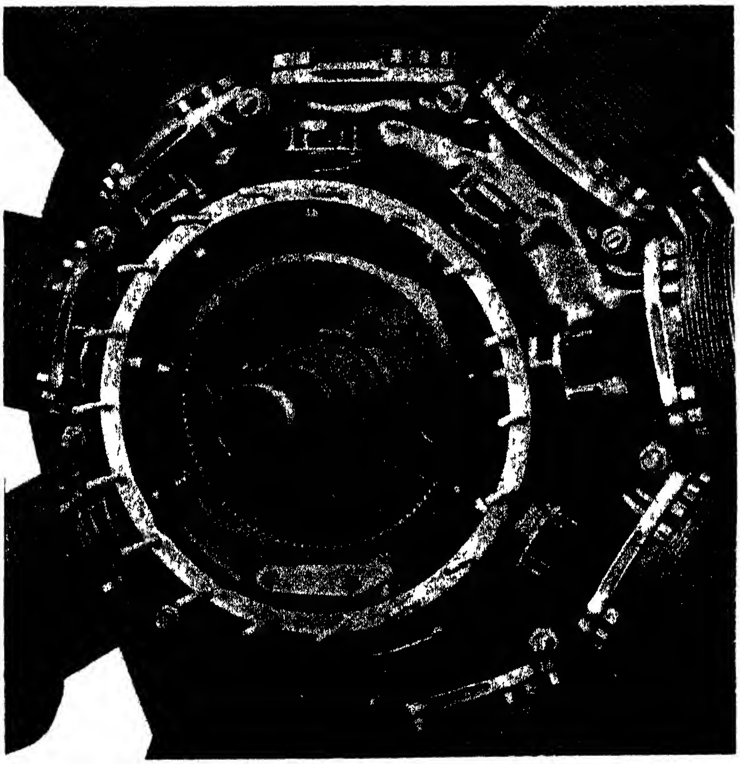

Fig 3 —Bristol Tappet Assembly—Pegasus

acting four-stroke N-cylindered single-crank radial engine, all the inlet valves may be operated by a cam having $\frac{1}{2}(N + 1)$ lobes, and driven in the same direction as the crank-shaft, but at $\left(\frac{1}{N+1}\right)$th of its speed. Similarly for the exhaust valves.

Consider next the right-hand diagram, *b*, of fig. 2; this illustrates the case of a multi-lobed cam driven in the *opposite* direction to that of the crank-shaft.

As before, valve No. 1 being operated by lobe A, a second lobe B is required by valve No. 2 at an angle $\beta$ behind 2; a third lobe C by valve

No. 3 at $2\beta$ behind 3; and so on. The valve immediately to the left of valve No. 1 is the $\frac{N+1}{2}$th to be operated, and this valve is also the last to lift during the first revolution of the crank-shaft; it must therefore be operated by the lobe A. Hence while the crank-shaft turns through an

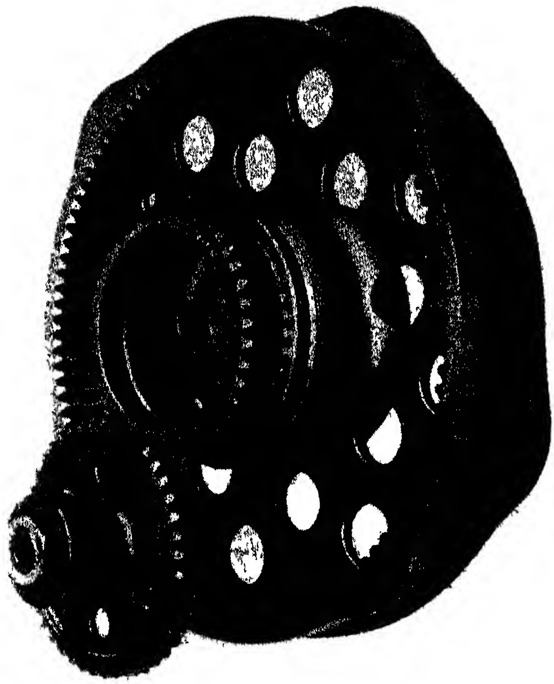

Fig. 4.—Bristol Cam-gear—Pegasus

angle of $360 - \frac{N}{360}$ degrees, the cam turns negatively through $\frac{N}{360}$ degrees, and thus in this case

$$\frac{\text{speed of cam}}{\text{speed of crank}} = \frac{\frac{360}{N}}{360 - \frac{360}{N}};$$

and this reduces to

$$\frac{\text{speed of cam}}{\text{speed of crank}} = \frac{1}{N-1}. \quad \ldots\ldots\ldots\ldots\ldots\ldots (9)$$

*Number of Lobes.*—Counting No. 1 in the first crank-shaft revolution, $1 + \left(\frac{N-1}{2} - 1\right)$ lobes are evidently required, as the valve immediately to the left of 1 is lifted by lobe A; that is, there must be $\frac{N-1}{2}$ lobes.

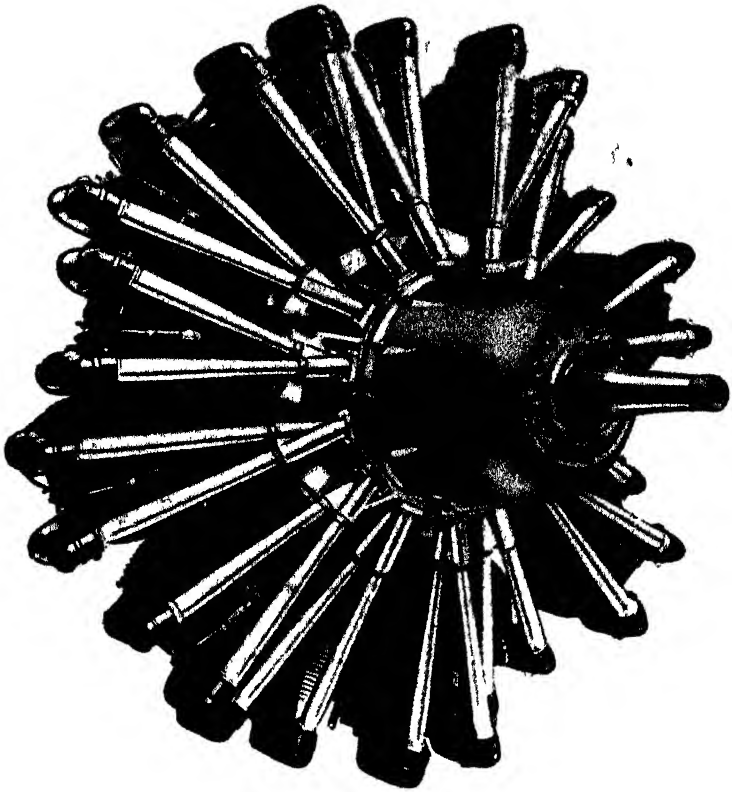

Fig 5 —Front View of A -S Tiger VIII

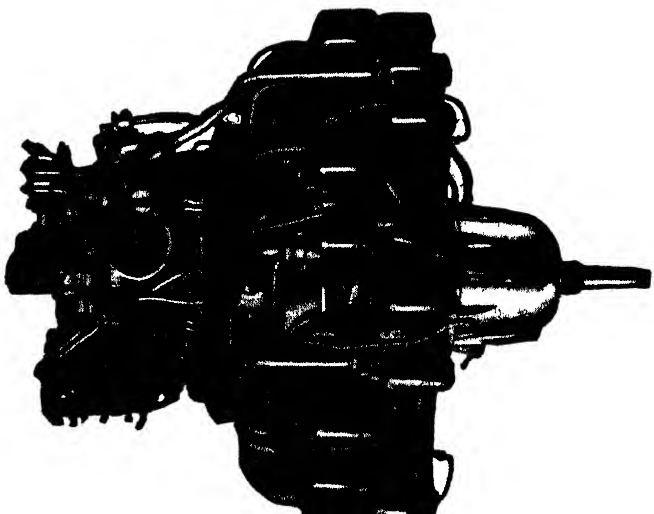

Fig 6 —Side View of A -S Tiger VIII

Hence with an N-cylindered radial engine as before, all the inlet valves may be operated by a cam having $\frac{1}{2}(N - 1)$ lobes, and driven in the opposite direction to that of the crank-shaft at $\left(\frac{1}{N - 1}\right)$th of its speed. Similarly for the exhaust valves.*

For the ordinary cases of single-crank radial engines we have:

| Number of Cylinders on One Crank, N | Cam Driven Forwards | | Cam Driven Backwards | |
|---|---|---|---|---|
| | Number of Lobes | Speed Relative to Crank-shaft | Number of Lobes | Speed Relative to Crank-shaft |
| 3 | 2 | 1/4 | 1 | −1/2 |
| 5 | 3 | 1/6 | 2 | −1/4 |
| 7 | 4 | 1/8 | 3 | −1/6 |
| 9 | 5 | 1/10 | 4 | −1/8 |

Figs. 3, 4 (pp. 393, 394) show the tappet assembly and cam driving-gear respectively of the 9-cylinder "Bristol" Mercury and Pegasus engines. There are two cams, one for induction and one for exhaust. Each has four lobes and revolves at one-eighth of engine speed.

The tappet guides (figs. 3 and 4) are located at their inner ends in a ring fixed to the crank-case wall. Their outer ends are flanged, and are secured to the crank-case by studs which also serve for the lower attachment of the push-rod casings. The cam driving-gear is coupled to the crank-shaft by means of serrations, thus facilitating accurate valve timing adjustments.

**Armstrong-Siddeley Tiger VIII Engine.**—The Tiger VIII engine has 14 cylinders arranged round the crank-case in two rows of seven.

A two-speed supercharger is enclosed in the induction-case and the engine is designed to work with a variable-pitch airscrew.

Fig. 5 is a front view and fig. 6 is a side view of the engine.

Fig. 7 is a rear view showing the arrangement of the auxiliaries. Leading particulars of the engine are as follows:

Bore, $5\frac{1}{2}$ in. Stroke, 6 in.
Compression ratio, 6·2 : 1.
Take-off horse-power, 920 at 2375 r.p.m.
Economical cruising horse-power, 590 at 2200 r.p.m.
Fuel consumption at economical cruising speed, 0·48 lb. per b.h.p. hr.
Gear ratio, airscrew/engine, 0·594.
Overall diameter, 50·8 inches.
Net dry weight, 1290 lb.

* For a still more general investigation of this problem *The Automobile Engineer* for October, 1925, may be referred to.

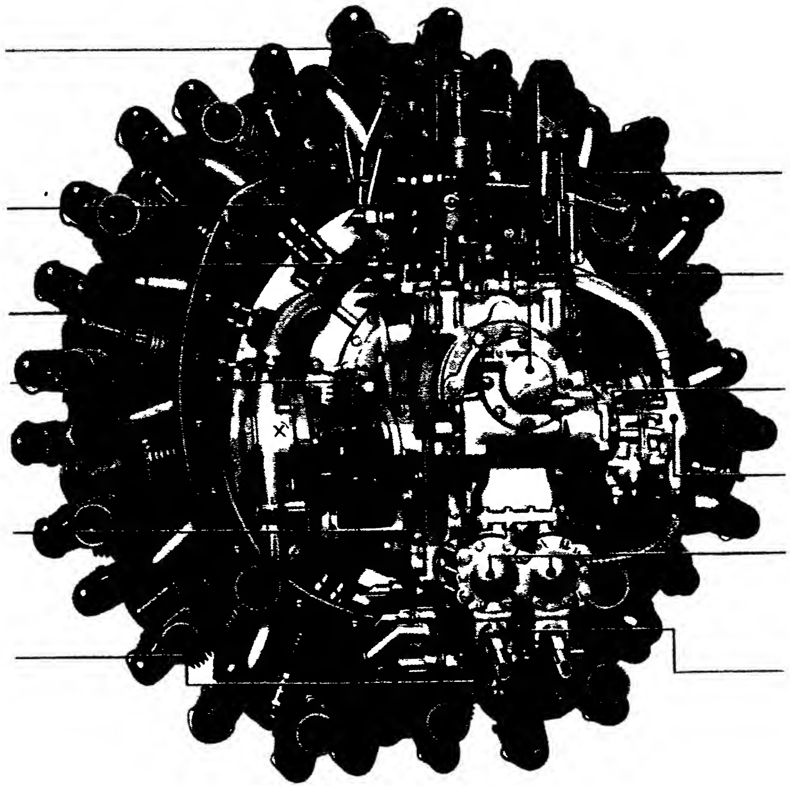


Fig 7 —A Rear View of A -S Tiger VIII showing how the Auxiliaries are arranged on the Rear Cover

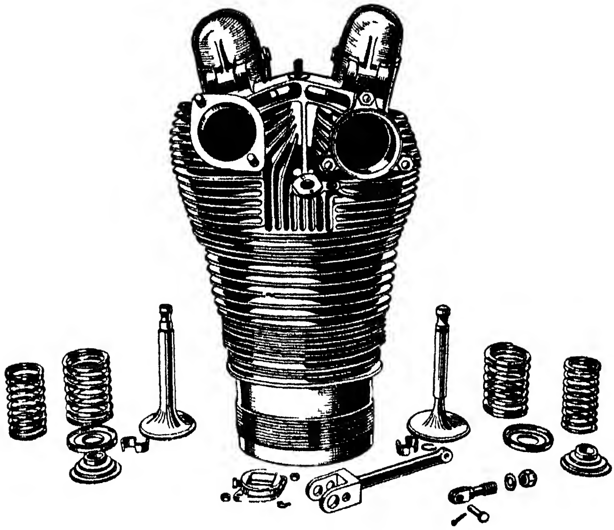

Fig. 8.—Cylinder Unit

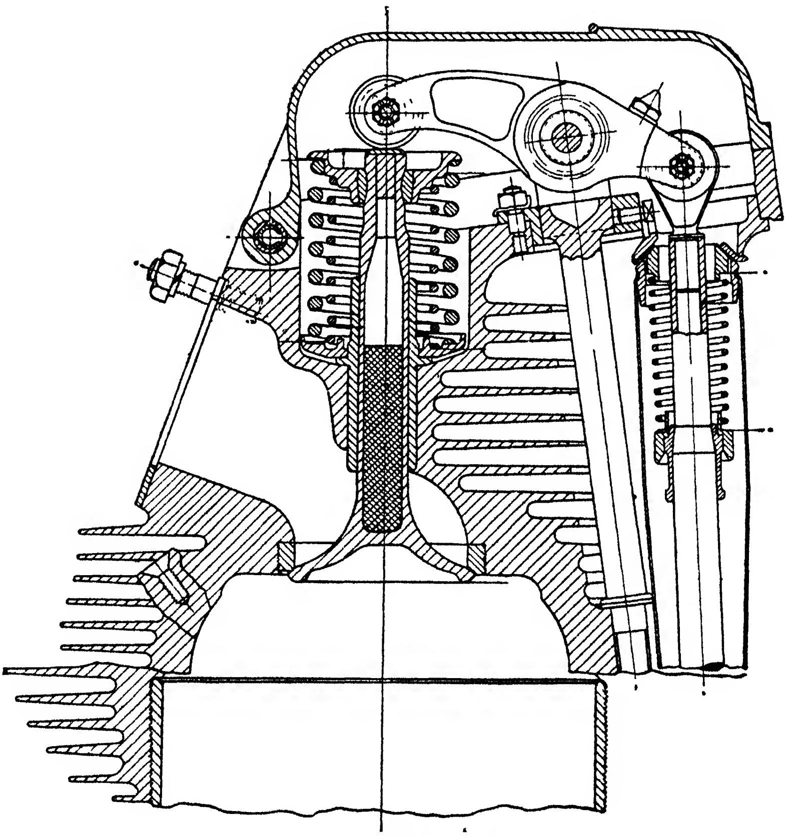

Fig 9.—Cylinder Valves and Rocker Gear

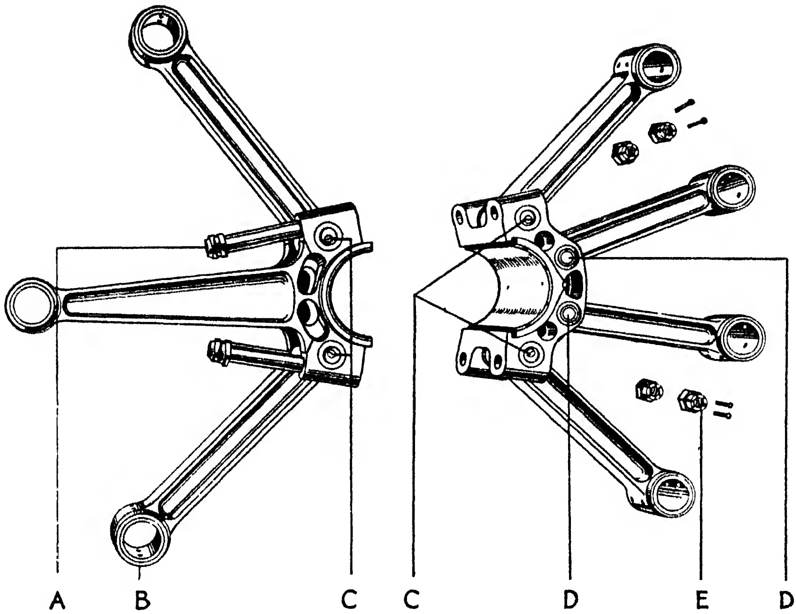


Fig 10.—Connecting-rods

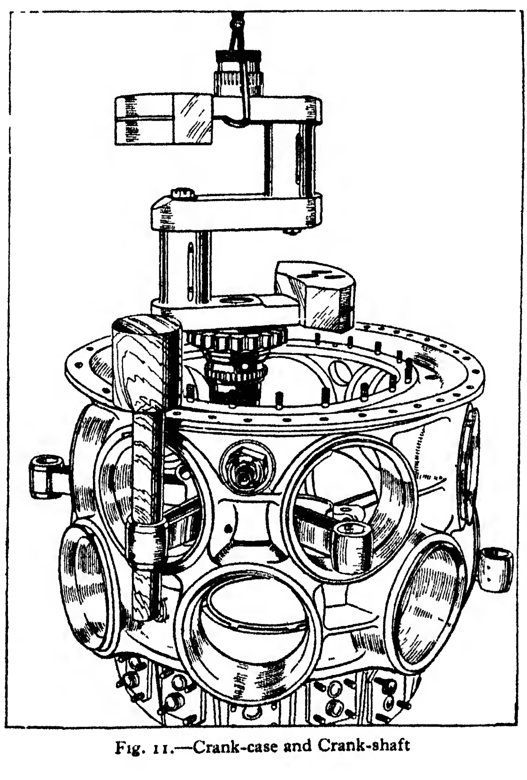

Fig. 11.—Crank-case and Crank-shaft

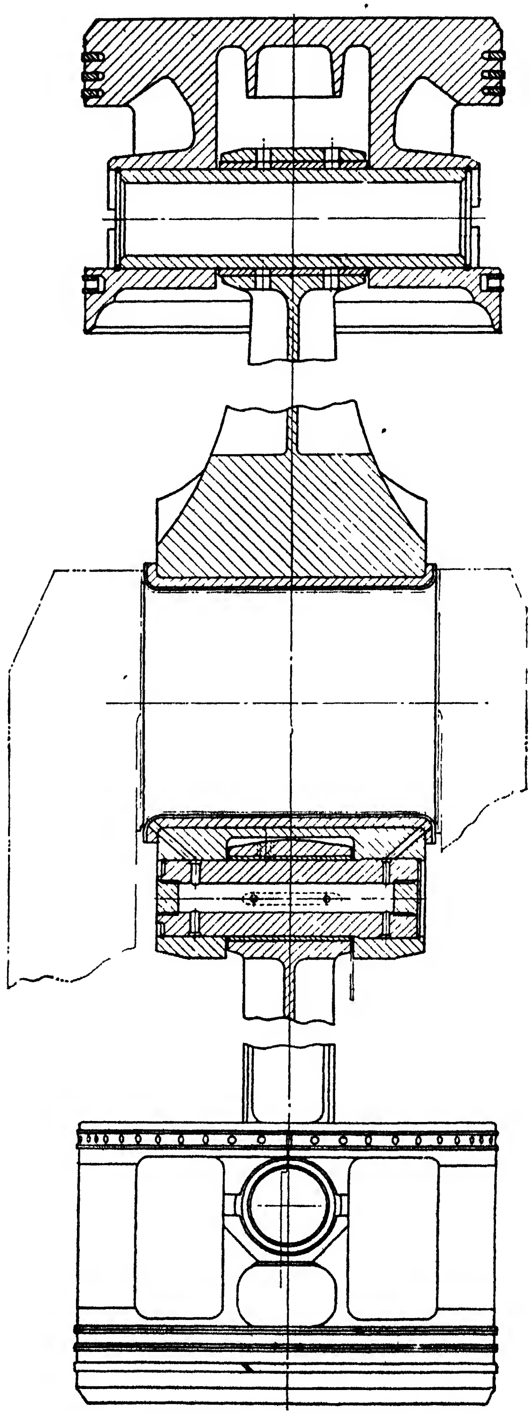

Fig. 12.—Pistons and Connecting-rods

The cylinder unit (fig. 8, p. 398) consists of a steel barrel screwed and shrunk into an aluminium alloy head, the two parts being secured by a locking-ring which also acts as a fin. Cylinder-head cooling is ensured by deep close-pitched finning.

The valve seats are of nickel-chrome manganese steel, and the exhaust valve is sodium-cooled (fig. 9). The rocker standards fitted to the cylinder head provide for compensation of valve clearance.

The master rod and auxiliary rods for the set of 7 cylinders are shown in fig. 10 (p. 399). It should be noted that the anchor pins C and D transmit the same load to the crank-pins as the main rod, but since the angular movement of each rod relative to the main rod is comparatively small a large bearing surface is not required.

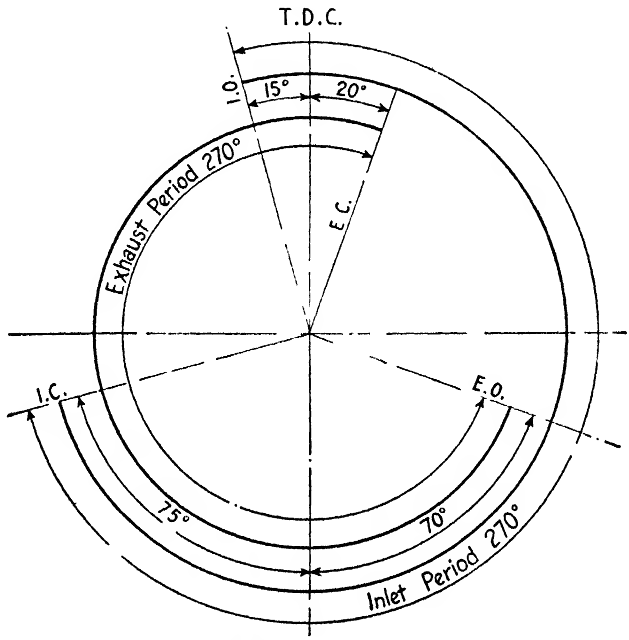

Fig. 13.—Valve-timing

The crank-case with balanced crank-shaft ready for lowering into position is shown in fig. 11. The crank-shaft revolves in two roller bearings.

The pistons are flat-topped and each is fitted with one plain and two one-degree-bevel compression rings and one double scraper, the latter being below the gudgeon-pin (fig. 12).

The cam drum revolves in the *same* direction as the crank-shaft at one-eighth crank-shaft speed. Fig. 13 shows the valve-timing with 0·01-inch clearance.

The two-speed supercharger and other accessories are described later (p. 434).

Table I (p. 402) gives particulars of other types manufactured by Armstrong-Siddeley Motors, Ltd.

# Table I —ARMSTRONG-SIDDELEY AERO ENGINES

| Engine Type and Series | Take off: Power (h.p.) | Take off: Engine Speed (r.p.m.) | Maximum (Level Flight): Power (h.p.) | Maximum (Level Flight): Altitude (feet) | Maximum (Level Flight): Engine Speed (r.p.m.) | Economic Cruising (Maximum Continuous): Power (h.p.) | Economic Cruising (Maximum Continuous): Engine Speed (r.p.m.) | Economic Cruising: Fuel Consumption: Lbs./B.H.P./Hour | Economic Cruising: Fuel Consumption: Gallons/Hour | Economic Cruising: Oil Consumption Gallons/Hour | International Rating: Power (h.p.) | International Rating: Altitude (feet) | International Rating: Engine Speed (r.p.m.) | Ratio of Airscrew Reduction Gear | Cylinders: Number | Cylinders: Bore (inches) | Cylinders: Stroke (inches) | Cylinders: Swept Volume (litres) | Fuel Octane Value | Overall Diam. (inches) | Weight—Bare and Dry (lb.) |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Tiger IX C.P. Airscrew | 880 | 2375 | 810 | 6,500 | 2450 | 550 | 2150 | ·458 | 34 | 1–2 | 805 | 6,250 | 2375 | ·594 1 | 14 | 5½ | 6 | 32·7 | 87 | 50·8 | 1260 |
| Tiger VIII * C.P. Airscrew | 920 | 2375 | 862 | 6,600 | 2450 | 590 | 2200 | ·479 | 38 | 1¼–2½ | 840 | 6,200 | 2375 | ·594,1 | 14 | 5½ | 6 | 32 7 | 87 | 50·8 | 1290 |
| 2 Speed Super † | | | 782 | 15,000 | | | | | | | 756 | 12,800 | 2200 | | | | | | | | |
| Tiger VI | 840 | 2150 | 810 | 6,400 | 2450 | 560 | 2150 | ·458 | 34·5 | 1–1½ | 760 | 5,000 | 2150 | ·594 1 | 14 | 5½ | 6 | 32 7 | 87 | 50·8 | 1180 |
| Panther X | 735 | 2250 | 752 | 4,800 | 2600 | 504 | 2250 | ·475 | 31·5 | 1–1½ | 700 | 3,000 | 2250 | ·594 1 | 14 | 5¼ | 5½ | 27·31 | 87 | 47·7 | 1068 |
| Panther VII | 590 | 2100 | 600 | 13,750 | 2400 | 417 | 2100 | ·526 | 28 9 | ¾–1⅛ | 560 | 12,000 | 2100 | ·657 1 | 14 | 5¼ | 5½ | 27·31 | 77 | 47·7 | 1045 |
| Jaguar VIc | 460 | 2000 | 490 | S.L. | 2200 | 378 | 2000 | ·515 | 25·5 | ¾–1⅛ | 460 | S.L. | 2000 | ·657 1 | 14 | 5 | 5½ | 24·78 | 77 | 45·6 | 910 |
| Serval V | 370 | 2000 | 381 | 7,700 | 2300 | 260 | 2000 | ·513 | 17·55 | ½–1 | 340 | 6,500 | 2000 | — | 10 | 5 | 5½ | 17·7 | 77 | 45·6 | 720 |
| Serval I | 346 | 2000 | 365 | S.L. | 2200 | 280 | 2000 | ·513 | 17 55 | ½–1 | 340 | S.L. | 2000 | 657 1 | 10 | 5 | 5½ | 17·7 | 77 | 45·6 | 714 |
| Cheetah X C.P. Airscrew | 375 | 2300 | 350 | 7,500 | 2425 | 230 | 2100 | ·455 | 14 | ½–1 | 340 | 7,000 | 2300 | — | 7 | 5¼ | 5½ | 13·65 | 87 | 47·7 | 694 |
| Cheetah IX | 340 | 2100 | 350 | 7,300 | 2425 | 235 | 2100 | ·449 | 14 2 | ½–1 | 310 | 6,000 | 2100 | — | 7 | 5¼ | 5½ | 13·65 | 87 | 47·7 | 635 |
| Cheetah VA | 296 | 2100 | 326 | S.L. | 2400 | 225 | 2100 | ·478 | 14 1 | ⅜–¾ | 285 | S.L. | 2100 | — | 7 | 5¼ | 5½ | 13·65 | 77 | 47·7 | 596 |
| Lynx IVc | 225 | 1900 | 240 | S.L. | 2090 | 184 | 1900 | ·513 | 12·4 | $\frac{5}{16}$–¾ | 215 | S.L. | 1900 | — | 7 | 5 | 5½ | 12·39 | 77 | 45·6 | 515 |
| Genet Major IA | 150 | 2200 | 165 | S.L. | 2425 | 119 | 2200 | ·494 | 7·75 | $\frac{3}{16}$–$\frac{7}{16}$ | 150 | S.L. | 2200 | — | 7 | 4¼ | 4½ | 7 32 | 77 | 38·15 | 327 |

* Moderately supercharged. † Fully supercharged.

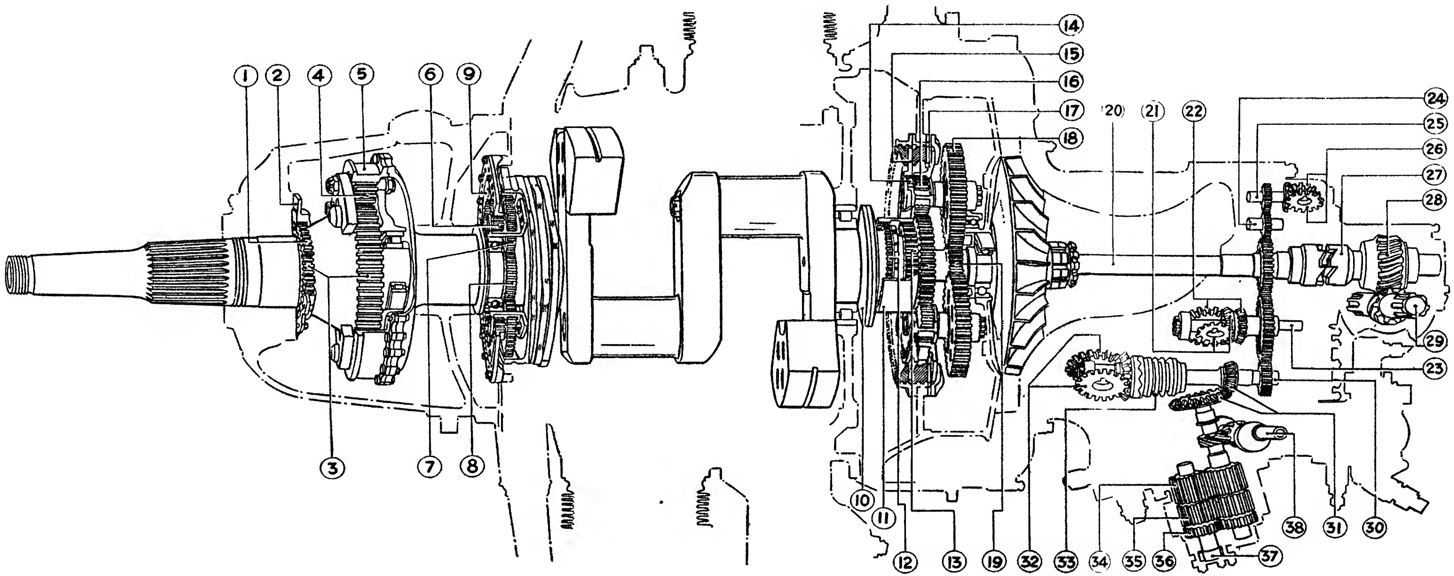


FIG. 13A—CYLINDER UNIT OF TIGER VIII ARMSTRONG-SIDDELEY AERO ENGINE

1. Airscrew shaft intermediate gear.
2. Airscrew shaft fixed gear.
3. Airscrew shaft sun gear.
4. Airscrew shaft pinion (10 off).
5. Airscrew shaft internal gear.
6. Cam drum pinion.
7. Cam drum driven gear.
8. Cam drum driving gear.
9. Cam drum stationary gear.
10. Rear spring drive.
11. Intermediate rear driving gear.
12. Rear driving gear.
13. Rear driven gear.
14. Two-speed supercharger planet cage cover.
15. Two-speed supercharger clutch plate (high speed).
16. Two-speed supercharger planet pinion.
17. Two-speed supercharger internal gear (low speed).
18. Two-speed supercharger planet gear.
19. Two-speed supercharger rotor driving gear.
20. Auxiliary drives unit driving shaft.
21. High-pressure compressor driven and driving gears.
22. Low-pressure compressor driven and driving gear.
23. Compressor driving spindle.
24. Generator driving shaft intermediate gear.
25. Generator driving shaft.
26. Generator driving gears.
27. Hand/Electric starter engaging dog.
28. Hand/Electric starter wormwheel.
29. Hand/Electric starter worm shaft.
30. Magneto driving shaft.
31. Oil pump driven and driving gears.
32. Magneto driving gears.
33. Magneto driving shaft spring drive.
34. Oil pump scavenge gears.
35. Oil pump pressure gears.
36. Oil pump auxiliary scavenge gears.
37. Auxiliary drives units scavenge pump
38. Petrol pump driving shaft.

RATIOS TO ENGINE SPEED

| | | | |
|---|---|---|---|
| Reduction gear .. .. | ·594 : 1 | Port compressor .. .. | ·97 : 1 |
| Cam drum .. .. .. | ·125 : 1 | Magnetos .. .. .. | 1·75 : 1 |
| Supercharger (MS) .. | 5·34 : 1 | Oil pump .. .. .. | ·8 : 1 |
| Supercharger (FS) .. .. | 7·96 : 1 | Petrol pump .. .. | ·25 : 1 |
| Starboard compressor .. | ·5 : 1 | Generators .. .. | 2·46 : 1 |

Facing p. 402

**Power Transmission Diagram.**—As an illustration of one method of transmitting the power to the airscrew and of transmitting the drive to the various auxiliaries, fig. 13A is given, showing the power transmission of the Armstrong-Siddeley Tiger VIII engine. The key to fig. 13A is given below the diagram and particulars of speed ratios are also appended.

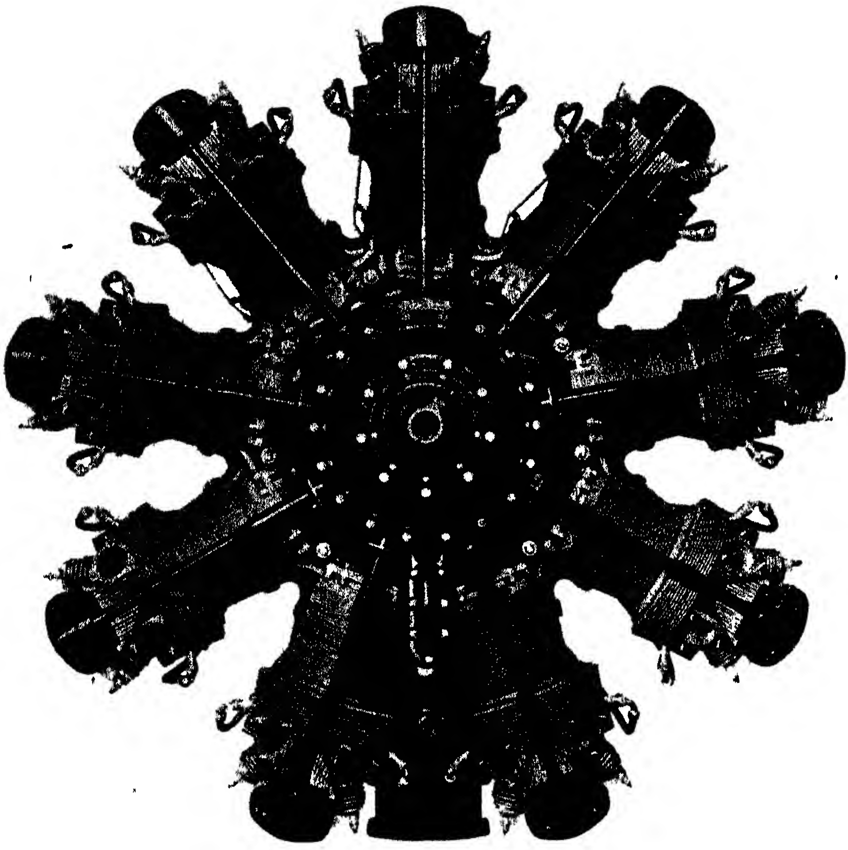

Fig 14—Front View of "Bristol" Pegasus

The use of an epicyclic speed-reducing gear for the airscrew shaft allows this shaft and the crank-shaft to remain co-axial. Reference to fig. 13A will show that the sun wheels 3 are fixed, 5 is driven by the crank-shaft, and the planet pinions 4 are connected to a frame (the "arm" of the epicyclic gear), which in turn is connected to the airscrew shaft. If B is the number of teeth in the right-hand sun wheel 3, and C is the number of internal teeth in wheel 5, the speed reduction ratio is $\frac{C}{B + C}$.

Reference to fig. 26 (p. 415) will show that in V type engines such as the Rolls-Royce "Merlin" the use of an ordinary spur reduction gear enables the engine to be dropped relative to the airscrew, thus offering less obstruction to the view of the pilot.

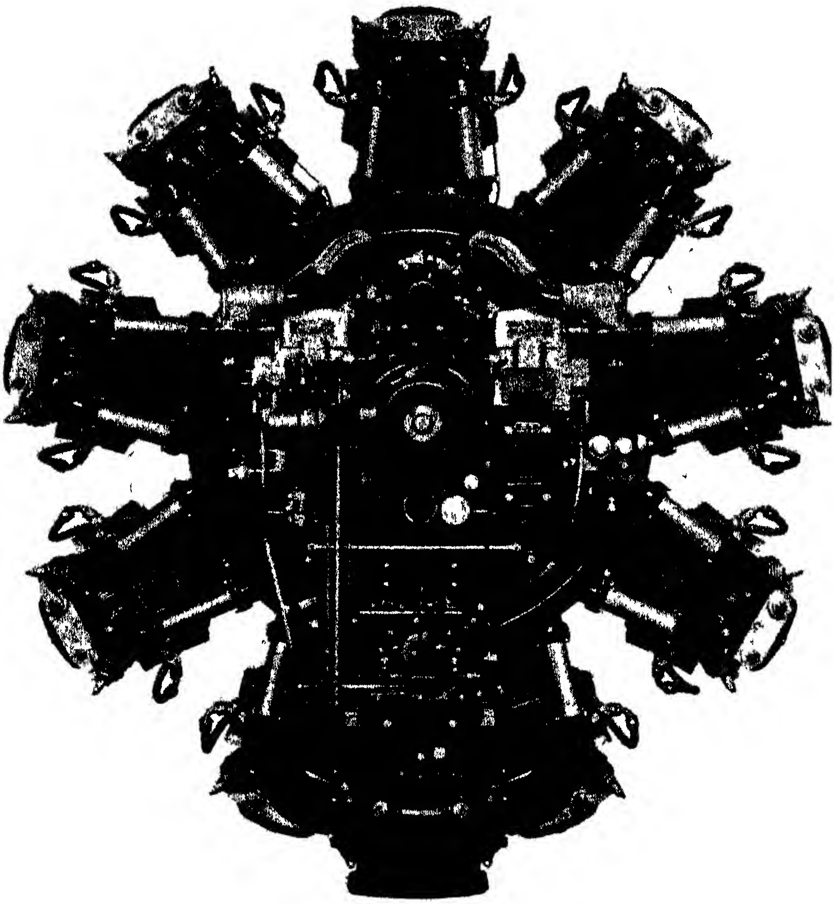

Fig 15.—Rear View of "Bristol" Pegasus

**"Bristol" Mercury and Pegasus Engines.**—These engines are similar in general construction, and figs. 14 and 15 show front and rear views of the Pegasus engine.

Each of the engines has 9 cylinders working on a single crank. Leading particulars of the engines are as follows:

| | *Mercury XII* | *Pegasus XXII* |
|---|---|---|
| Bore (in.) .. .. . .. | 5¾ | 5¾ |
| Stroke (in.) .. .. . .. | 6½ | 7½ |
| Compression ratio .. .. .. | 6·0 | 6·55 |
| Maximum take-off power/r.p.m. | 830/2650 | 1010/2600 |
| Gear ratio, airscrew/engine .. | ·500 | ·500 |
| Brake m.e.p. .. .. .. | 164 | 187 |
| Overall diameter (in.) .. .. | 51·5 | 55·3 |
| Net dry weight (lb.) .. .. | 1005 | 1030 |

The improvements achieved with the same cylinder capacity, from the first " Bristol " Jupiter in 1920 to the latest Pegasus, may be summarized as follows: The output per litre of cylinder capacity has been increased by 150 per cent and the gross brake m.e.p. from 112 to 188 lb./sq. in. Maximum crank-shaft speeds have risen from 1625 to 2925 r.p.m. Notwithstanding the addition of gearing, supercharger, and the extensive range of

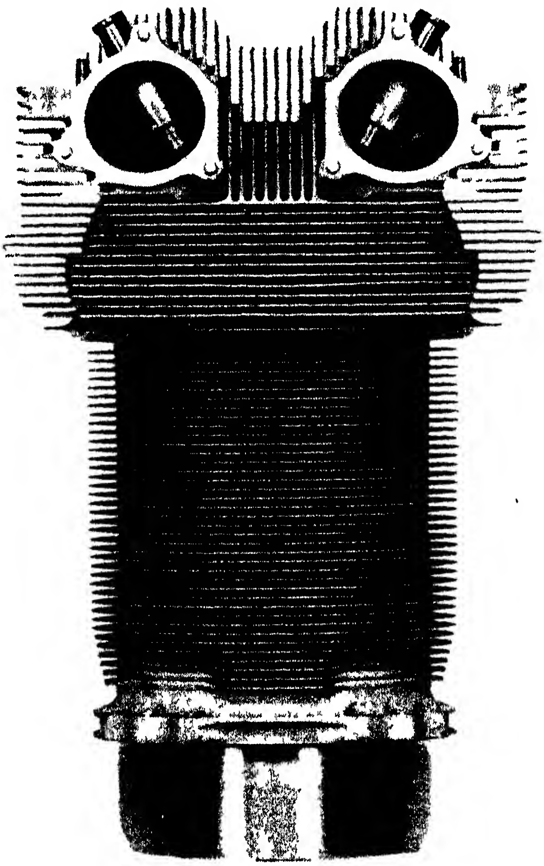

Fig. 16.—Cylinder of " Bristol " Pegasus

modern accessories, the specific weight has been reduced by 43 per cent. A saving in fuel consumption of 25 per cent has also been effected.

Leading features of the new series are as follows:

Cylinder heads machined from aluminium alloy forgings, giving much greater uniformity of structure and perfection of finish than is possible with a cast head. Cylinder barrels of special alloy steel, with nitrogen-hardened bores. Large cooling-fin area, 150 to 180 sq. ft., according to type (fig. 16). Exhaust valves sodium-filled and stellite-faced. Valve seats

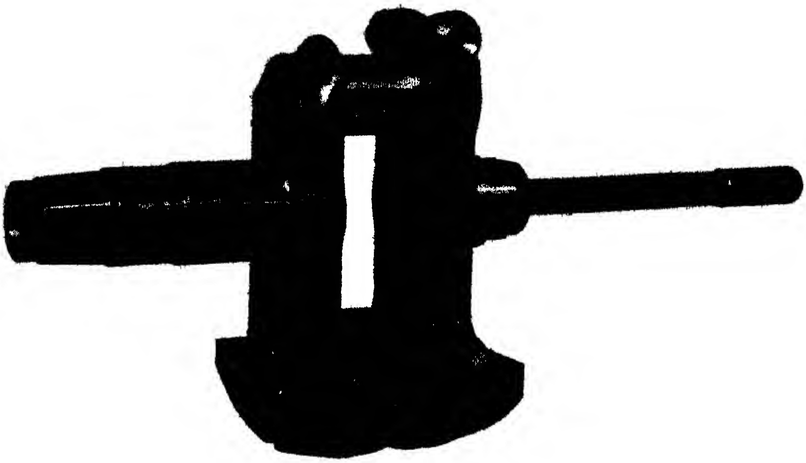

Fig 17 —Crank-shaft of Bristol Pegasus

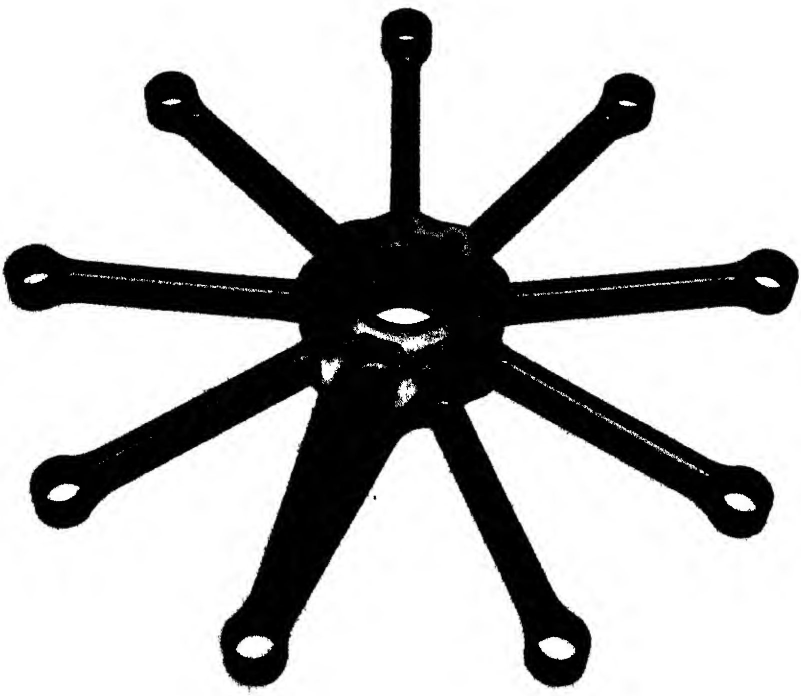

Fig 18 —Connecting-rod Assembly of "Bristol" Pegasus also stellited. Valve-operating mechanism incorporating automatic clearance compensation and automatic lubrication. Crank-shaft (fig. 17) and connecting-rod assembly (fig. 18) of the one-piece master rod and coupled crank-shaft type. Fully floating bush in big end bearing. Cylinder-temperature measuring equipment for permanent use in flight.

**"Bristol" Sleeve-valve Aero Engines.**—The single sleeve-valve engine was invented independently and simultaneously in 1909 by the Scottish engineer Peter Burt, of the Argyll Co., and the Canadian engineer James McCollum, and is generally referred to as the Burt-McCollum type.

The simplicity of action and enlarged possibilities of the new type prompted a considerable amount of experimental work which was checked by the outbreak of war in 1914. Any difficulties experienced were mainly due to difficulties in designing a reliable drive for the sleeve at high speeds and obtaining an effective seal of the sleeve in the cylinder head. In the case of the air-cooled engine adequate cooling of the cylinder head also presented a difficult problem. The present satisfactory stage of development is due to the later researches of H. R. Ricardo and the enterprise of the Bristol Aeroplane Co. and their chief engineer, Dr. A. H. R. Fedden.

Following prolonged research on single-cylinder units, the first complete "Bristol" sleeve-valve engine, a 9-cylinder air-cooled radial engine of 24·9 litres capacity, was designed and built in 1932, completing its official trials with great success in the following year. This engine was named the Perseus, and with further development has become the first sleeve-valve aero engine to be manufactured in large quantities. A similar engine on a smaller scale, the 15·6 litre Aquila, was first type-tested in 1934, with equally satisfactory results, and has since been actively developed for medium-sized commercial aircraft.

The potential advantages of the sleeve-valve for high-output double-bank radial engine design were also apparent, and in 1936 the Hercules 14-cylinder radial sleeve-valve engine of capacity 38·7 litres made its appearance. It is (1939) the most powerful type-tested British aero engine.

After the most thorough endurance and overload testing the sleeve-valve engine already occupies a leading position in the development of both military and commercial aviation, and several important new types of British fighting aircraft will have these engines as standard equipment. After extensive flight trials on regular passenger services, Imperial Airways have specified "Bristol" sleeve-valve engines for their new flying boats.

**Sleeve-valve Operation.**—The internal construction, up to and including the piston and connecting-rod assemblies, is based upon the firm's established radial engine practice. Within the crank-case cover, however, in place of the usual poppet-valve cam-gear and tappet assembly, there is a train of spur gears driven by the crank-shaft, by which rotation at half-engine speed is given to a series of small sleeve-valve operating-cranks. These have their bearings in the crank-case and the front cover respectively, and their crank-pins protrude rearwards inside the crank-case front wall. Each crank-pin engages, through a spherical and sliding coupling, with the lug of its appropriate sleeve-valve, which is a steel tube "tele scoping" between the piston and cylinder (fig. 19).

Owing to the rotation of the sleeve crank-pin at half-engine speed the sleeve-valve receives a motion which combines reciprocation and partial rotation. This combined motion is such that any given point on the sleeve describes the path of an ellipse wrapped round the circumference of the sleeve, and it completes this closed circuit once every two revolutions of the crank-shaft.

Four specially shaped ports are disposed around the circumference of the sleeve, near the top. At the correct points of the cycle for inlet and exhaust these sleeve ports, as a result of the motion of the sleeve, traverse similar ports cut in the walls of the cylinder, progressively enlarging and

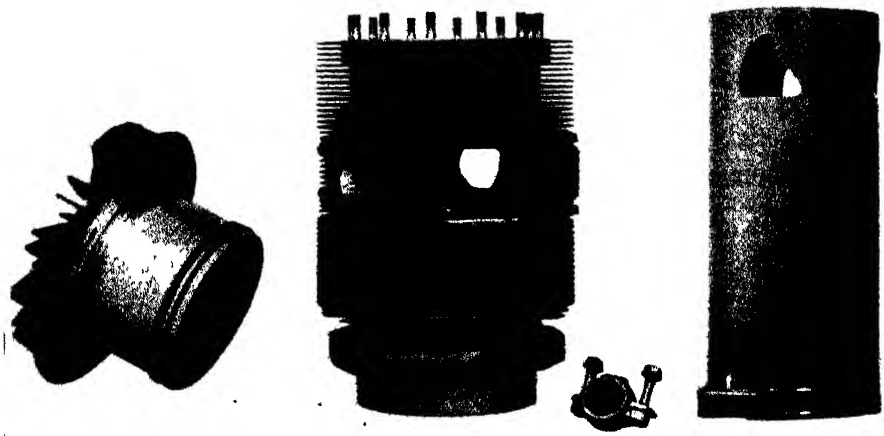

Fig. 19.—"Bristol" Sleeve-valve

then closing the resulting passages, from within the sleeve to the inlet or exhaust manifold as required.

The motion of the sleeves throughout the cycle is smooth, sliding and positive at any speed as compared with the hammer-and-anvil action of poppet valves controlled by cams and springs. The opening and closing characteristics also are such that higher volumetric efficiency and improved scavenging are secured.

During the high-pressure portions of the compression and firing strokes the sleeve is at the top of its path and its ports have risen above the level of two sealing-rings in the cylinder head (or "junkhead", as it is termed in these engines). It will be seen from fig. 19 that the junkhead extension, somewhat resembling an inverted piston, protrudes into the bore of the sleeve sufficiently for this purpose. The two sparking-plugs are fitted in the centre of the smooth, slightly convex face of the junkhead, which forms the roof of the combustion chamber.

It will be clear that, since the junkhead contains a fairly deep pocket corresponding to the extension which enters the sleeve, the bottom of this pocket would ordinarily be out of the direct flow of the cooling air. By an ingenious arrangement of finning and cowling, however, an air current is continually induced through it. A small splayed cowl is fixed to the top of each head and has rubber edges forming a seal with the ring cowling of the engine. Air is deflected by this cowl down into the pocket, over the cooling fins formed therein, and out at the rear of the head.

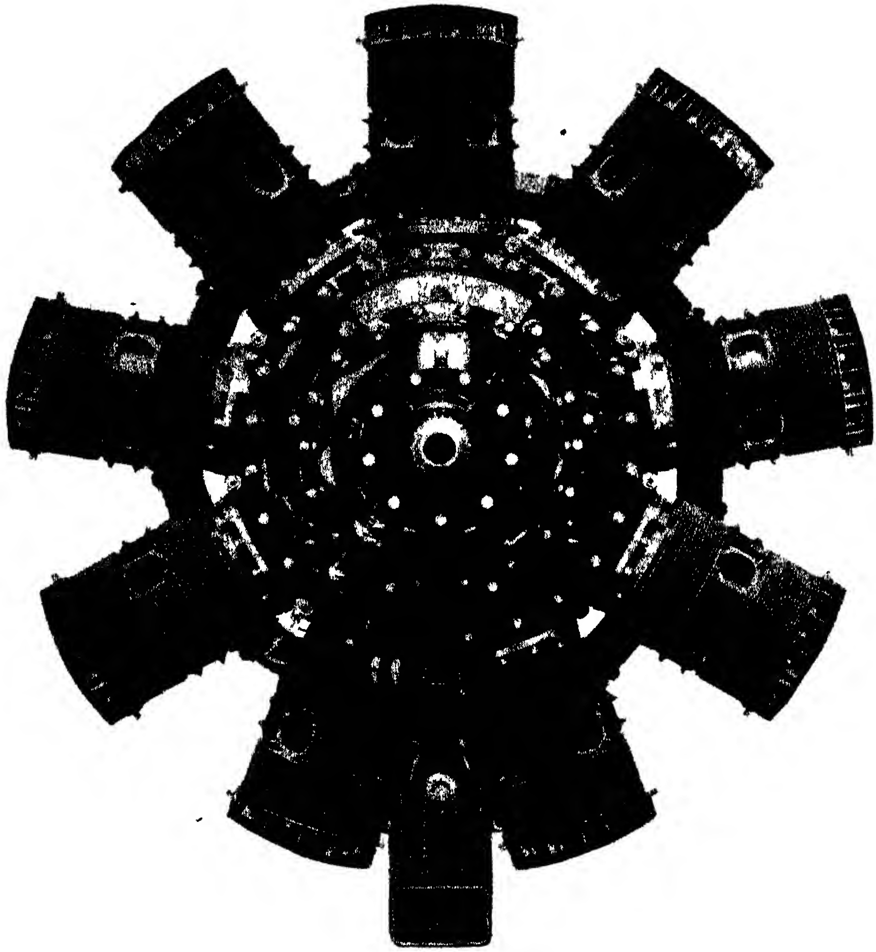

Fig 20 —Front View of "Bristol" Perseus

The cylinders are machined from light alloy forgings; the junkheads are die-cast in a similar material, and are secured to the cylinders by studs and nuts. The induction-pipes each have a manifold which embraces half the diameter of the cylinder. The latter has three inlet ports which are progressively served by the sleeve ports. The two exhaust ports of each cylinder are connected by short pipes to a front-exhaust manifold of normal type.

Fig. 20 shows a front view and fig. 21 a rear view of the " Bristol " Perseus engine. The induction-pipes are shown in the rear view, but the exhaust-pipes are omitted from the front view. The clean appearance due to the absence of external valve-gear will be noted.

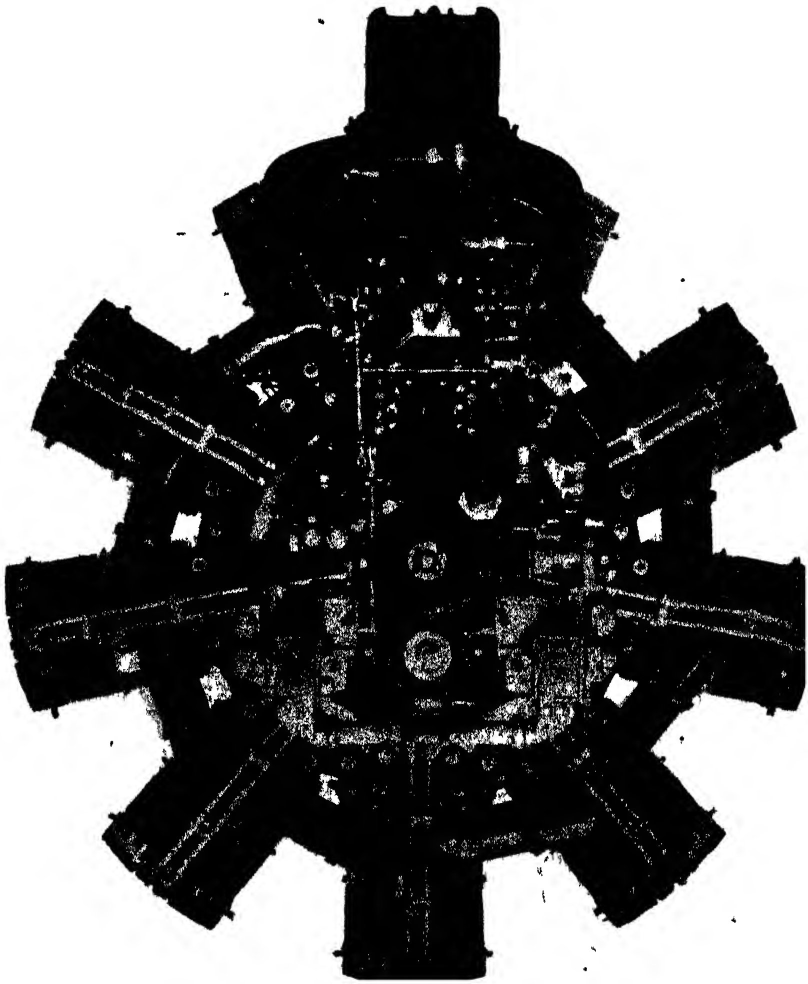

Fig. 21.—Rear View of " Bristol " Perse

The bore and stroke of the Perseus engine are the same as those of th Mercury (see p. 404). The maximum take-off power is 890 b.h.p. at 270 r.p.m., corresponding to a brake m.e.p. of 172 lb. sq. in., and the net dr weight is 1090 lb.

The above rating by no means represents the limit of possible powe from an engine of this size. Dr. Fedden, in a paper read before th

Institution of Automobile Engineers in February, 1939, states that in a test of a single-cylinder sleeve-valve unit of capacity 168·8 c. in., with a supercharge of 14 lb. sq. in., the brake m.e.p. obtained was 305 lb. sq. in. at 2400 r.p.m. and 273 lb. sq. in. at 3000 r.p.m. Mr. Ricardo, in 1927, in a unit designed for high supercharge pressures, reached a brake m.e.p. of 570 lb. sq. in. with a supercharge of 4 atmospheres. This was for a liquid-

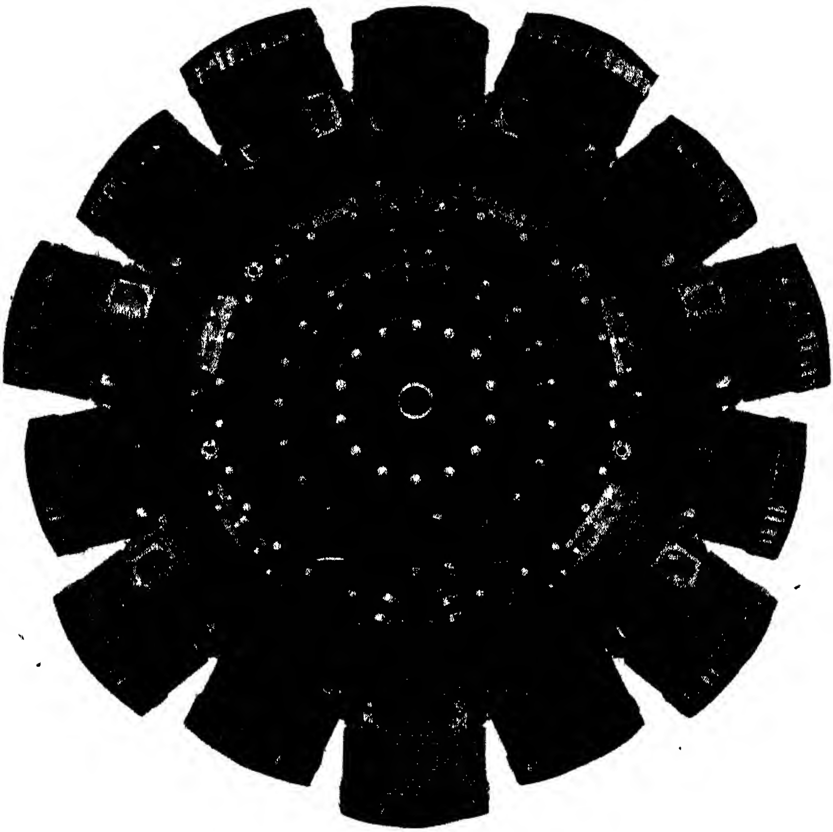

Fig. 22.—Front View of "Bristol" Hercules

cooled unit, but it emphasizes the fact that the sleeve-valve is in no way handicapped by high cylinder pressures, and that there is no difficulty in running it continuously at high speeds.

Fig. 22 shows a front view and fig. 23 a rear view of the Hercules engine. This engine has 14 cylinders in two banks of 7, and develops the same horsepower per cylinder as the Perseus.

Fig. 24 (p. 413) is a view of an installation with cowling removed, showing the mounting arrangements, exhaust system, &c.

Dr. Fedden, in the paper referred to above, gives the following list of advantages of the sleeve-valve engine:

(*a*) Total absence of maintenance, except for plug and magneto servicing.

(*b*) Elimination of hot spots in the combustion chamber.

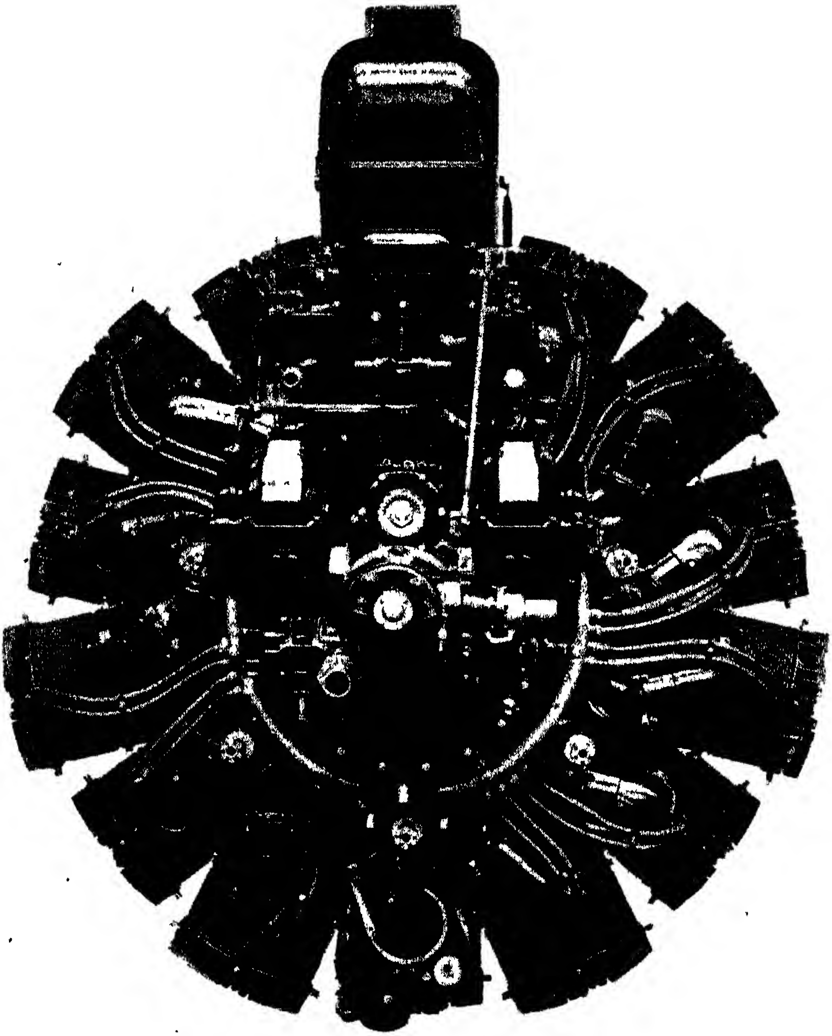

Fig. 23.—Rear View of "Bristol" Hercules

(*c*) Use of higher compression ratios or boost pressures.

(*d*) Improved volumetric efficiency.

(*e*) Centrally situated plugs, giving, if necessary, good performance on single ignition.

(*f*) Very flat mixture loops, permitting smooth running under conditions of extreme economy.

(*g*) Smooth running, due to the good combustion-chamber shape and to the accurate and simple valve-timing.

(*h*) More silent operation.

(*i*) Good accessibility and clean exterior appearance.

(*j*) Complete enclosure of all working parts, absence of external oil leads, and impossibility of oil leakage.

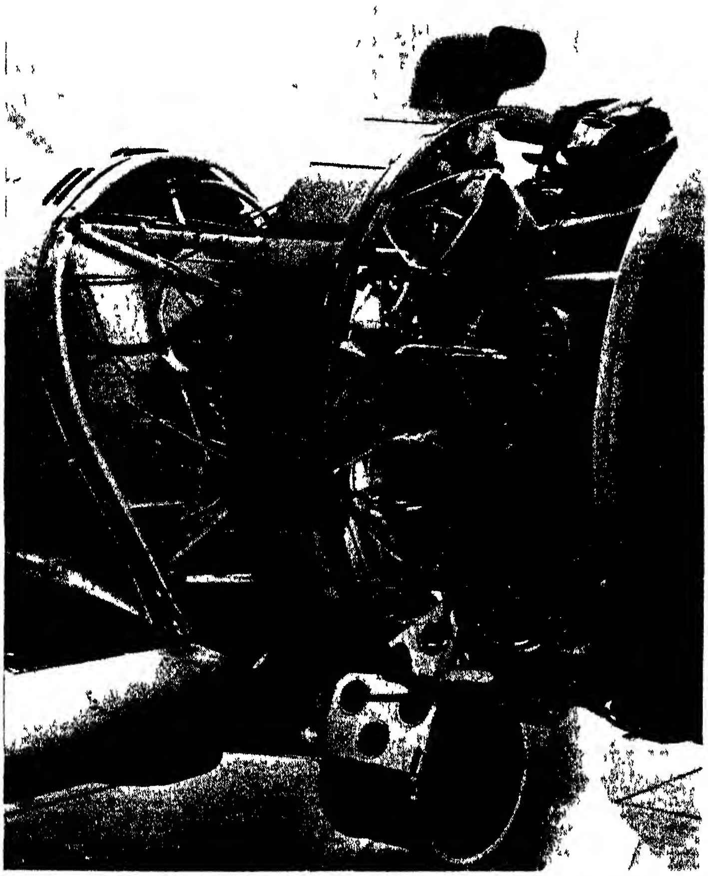

Fig 24—View of Installation with Cowling removed—"Bristol" Hercules

(*k*) Cooler exhaust.

(*l*) Freedom from lead corrosion.

(*m*) Regular cylinder shape, permitting the simplest form of baffling.

(*n*) Marked decrease in number of parts, with consequent reduced production and maintenance costs.

(*o*) Greater reliability, due to most of the causes mentioned above.

(*p*) Any desired control of cylinder turbulence.

# CHAPTER IV

## V Type Engines

The best known of the liquid-cooled V type engines and those having the highest reputation are the series manufactured by Messrs Rolls-Royce, Ltd. The first Rolls-Royce " Eagle " was produced soon after the outbreak of the war in 1914, and this very successful aero engine may justly be considered as the prototype of all the liquid-cooled aero engines so far produced. In 1926 the " Kestrel " appeared and was shortly afterwards adopted as a standard engine of the British Royal Air Force and of many of the naval and military air forces of foreign powers. The " Kestrel " has since been followed by the " Peregrine " and " Merlin " engines, all of the supercharged 12-cylinder type.

A comparison of the chief characteristics of the three types is given below:

| | *Kestrel* | *Peregrine* | *Merlin* |
|---|---|---|---|
| Bore (in.) .. .. .. | 5·0 | 5·0 | 5·4 |
| Stroke (in.) .. .. .. | 5·5 | 5·5 | 6·0 |
| Take-off b.h.p./r.p.m .. | 700/2240 | 765/3000 | 1075/3000 |
| Brake m.e.p. (lb./sq. in.) .. | 173·5 | 141·5 | 173 |
| Maximum b.h.p./r.p.m. .. | 640/2900 (at 15,000 ft.) | 885/3000 (15,000 ft.) | 1030/3000 (16,250 ft.) |
| Compression ratio .. .. | 6 : 1 | 6 : 1 | 6 : 1 |
| Net dry weight (lb.) .. .. | 955 | 1106 | 1335 |

The unsupercharged " Kestrel " (normally aspirated) has a compression ratio of 7: 1 and its take-off power is 550 b.h.p. at 2375 r.p.m. This type is intended for bombing, reconnaissance and other aircraft which depend upon long range and low fuel consumption.

**Constructional Features.**—The cylinders (fig. 25) are of aluminium block design, gas and water passages being cast integrally with the walls forming the jackets. Renewable valve seatings (aluminium-bronze for the inlet and nickel-manganese-chromium for the exhaust) are screwed into the heads. Carbon-steel cylinder liners are fitted, made gastight by means of a soft aluminium ring between their upper ends and the head, and by rubber rings fitted in grooves in the liners at the lower ends. No cylinder walls are fitted between the liners.

The connecting-rods are H-section forgings machined all over. Each throw of the crank-shaft carries a pair of rods, one of which is forked and carries a steel block big-end metalled with lead bronze. The outer surface of this block forms the bearing for the other rod of the pair, which is straight.

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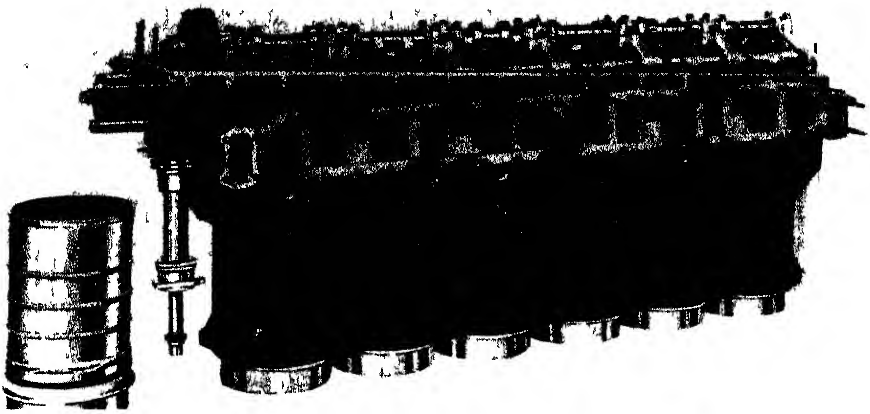

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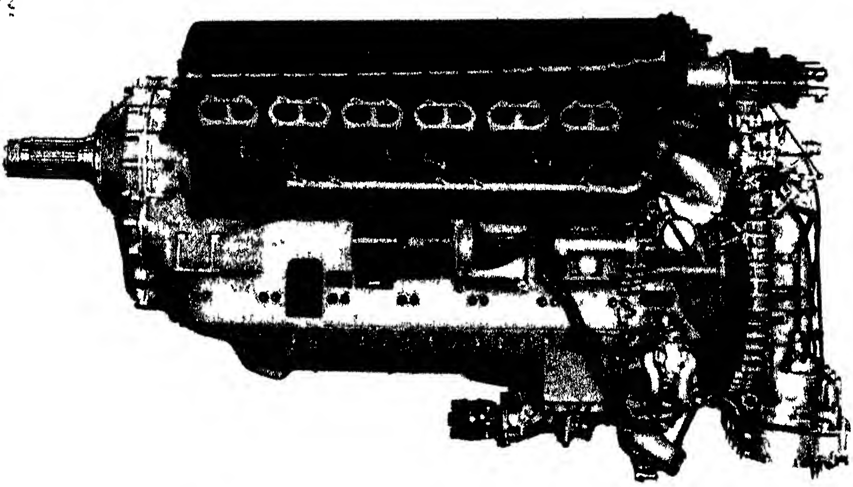

27A

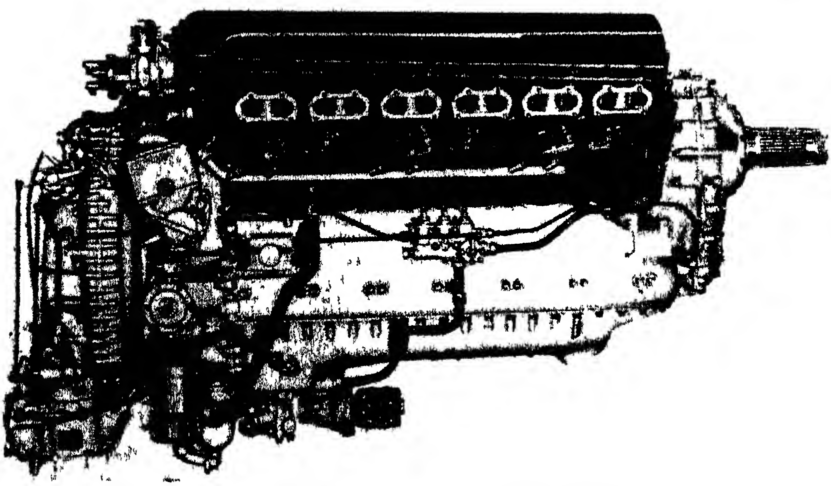

Fig 25.—Rolls-Royce Merlin Aero Engine. Fig 27.—Port Side of Rolls-Royce Merlin Aero Engine.
Fig. 27A —Starboard Side of Rolls-Royce Merlin Aero Engine.

Two inlet and two exhaust valves per cylinder are fitted, operated by overhead cam-shafts by means of a separate rocker for each valve. Exhaust valves are made of KE965 steel, sodium-filled, the inlet valves being of silicon chromium.

The crank-shaft is carried on seven main bearings, which are split mild-steel shells lined with lead-bronze and fitted into recesses machined in the crank-case. The lower half of the crank-case can be withdrawn without disturbing the bearings.

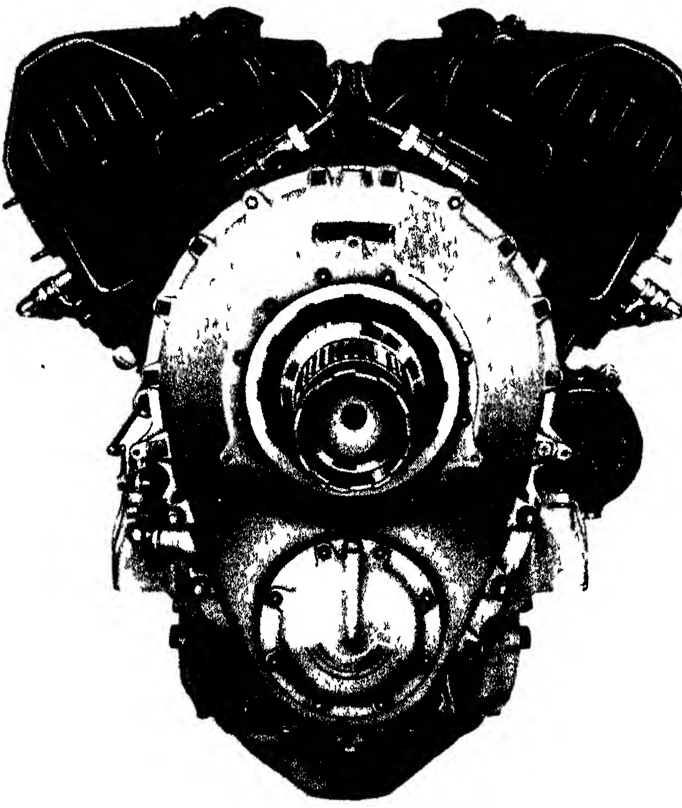

Fig 28—Rolls-Royce Merlin II Aero Engine (Front View)

The cam-shafts are operated from the rear end of the crank-shaft by spring drives which eliminate torsional crank-shaft vibration.

Fig. 26 shows the main features of construction of the Merlin engines, also the drives for the accessories.

Figs. 27 and 27A show the port and starboard sides of the Merlin engine, and figs. 28 and 29 are front and rear views respectively. Here the crank-shaft is fitted with integral balance weights opposite each crank, and the exhaust valves are treated over the crown and seat engaging surfaces with "Brightray".

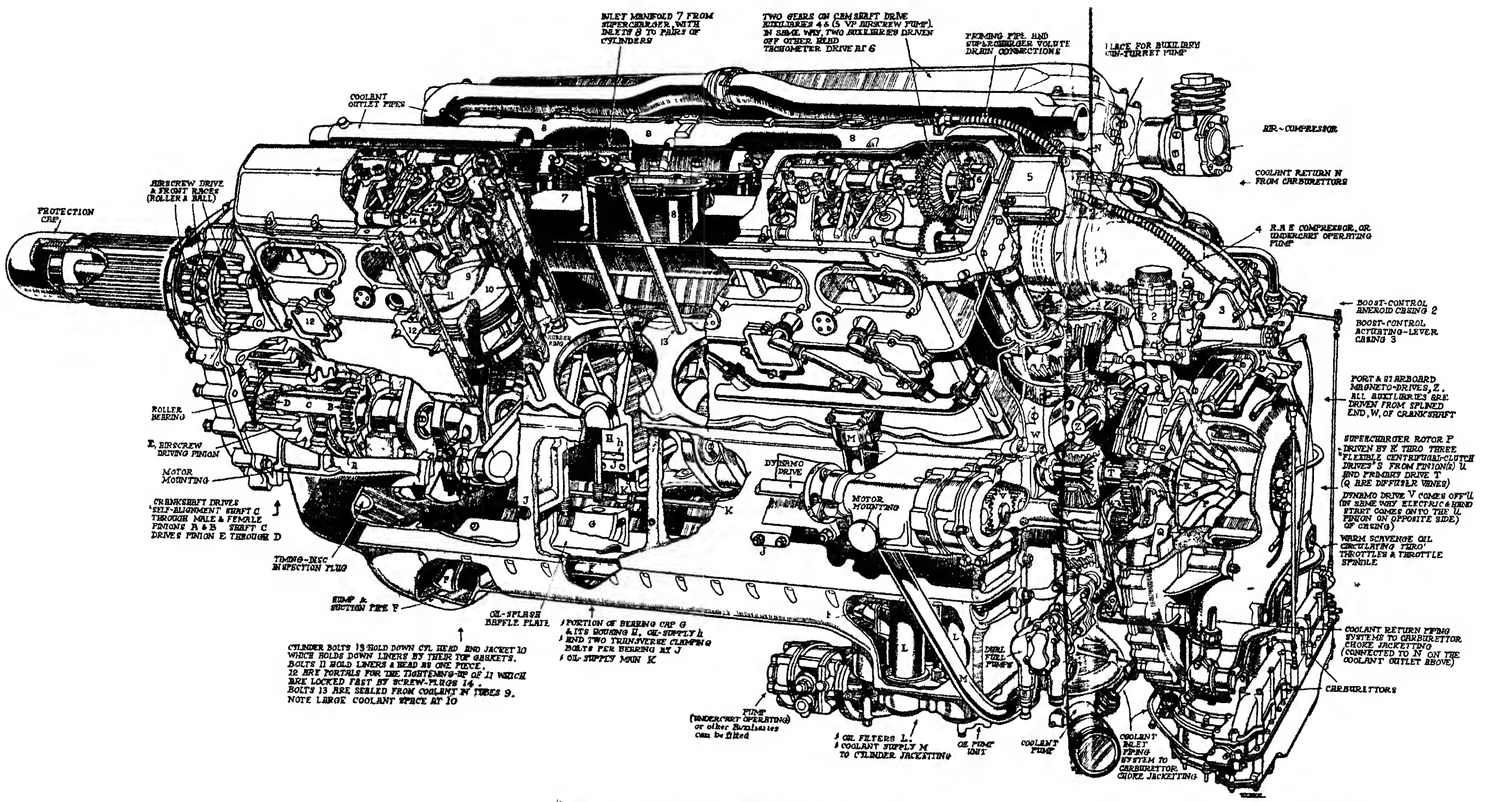


Fig. 26.—ROLLS-ROYCE MERLIN ENGINE

*By courtesy of Rolls-Royce, Ltd., and "The Aeroplane".*

Facing p. 416

**Air-cooled Inverted V Type.**—The De Havilland Gipsy Twelve is the latest and most outstanding engine of this type. It has been specifically designed for cooling by ducts from the rear, an important departure from

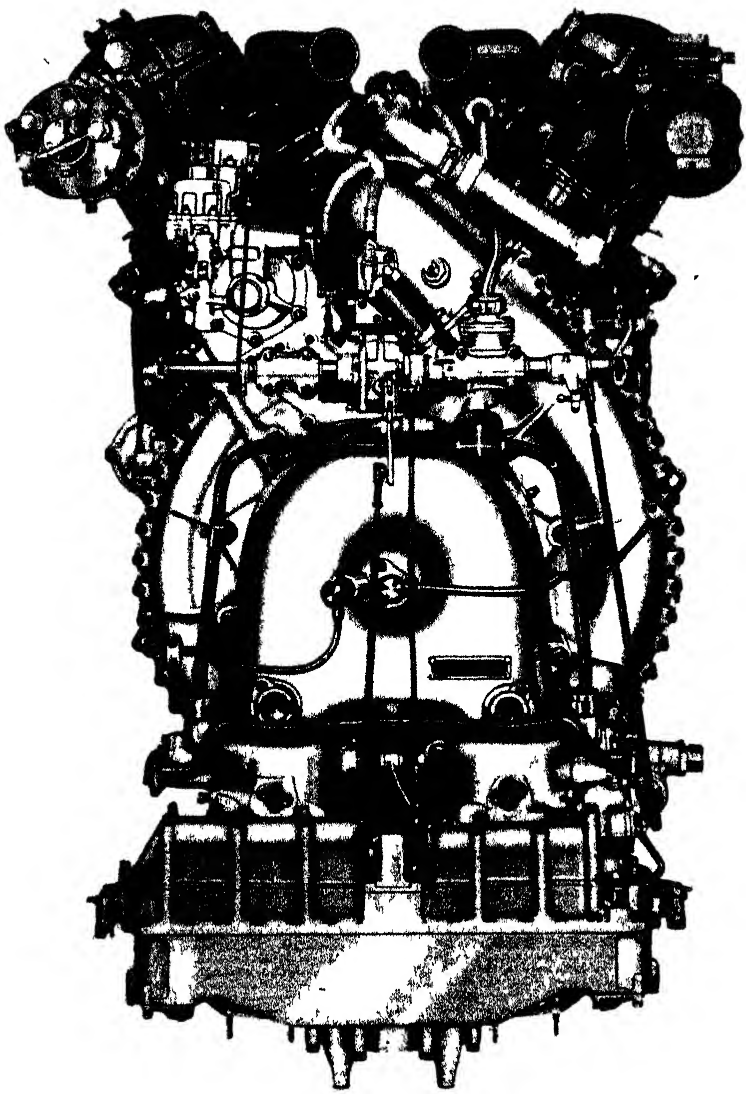

Fig. 29.—Rolls-Royce Merlin II Aero Engine (Rear View)

all previous aero engine practice. The ducts deliver air at controlled pressure and flow rate from orifices which are incorporated in the leading edge of the wing in such a way that lift and drag are not materially affected. In addition to its very low frontal area, therefore, the Gipsy Twelve has a

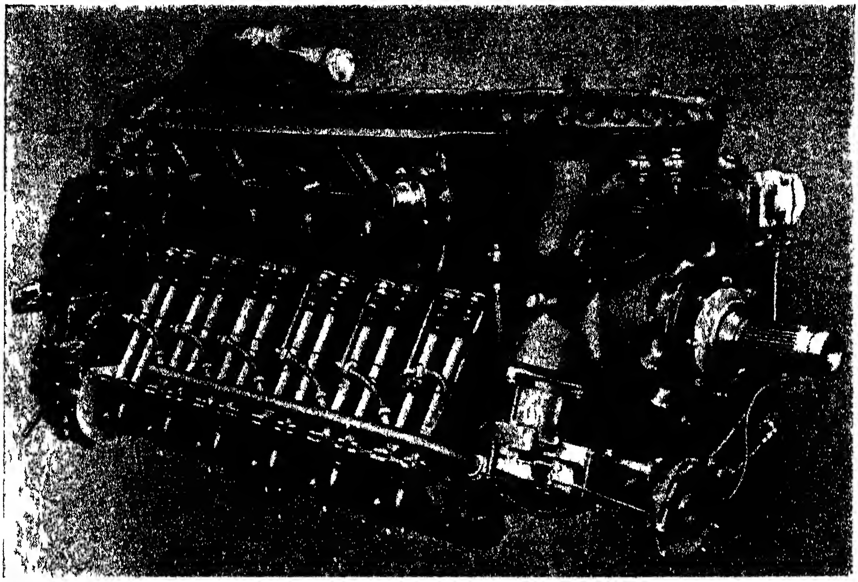

Fig 30 —Front View of D H Gipsy Twelve

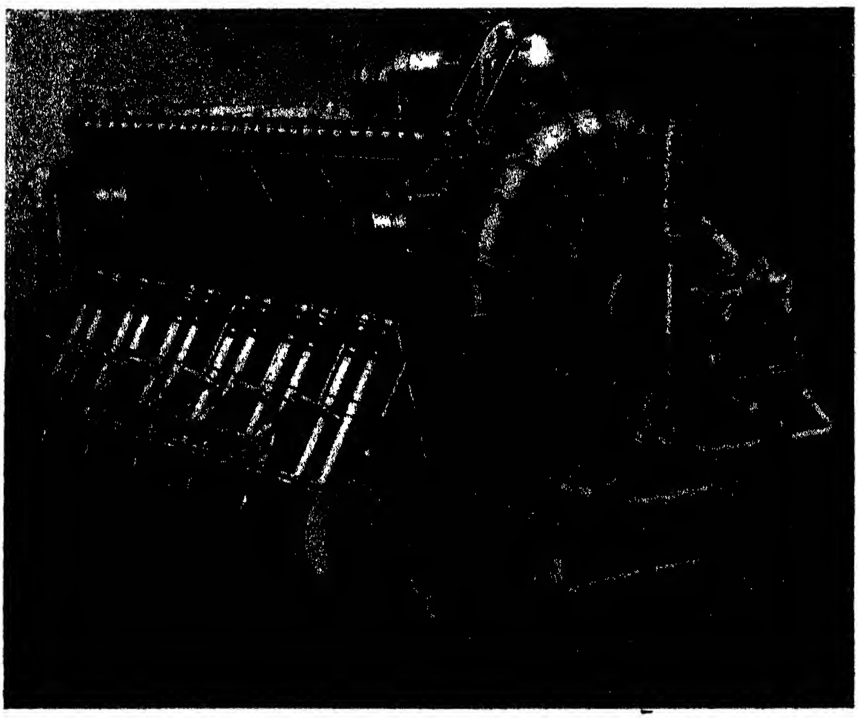

Fig 31 —Rear View of D. H Gipsy Twelve

cleaner entry and is believed to have a lower cooling loss than any other engine in production.

Figs. 30 and 31 show three-quarter front and rear views respectively of the engine.

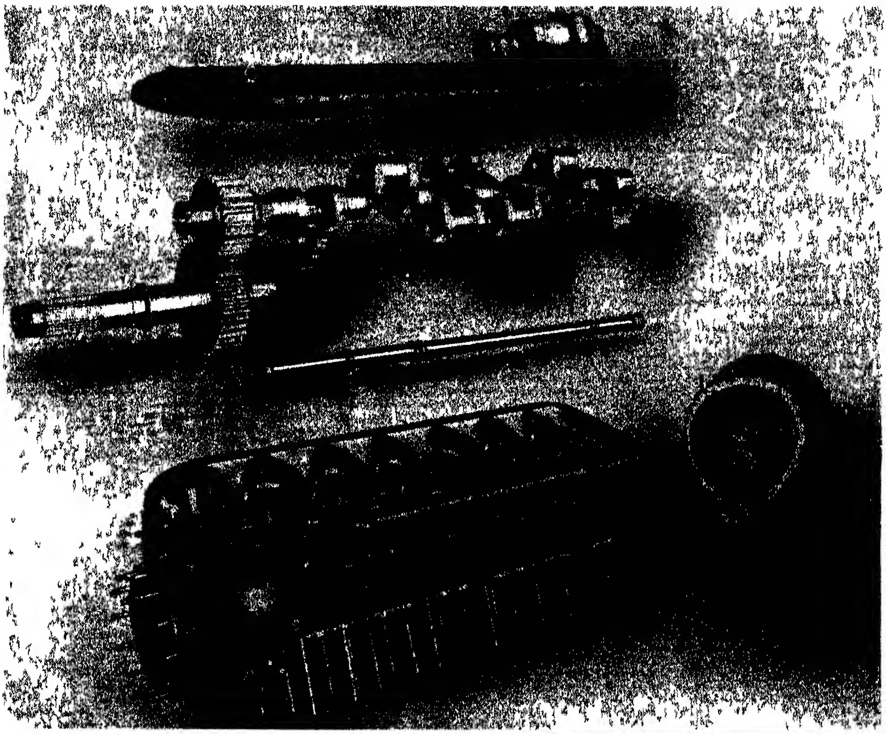

Fig 32 —Crank-shaft and Crank-case of D H Twelve

Leading particulars of the engine are as follows:

Bore, 118 mm. (4 646 in.).
Stroke, 140 mm. (5·512 in.).
Swept volume, 18·37 litres (1121·2 c. in.).
Compression ratio, 6 : 1.
Dry weight, 1058 lb.
Maximum power rating, zero boost, 410/425 b.h.p. at 2400 r.p.m. at 7750 ft.
Maximum take-off power at sea-level, 505/525 b.h.p. at 2600 r.p.m.
Maximum brake m.e.p., 137 lb./sq. in.

The cylinder head is an aluminium alloy casting which is held to the cylinder barrel by four high-tensile steel studs screwed at their upper ends to the crank-case.

The cylinder is a carbon-steel forging, machined externally to form cooling fins and ground internally. It is afterwards specially treated for protection against corrosion. The cylinder barrel projects into the crank-case, thus preventing oil from draining into the cylinders, the joint being made by a dermatine ring. The other end is spigotted into the cylinder head, the joint being made by a copper-asbestos washer.

The piston is machined from an upset forging of aluminium alloy, and is of the slipper type. Three rings are fitted to each piston, the inner ring being of the scraper type, which scrapes oil from the cylinder wall and deflects it through a series of small drilled holes to the inside of the piston and so back to the crank-case.

The H-section connecting-rods are machined all over from forgings of 65-ton nickel-chrome steel. Their construction is similar in principle to those of the Rolls-Royce engine.

The crank-shaft (fig. 32) is a nickel-chrome steel forging machined all over and rotates in 8 main bearings.

There are two steel cam-shafts, each with 12 integral cams. The 7 bearings are of large diameter to enable the cam-shaft to be withdrawn from the front.

The crank-case (fig. 32) is a deep-section elektron casting with 7 massive cross webs which carry the main bearings low down in the crank-case, thereby forming an extremely stiff box section.

The exhaust valve is stellited, a process found by the makers to be essential when running engines on leaded fuel. The ends of inlet- and exhaust-valve stems are also stellited in order to withstand wear.

The civil test of this engine included five hours of continuous running at maximum take-off conditions (3½ lb./sq. in. boost, 525 b.h.p., 2600 r.p.m.), such as are experienced for only a minute or two at a stretch in everyday service. It also included a run of one hour at 2575 r.p.m. against a very light brake, which naturally imposes severe inertia loads on the bearings.

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## CHAPTER V

# Inverted Vertical Engines

Vertical cylinders, both air-cooled and water-cooled, have been used almost from the beginnings of heavier-than-air flight, but in recent years most firms have discarded these in favour of other types. With the exception of a few engines of small power for light aeroplanes the inverted vertical type is now favoured as offering the least obstruction to the view of the pilot, and these are all air-cooled. The De Havilland Co. specialize in this type, and the many flight records achieved by aircraft engined by the famous "Gipsy" engines testify to the reliability and endurance of these engines under the most arduous conditions.

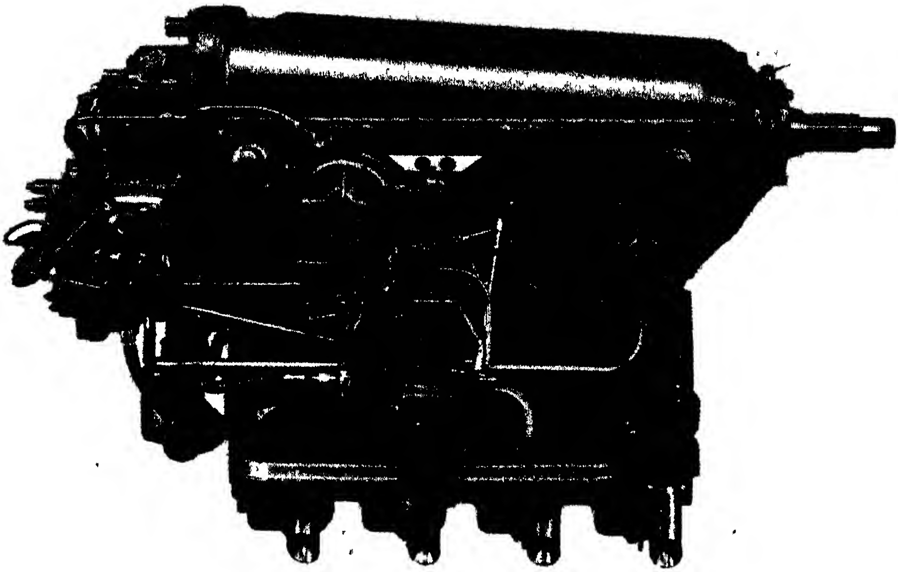

Fig. 33.—Starboard Side View of D. H. Gipsy Major

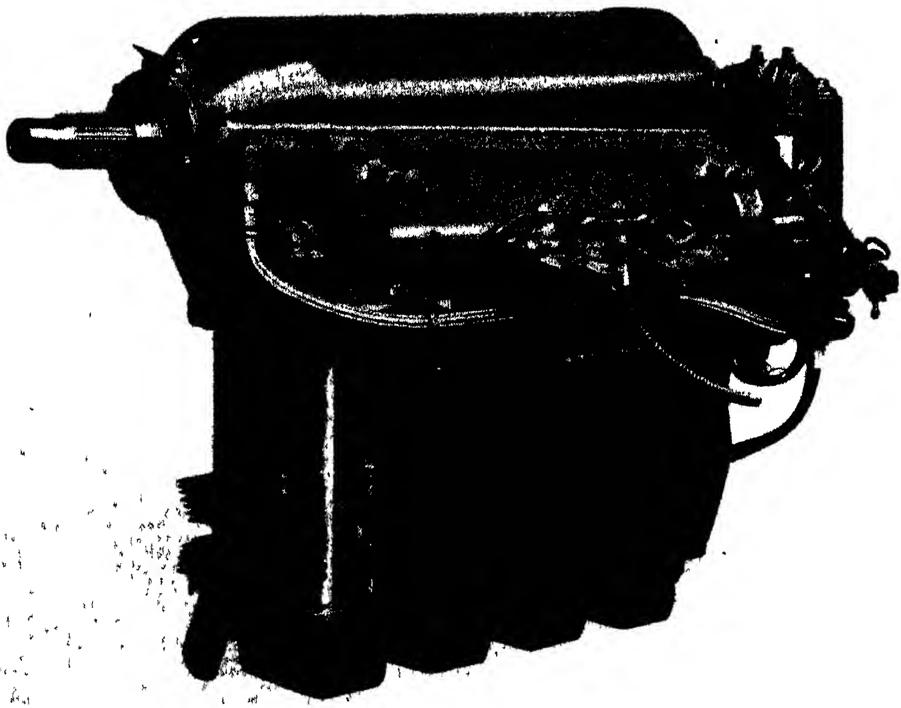

Fig. 34.—Port Side View of D. H. Gipsy Major

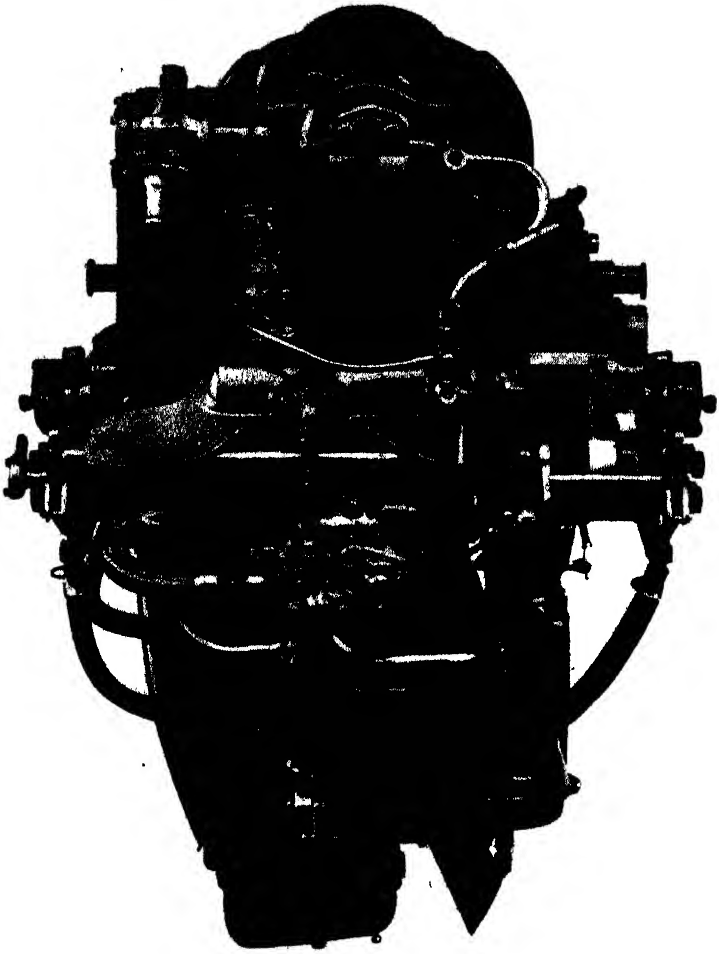

Fig. 35.—End View of D. H. Gipsy Major

The Gipsy Twelve has been described in Chapter IV, and Table II gives leading particulars of the main "Gipsy" types:

TABLE II

| | *Minor* | *Major I* | *Major II* | *Six I* | *Six II* | *Twelve* |
|---|---|---|---|---|---|---|
| Number of cylinders | 4 | 4 | 4 | 6 | 6 | 12 |
| Bore (in.) .. .. | 4·016 | 4·646 | 4·646 | 4·646 | 4·646 | 4·646 |
| Stroke (in.) .. .. | 4·528 | 5·512 | 5·512 | 5·512 | 5·512 | 5·512 |
| Compression ratio .. | 6 | 5·25 | 6 | 5·25 | 6 | 6 |
| Take-off b.h.p. .. | 80 | 130 | 140 | 200 | 205 | — |
| Revs. per min. .. | 2250 | 2350 | 2400 | 2350 | 2400 | — |
| Maximum brake m.e.p. | 125 | 126 | 134 | 126 | 128 | 137 |
| Piston speed (ft. sec.) | 32·80 | 35·95 | 36·70 | 35·50 | 36·70 | 37·50 |
| B.h.p. per litre .. | 23·9 | 21·3 | 22·5 | 22·2 | 22·4 | 28·6 |
| Weight per h.p. (lb.) | 2·32 | 2·34 | 2·28 | 2·39 | 2·29 | 2·02 |

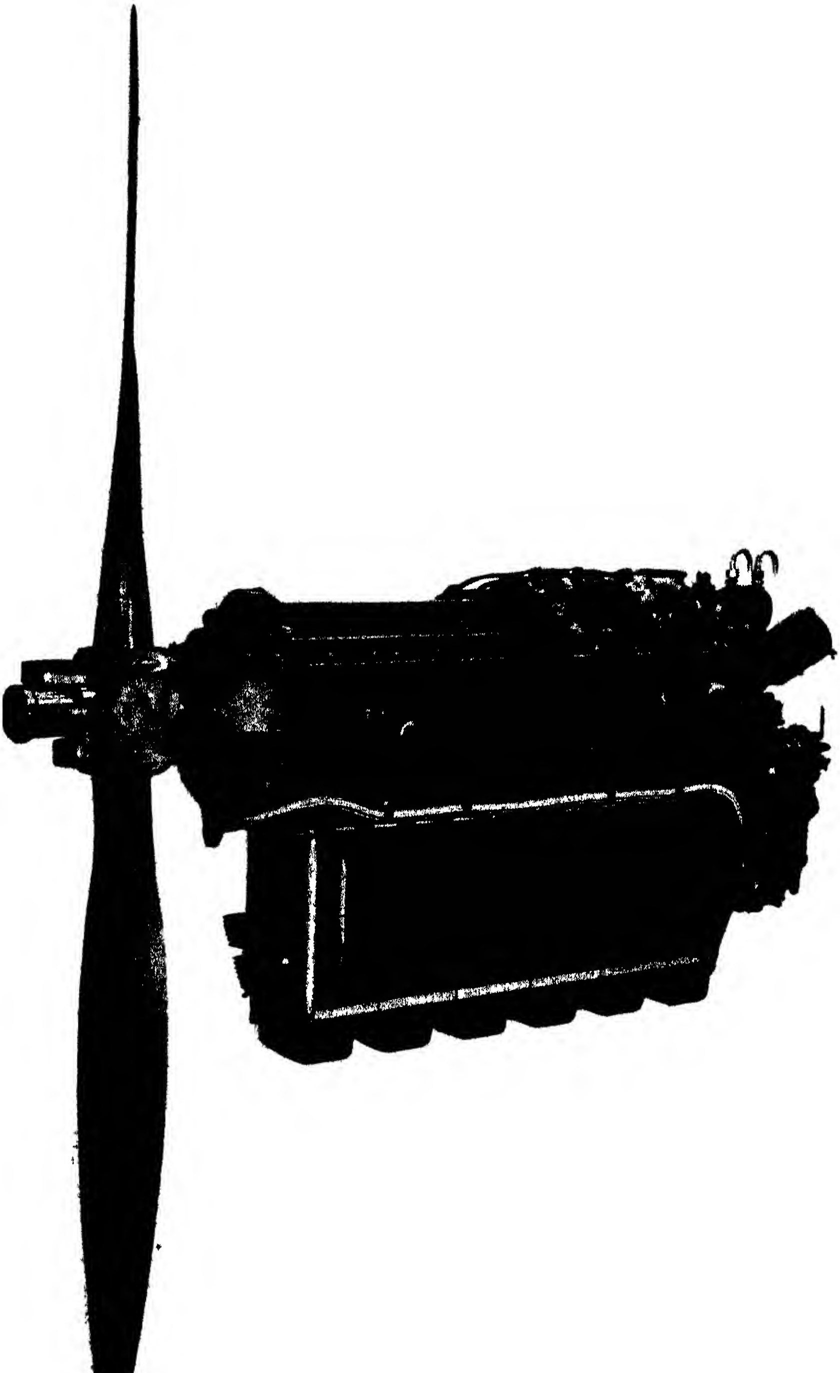

Fig 36.—Starboard Side View of D. H Gipsy Six

Some particulars of the Gipsy Twelve are repeated in Table II for convenience of comparison.

The Gipsy Minor is a smaller edition of the Gipsy Major, and is intended specifically to meet a new demand for light aircraft of essentially simple type, where robustness, economy and ease of maintenance are paramount requirements.

Fig. 33 (p. 421) is a view of the starboard side of the Gipsy Major, Series II, showing the carburettor, induction manifold and controls.

Fig. 34 is a view of the port side, showing the method of guiding the air over the cooling fins, one of the two B.T.H. magnetos, and long bolts holding the cylinder heads in position.

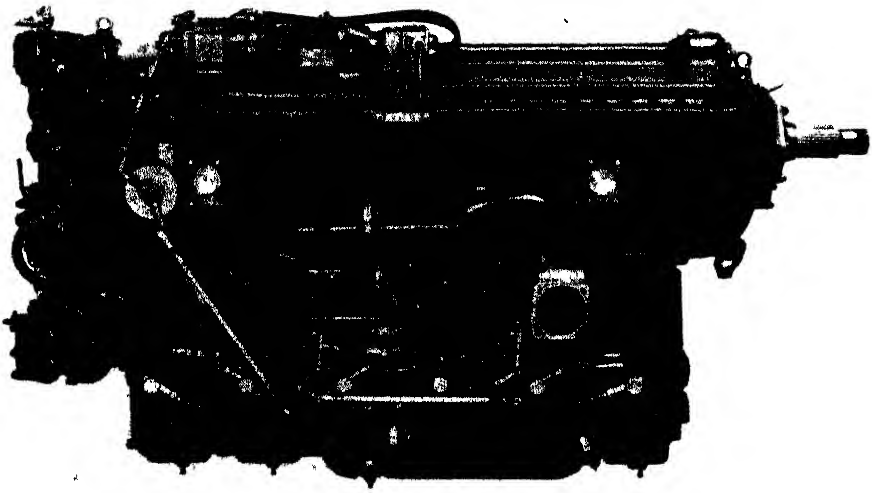

Fig. 37.—Port Side View of D H Gipsy Six

Fig. 35 (p. 422) is an end view showing positions of the principal accessories.

The general features of construction are similar to those detailed for the Gipsy Twelve. The valve gear is fully enclosed, and cam-shaft, oil pumps and dual tachometer drives are driven by spur trains from a pinion on the rear end of the crank-shaft.

A spiral gear train from the intermediate shaft of the cam-shaft driving train drives the magneto cross-shaft. A drive for a vacuum pump is provided by a second cross-shaft which itself is driven from the crank-shaft by a spur and bevel train. Starter drive is transmitted through a dog mounted on the rear end of the crank-shaft. All the above drives and auxiliaries are housed in or mounted on a magnesium alloy timing cover at the rear of the engine.

Figs. 36, 37 and 38 show three views of the Gipsy Six Series II engine. Fig. 36 is a view of the starboard side, showing the air-scoop and controllable-pitch airscrew.

Fig. 37 shows the port side, with the two carburettors, induction manifold, magnetos and controls.

Fig. 38 is an end view showing the auxiliaries, which include a vacuum pump for the operation of flying instruments and an electric generator of output 500 watts.

The Gipsy Six I is intended to operate a fixed-pitch airscrew, has a compression ratio of 5·25, and uses fuel of 70 octane, and has therefore a

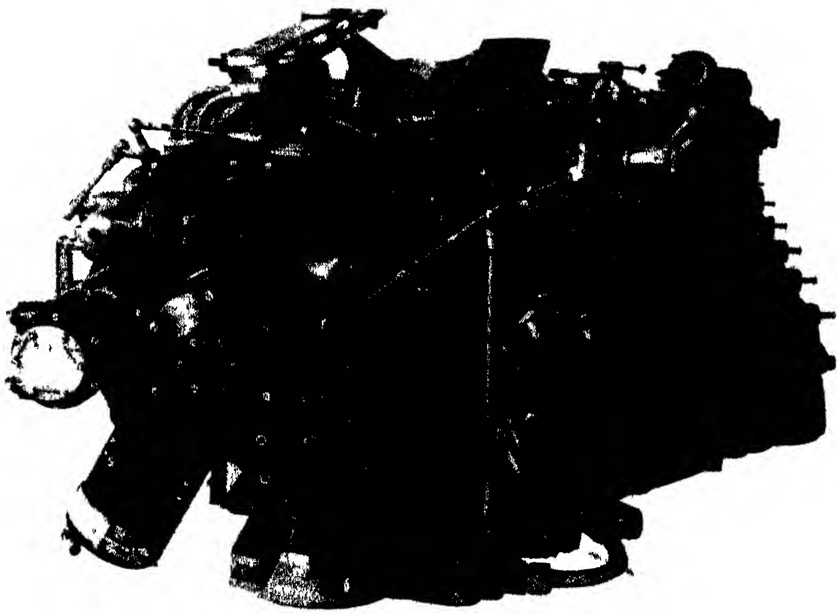

Fig 38.—End View of D H. Gipsy Six

somewhat wider field of application than the Gipsy Six II, which was developed to operate a controllable-pitch airscrew, and thus benefits from a higher compression ratio of 6, a higher continuous power output for level cruising, and also a higher crank-shaft speed for take-off and climb. It uses fuel of 77 octane.

Owing to the conservative power rating and care taken with the general design and lay-out, duty periods of 1000 hours between overhauls are recommended and adopted.

# CHAPTER VI

## H Type Engines

This type of air-cooled engine is manufactured exclusively in this country by Messrs D. Napier & Sons, Ltd., and has the advantages of extremely compact construction combined with high output. It consists of four banks or cylinders, two vertical and two inverted vertical. Each pair of banks (vertical and inverted vertical) operates a separate crank-shaft, and the two crank-shafts are geared together to the airscrew shaft.

Fig. 39 shows the cross-sectional arrangement of the earlier Napier Rapier engine, which differs somewhat from later types in detail but illustrates the general lay-out.

The Rapier engine has 16 cylinders, while the latest Dagger VIII has 24 cylinders. Leading particulars of the Rapier engine, Series VI, are as follows:

Number of cylinders, 16.
Bore (in.), 3½.
Stroke (in.), 3½.
Compression ratio, 7·0.
Normal rating, 355/370 b.h.p. at 3650 r.p.m. at 4750 ft.
Maximum rating, 380/395 b.h.p. at 4000 r.p.m. at 6000 ft.
Maximum take-off power, 365 b.h.p. at 3500 r.p.m. at sea-level.
Maximum r.p.m. for T.V. dive, 4800.
Net dry weight (lb.), 713.
Capacity (litres), 8·83.
Maximum h.p. per litre, 47·4.
Brake m.e.p. at maximum rating, 145.

The progress which has been made in design and performance since the advent of the Rapier VI is shown by the corresponding particulars of the Dagger VIII:

Number of cylinders, 24.
Bore (in.), 3·813.
Stroke (in.), 3·75.
Compression ratio, 7·5.
Normal rating, 890/925 b.h.p. at 4000 r.p.m. at 9000 ft.
Maximum rating, 1000 b.h.p. at 4200 r.p.m. at 8750 ft.
Maximum take-off power, 955 b.h.p. at 4200 r.p.m. at sea-level.
Maximum r.p.m. for T.V. dive, 4800.
Net dry weight (lb.), 1390.
Capacity (litres), 16·8.
Maximum h.p. per litre, 59·5.
Brake m.e.p. at maximum rating, 184.
Overall length, 6 ft. 2 in.
Overall width, 2 ft. 3 in.
Height overall, 4 ft.

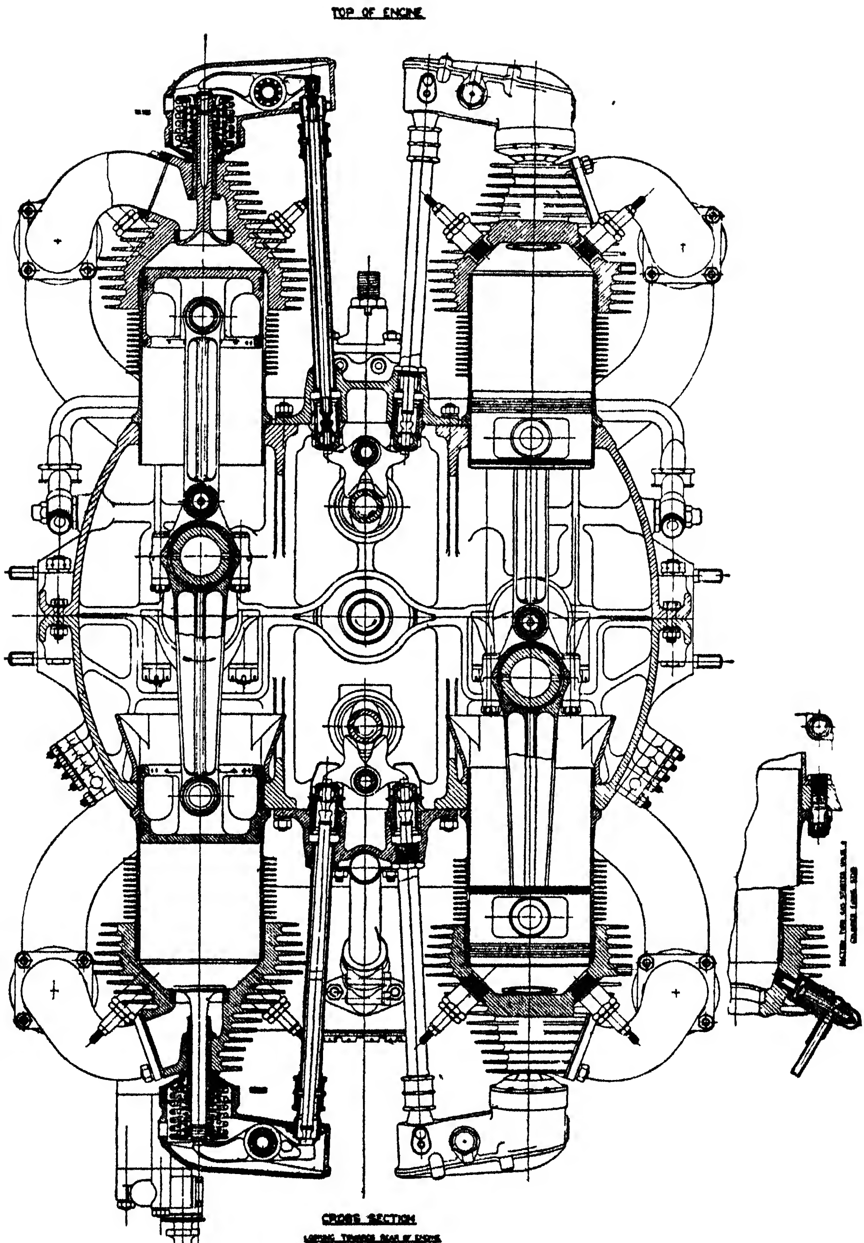


Fig. 39.—Arrangement of Cross-section of Napier Rapier Engine

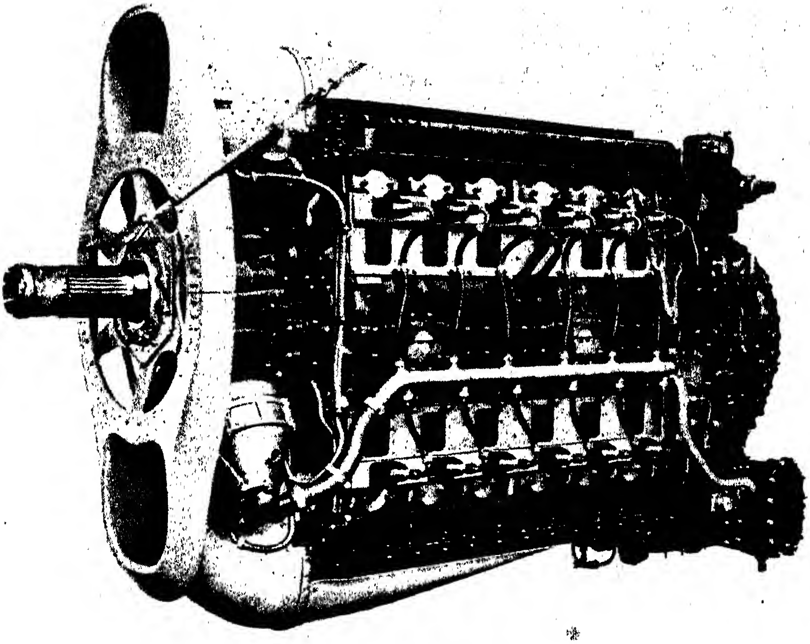

Fig. 40.—Port Side View of Dagger VIII

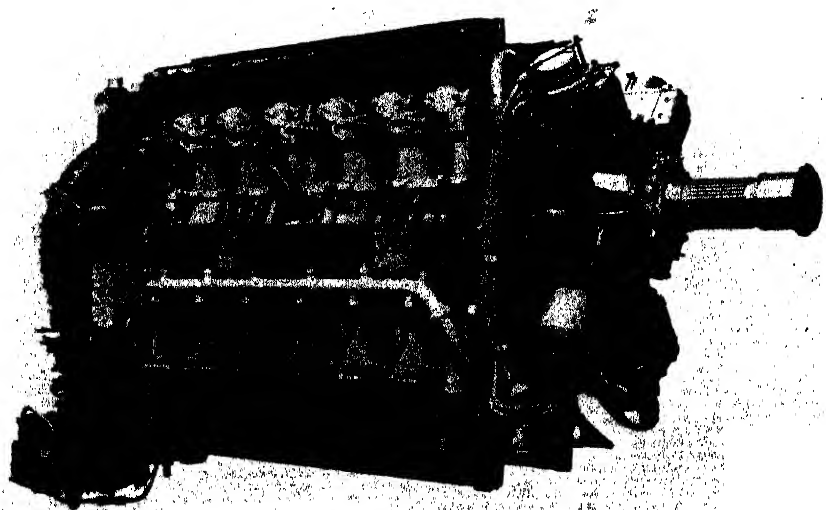

Fig. 41.—Starboard Side View of Dagger VIII

Fig. 40 is a view of the port side of the Dagger VIII engine and shows the compact arrangement. At the left-hand end is the distributor for the 24 sparking-plugs. The cowl and air-scoop are also shown. The electric generator and electric starter are shown at the top right-hand.

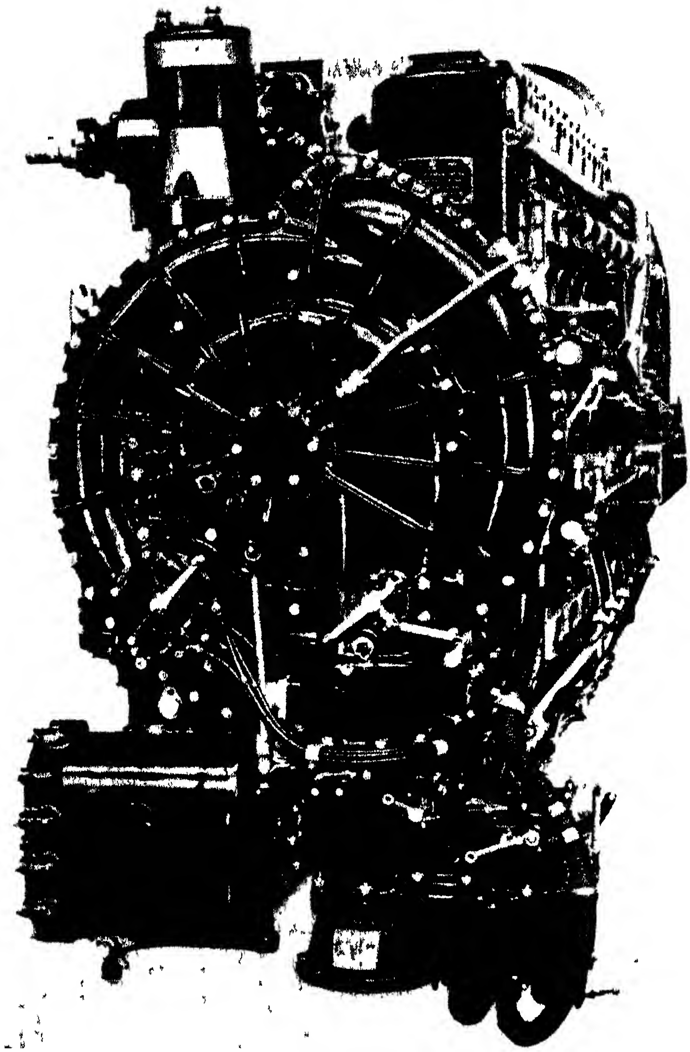

Fig 42.—Rear View of Dagger VIII

Fig. 41 shows the starboard side and fig. 42 is a rear view showing the supercharger casing.

The airscrew shaft is carried on two bearings, a roller bearing immediately in front of the gear and a ball and roller bearing at the front of the nosepiece. The reduction in speed between the crank-shafts and the airscrew shaft (0·308) is obtained through case-hardened spur gears.

Four overhead cam-shafts are fitted, one for each bank. Hydraulic tappets operated by oil pressure are incorporated in the valve rockers and take up the clearance automatically.

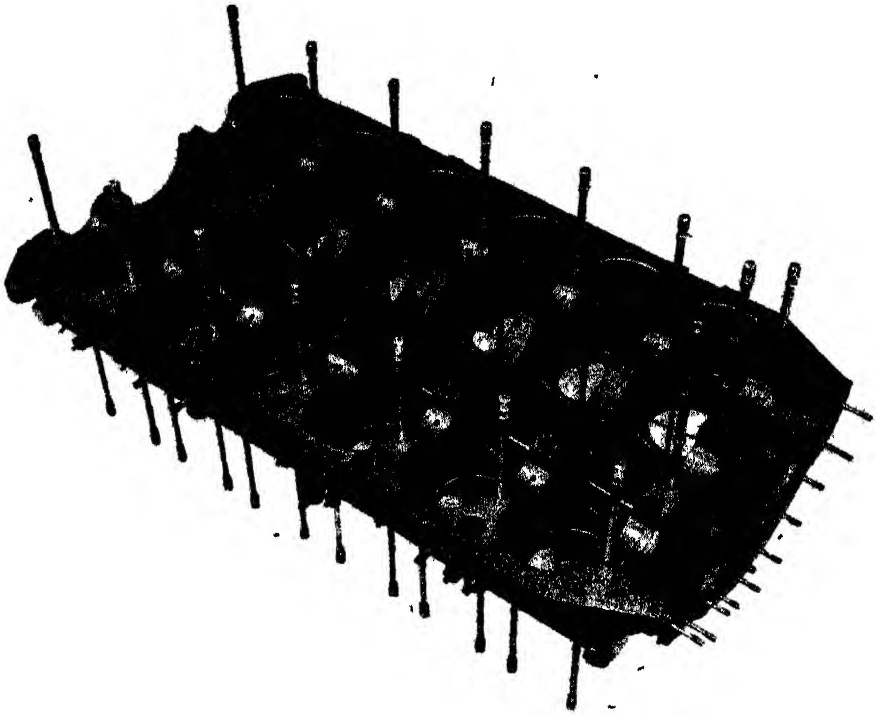

Fig 43 —Bottom Half of Dagger VIII Crank-case

The connecting-rod assembly for each crank consists of a forked rod in which the lead-bronze-lined steel bearing shell is fixed, and a plain rod which oscillates on the outside of the bearing shell.

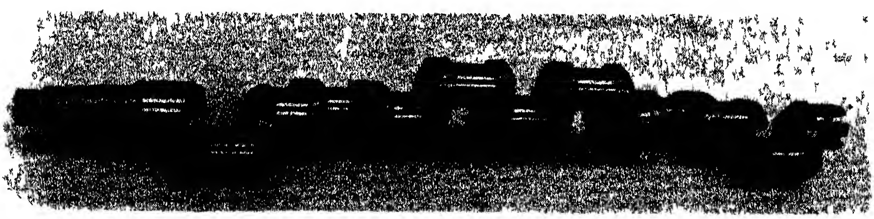

Fig 44 —Crank-shaft of Dagger VIII

The crank-case (the bottom half of which is shown in fig. 43) is of aluminium alloy and is in halves, joined along the horizontal centre line of the engine.

Each crank-shaft is machined from a solid steel forging, and is carried in eight lead-bronze bearings (fig. 44).

Separate close-finned steel cylinders, machined all over, are fitted. The cylinder heads are of forged aluminium alloy and are machined all over. These contain the inlet and exhaust passages and valves.

The pistons are of forged aluminium alloy and are fitted with two gas rings, two scraper rings and a hollow gudgeon-pin.

An important feature of this engine is the perfect balance obtained and the small torque variation, due to the number and arrangement of cylinders and cranks.

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# CHAPTER VII

## Supercharging and Superchargers

As was stated in Chapter I, the amount of fuel that can be consumed in the cylinder is limited only by the weight of air that is supplied to the cylinder, so that if we assume that no difficulties with pressures, temperatures and disposal of waste heat arise, there is no limit to the power obtainable by supercharging (more usually referred to by the more convenient term " boosting "). In actual practice the difficulties just referred to above limit the amount of boost which is practicable to quite a modest rise of pressure above atmospheric. High mean effective pressures mean also high maximum pressures but not necessarily high maximum temperatures, since the quantity of air to be heated increases in proportion to the amount of fuel to be consumed. Of the heat generated by combustion, approximately 30 per cent is converted into useful work, 40 per cent goes to piston and cylinder, and the remaining 30 per cent goes to exhaust, the relative proportions varying somewhat according to speed and compression ratio. Thus, doubling the horse-power, for instance, involves doubling the amount of heat to be dissipated by the cooling fins in the case of an air-cooled engine, and an increase of piston temperature which may weaken the material to a serious extent and, together with increased sparking-plug temperatures, increase the liability to detonation and pre-ignition.

The quantity of air passing over the fins, and the total cooling area provided, are, of course, limited, and extra heat can only be dissipated by an increase of cylinder temperature, the limitations of which are obvious.

At high altitudes the weight of air passing is reduced in proportion to the reduced density, although this disadvantage is lessened to a great extent by the lower temperature of the air, and with a variable-pitch airscrew the quantity of air passing for a given airscrew speed can be increased.

In actual practice the amount of boost provided is sufficient to maintain rather more than atmospheric pressure (at ground level) up to some specified height (known as the " rated height "); beyond this height the power of the engine falls off according to the reduction in density of the surrounding air.

In most cases a centrifugal supercharger is used, with diffuser vanes to convert the velocity of exit at the tips of the blades into pressure and so increase the efficiency. The supercharger, which is driven by the engine, produces a more or less constant ratio between the pressures of supply and delivery to and from the compressor. If the supply of air to the supercharger is unrestricted, the amount of boost at ground level will be sufficient to cause excessive loads and temperatures, so that the inlet to the supercharger must be throttled to restrict the air supply. The carburettor is fitted on the inlet side of the supercharger, so that it is not subjected to the boost pressure.

To relieve the pilot of the necessity for watching the boost gauge when taking-off, climbing and flying " all-out " in level flight, to prevent overboosting the engine, a device is fitted to the carburettor and connected to the throttle linkwork in such a manner that it automatically controls the opening of the carburettor throttle to maintain the boost pressure within the permitted limits. This device is known as the automatic boost control, and it may be set by the pilot to give one of two maximum conditions of boost, i.e. either maximum level and climbing boost, or maximum take-off boost.

Fig. 45 shows the principle of operation of the automatic boost control. The control itself consists of an airtight chamber connected to the pressure side of the blower and containing a stack of capsules such as are commonly used in an aneroid barometer. These capsules are connected to a piston valve which moves in one direction or the other, according as the capsules expand or contract under the influence of changing boost pressure. Movement of the piston valve admits high-pressure oil from the engine to one side or other of a piston, known as the servo-piston, and this piston operates the linkage interposed between the pilot's throttle lever and the carburettor throttle lever.

If, owing to change of altitude, engine load or speed, or change in throttle opening as a result of the pilot moving his throttle lever, the induction pressure changes, the capsule will change in length, thereby moving the piston valve. This in turn will allow oil to flow to one side or other of the servo-piston, which in turn will alter the throttle opening until the induction-pipe pressure once more returns to its original figure. It will be seen, therefore, that as the aeroplane climbs from the ground to its rated height the throttle will gradually be opened, the controlling factor being induction-pipe pressure, which keeps constant.

Fig. 46 shows diagrammatically the capsule, piston valve and servo-piston. The capsule is made of nickel-steel and is almost entirely exhausted of air. Failures in service are almost unknown, thanks to careful selection of material.

Fuller particulars of this and more elaborate arrangements for control are given in a treatise on " Automatic Engine Controls ", by E. W. Knott, M.I.A.E., M.S.A.E., published by Messrs H. M. Hobson (Aircraft & Motor) Components, Ltd., by the courtesy of whom figs. 45 and 46 and the descriptive matter are reproduced.

**Two-speed Supercharger.**—The advantages of a supercharger driven by some two-speed mechanism over one having a single high gear ratio are apparent at two main conditions of running: firstly, at take-off and the early stages of climbing, and secondly, for obtaining optimum fuel economy in level flight when cruising.

For a modern aeroplane fitted with a controllable-pitch airscrew the take-off r.p.m. is frequently equal to, and sometimes in excess of, the

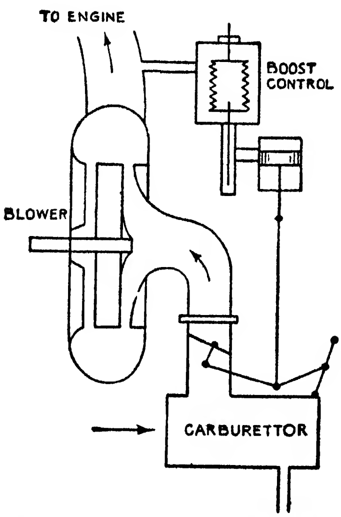

Fig. 45—One Way of applying a Boost Control

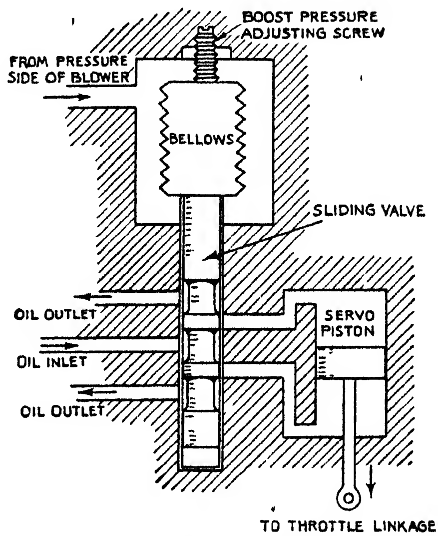

Fig 46—The Parts of a Boost Control

r.p.m. at maximum level power at altitude. This means that with a single-speed fully-supercharged engine the supercharger is absorbing maximum power and supplying mixture to the engine at an unreasonably high temperature. This condition also holds in a rather smaller degree during the early stages of the climb, as the engine must still run throttled to keep the boost pressure within the prescribed limits, and since the atmospheric temperature is still comparatively high the mixture temperature will still be excessive.

The provision of a lower gear ratio for use under these conditions removes both these disadvantages simultaneously. The blower in the low gear takes less power to drive and, owing to the lower peripheral speed of the rotor, the temperature rise of the mixture as it passes through the supercharger is greatly reduced.

Thus under take-off conditions the power available at the propeller is increased, owing, firstly, to the small loss in driving the supercharger, and, secondly, to the higher mixture density at any given boost pressure, resulting from the lower mixture temperature. In addition, the lower mixture temperature referred to is of great value in avoiding detonation, and may enable the engine to be run safely with a weaker mixture and consequently with an increase in fuel economy.

This effect on fuel economy provided by a two-speed supercharger is especially valuable under cruising conditions at the full operational height of the aircraft, and provides probably the chief argument in favour of such a unit. When cruising, an engine is required to supply two-thirds of its maximum power at about the same altitude as that given for maximum power. With a single-speed supercharger this condition can only be met by throttling. Now with a fully-supercharged engine with a single-speed blower rated at, say, 15,000 ft., and throttled back to cruising power at this altitude, the power taken by the blower and the temperature of the mixture entering the cylinders are not greatly altered from the figures at full throttle. This waste of power causes a high fuel consumption, and in practice it is not possible to reduce this much below ·54 pt./b.h.p. hr. With a two-speed blower, however, the cruising conditions can be achieved at or near full throttle by changing to the low gear. This cuts down the power to drive the blower by about half and also reduces the mixture temperature very considerably. This double advantage allows the fuel consumption to be reduced to about ·49 pt./b.h.p. hr., thus giving a very valuable 10 per cent increase in cruising range. By simply changing back to high gear the maximum power of the engine is immediately available if required.

In order to reap the full advantage of the two-speed blower the gear itself should be as little heavier than a single-speed mechanism as possible, the operating mechanism must be simple and positive in action and the reliability unimpaired. This latter condition calls for a gear change which is not too rapid in action, for otherwise the stresses in the mechanism due to the sudden acceleration of the rotor may become dangerous.

The following description and the diagram (fig. 47) will show how these requirements have been met in the Tiger VIII two-speed supercharger. Essentially it consists of an orthodox step-up gear with three sets of layshaft and planet gears.

In the high gear the drive is taken from the end of the crank-shaft through a spring-drive mechanism A to the gear-wheel C via a toothed coupling ring B. The planet gears are mounted in a cage F which is fixed when the high gear is in use. Thus the drive is taken from the sun-wheel C to the pinion D, which revolves with the planet gear E, the connection between the two being effected by means of sets of centrifugal clutch-pads. The gear E in its turn drives the rotor-spindle L, the whole forming a simple planet-gear train.

In a normal single-speed unit the mounting of the planet-gear spindles is fixed to the casing, but in this case it will be seen that they are mounted in the cage F, which is supported on the ball-bearing M at one end and

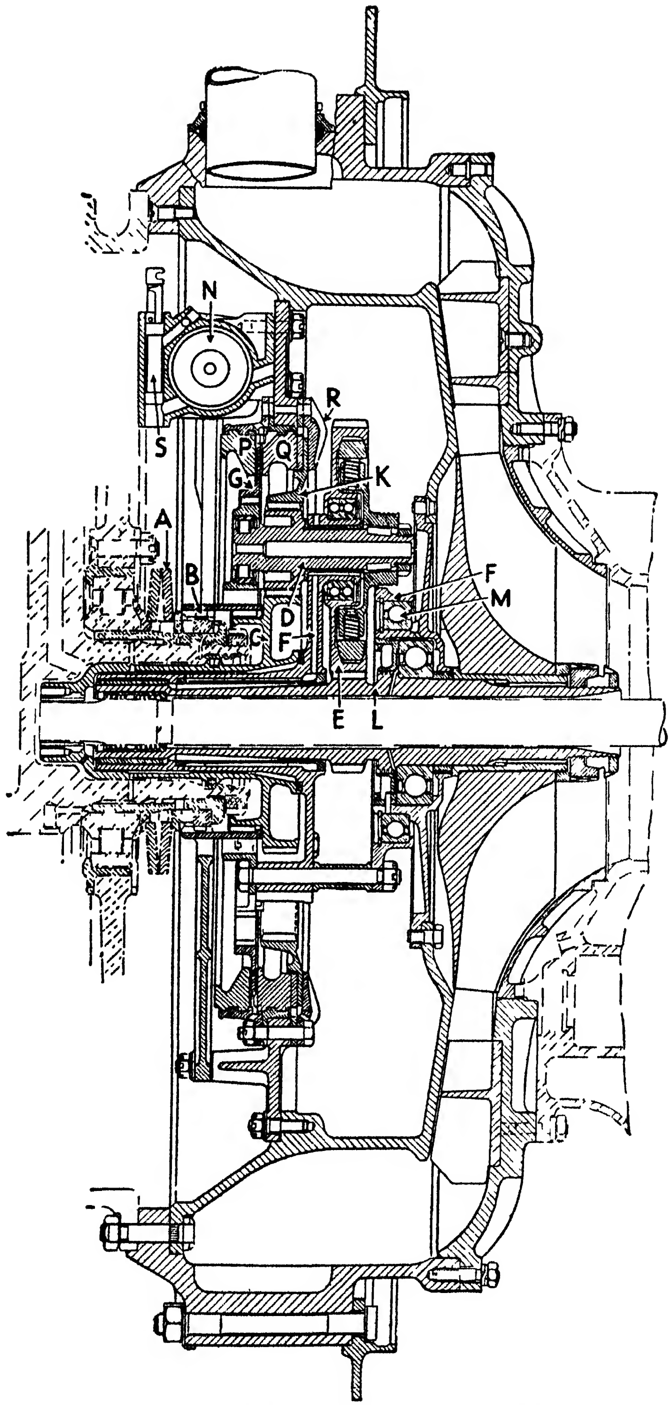


Fig. 47.—Armstrong-Siddeley Two-speed Supercharger

in the hollow end of the crank-shaft at the other. The cage can be prevented from turning by means of the clutch-plate G, which is held between the clutch-nuts P and Q, and is connected to the cage by means of internal teeth meshing with an interrupted ring of external teeth on the cage.

The change-speed mechanism operates by moving the nuts P and Q. These are two multi-start square threaded nuts, one left-hand and the other right-hand, which are constrained to move round together by means of dogs. The nut P is connected to two double-acting pistons working in the cylinder N by means of a toggle system designed to increase the turning moment. Oil can be admitted to the cylinder either in the centre, in which case it forces both pistons outwards, or at the ends, forcing the pistons inwards.

When the pistons are forced outwards the nuts P and Q are moved round together and thus approach one another, gripping the clutch-plate G firmly between them. The unit then operates in high gear as previously described. If now, by means of a simple push-pull mechanism S, the oil pressure is transferred to the ends of the cylinder, the pistons move in and the nuts move round in the opposite direction. This immediately releases the clutch-plate G and hence the cage F. The nut P moves out of the way, while the nut Q comes up against the clutch-plate K, holding it firmly against the backing-ring R.

It will be seen that K has a ring of internal teeth which mesh with the three planet-gear pinions D. The cage being free to rotate, the mechanism now acts as an epicyclic gear train with the annulus K as the reaction member. The effect of this is to reduce the step-up ratio between the gear C and the rotor-spindle L, thus providing the low gear of the supercharger.

It will be seen that the only additions which have been made to the simple single-speed gear are the operating clutches and nuts, the oil-cylinder pistons and toggle levers, and the bearings supporting the planet cage. By this method of construction it has been found possible to cut down the extra weight involved to a low figure.

The method of operation and the friction clutches employed ensure that the change-over from low to high gear or vice versa is not unduly fierce, while being at the same time positive in operation. Assuming that engine-oil pressure is present at the main cylinder, one gear or the other is always in engagement.

The above description and the accompanying fig. 47 are included by the courtesy of Armstrong-Siddeley Motors, Ltd.

**Temperature rise due to Compression.**

Let $T_1$ = initial temperature (absolute),
$T_2$ = final temperature (absolute),
$p_1$ = initial pressure,
$p_2$ = delivery pressure,
$n$ = index of compression curve ($pv^n$ = constant).

Then
$$T_2 = \left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} . T_1, \text{ and}$$

$$\text{Rise of temperature} = T_2 - T_1 = T_1\left\{\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} - 1\right\}.$$

For adiabatic compression of air, $n = 1{\cdot}4$, but in actual practice $n$ is about $1{\cdot}5$, since frictional losses in the supercharger increase the final temperature.

If $t_1$ is the initial temperature in deg. F., then $T_1 = t_1 + 460$. If $t_1 = 60$, $\frac{p_2}{p_1} = 1{\cdot}2$, then temperature rise $= 520 \times {\cdot}062 = 32{\cdot}2$ deg. F. If $\frac{p_2}{p_1}$ is increased to $1{\cdot}6$, then temperature rise $= 520 \times {\cdot}17 = 88{\cdot}5$ deg. F.

It will thus be seen that with a high supercharge the temperatures throughout the engine cycle will be increased very appreciably unless an intercooler is used, involving additional weight.

**Effect of Supercharge on Volumetric Efficiency and Induction Temperature.**

Let $v_1$ = clearance volume,
$p_1$ = pressure of gas in clearance before induction,
$T_1$ = absolute temperature of clearance before induction,
$V$ = stroke volume,
$v_2$ = volume of air taken in at supercharge pressure $p_2$,
$T_2$ = abs. temp. of supercharged air at admission,
$T_3$ = abs. temp. of mixture at end of induction stroke.

Then weight of gas in clearance, $w_1 = \frac{p_1 v_1}{RT_1}$;

weight of air taken in, $w_2 = \frac{p_2 v_2}{RT_2}$;

total weight in cylinder, $w_3 = \frac{p_2(v_1 + V)}{RT_3}$;

$$\therefore \frac{p_2(v_1 + V)}{T_3} = \frac{p_1 v_1}{T_1} + \frac{p_2 v_2}{T_2}. \quad \ldots\ldots\ldots\ldots(9)$$

Since loss of heat by exhaust residuals = gain of heat by entering air,

$$w_1(T_1 - T_3) = w_2(T_3 - T_2). \quad \ldots\ldots\ldots\ldots(10)$$

From equations (9) and (10) approximate values of $T_3$ and $v_2$ can be calculated.

For example, let $p_1 = 15$, $T_1 = 1200$, $v_1 = 10$,
$p_2 = 20$, $T_2 = 540$, $V = 50$.

From (9),
$$\frac{20 v_2}{540} + \frac{150}{1200} = \frac{1200}{T_3}.$$

From (10),
$$\frac{150}{1200}(1200 - T_3) = \frac{20 v_2}{540}(T_3 - 540).$$

From these two equations we find that $T_3 = 579$ deg. absolute $= 119$ deg. F. and $v_2 = 52{\cdot}6$, i.e. the volume of air taken in at supercharge pressure is 5·2 per cent greater than the stroke volume.

With no supercharge and $T_2 = 520$ the value of $T_3$ is 633 deg. absolute $= 173$ deg. F.

In the example given, $\frac{p_2}{p_1} = 1{\cdot}33$, and the result holds for other supercharge pressures so long as $\frac{p_2}{p_1}$ remains at that value, for instance with $p_1 = 12$ and $p_2 = 16$.

The above calculation is only approximate, since we have assumed that air is being dealt with throughout and there are no heat losses during induction; the figures, however, illustrate additional advantages of supercharging.

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## CHAPTER VIII

# Carburation

The conditions under which a carburettor has to function in an aero engine differ in a number of important respects from those under which a carburettor for a motor vehicle functions:

(*a*) The variation of engine speed is much smaller.

(*b*) The density of the air supply to the carburettor decreases as the altitude increases.

(*c*) The mixture strength must be capable of variation in flight, a rich mixture being required for slow running, climbing and short periods of high-speed level flying, and a weak mixture for economical running.

Compensating jets or similar devices for obtaining approximately constant mixture strength with widely varying engine speeds are therefore not required, but some means of counteracting the tendency to increase of richness with reduced air density must be provided, and the operation of this is usually controlled by the position of the mixture lever and (in the case of some supercharged engines) by the amount of boost.

Uniform distribution of the mixture to all the cylinders is all-important if maximum power and economy are to be obtained, and careful arrangement of the induction system is necessary. Better distribution with a single carburettor is usually obtained with a supercharged engine, since change of direction of the mixture flow is stimulated by the higher pressure in the induction manifold.

Modern aeroplane carburettors require a mixture control which can give a 40 per cent change of mixture strength from the richest to the weakest mixture used, and it is obvious that with this range excessive use of the mixture control may cause damage to the engine through overheating from mixtures being too weak and, on the other hand, wastage of fuel will reduce the range of action of the aeroplane. Many attempts on the long-distance records have failed through the incorrect use or abuse of the mixture control.

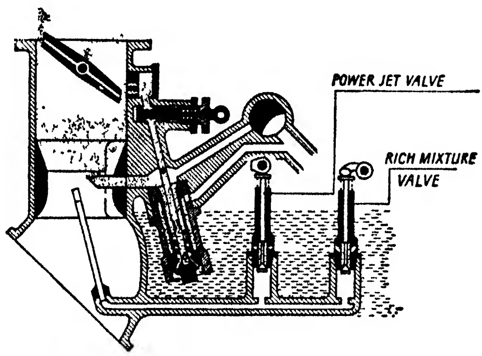


Fig. 48.—Small Throttle Opening

It is a well-known fact that as mixture strengths are weakened to a point where maximum powers are obtained with minimum fuel consumption (known as W.M.M.P.) engine temperatures rise to a dangerously high point, and carburettor settings are always set to a mixture strength sufficiently richer than this to ensure safety. It was found, however, that if mixture strengths considerably weaker than that known as W.M.M.P. are used, temperatures fall if the mixture is made weak enough to cause also a fall in engine revolutions. It was found desirable to limit this drop in revolutions to about 3 per cent. The normal mixture for full-power conditions is approximately 10 per cent richer than W.M.M.P., and the over-rich mixture for take-off conditions is 12 per cent richer than the normal mixture. The cruising mixture strength giving a drop in revolutions with steady running and safe temperatures is about 8 per cent weaker than W.M.M.P. Particulars of the Hobson-Penn Automatic Mixture-Control, which corrects the mixture automatically according to requirements, are given in the treatise by E. W. Knott referred to on p. 432.

The principal features of the Hobson Aircraft Carburettor are shown in figs. 48, 49, 50 and 51. The chief parts consist of a slow-running jet, power-jet valve, rich mixture valve and altitude valve. Where variable datum boost control and automatic mixture control are used these are now incorporated in the carburettor. A slow-running cut-out is shown for use in case the engine, owing to overheated plugs or carbon deposit, continues to run after the ignition is switched off.

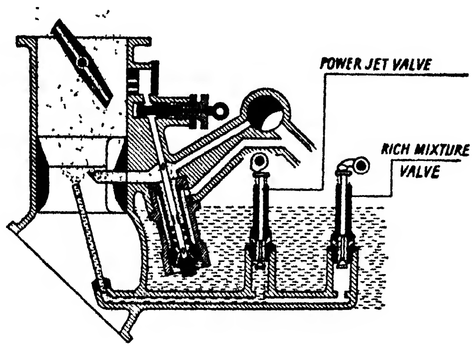


Fig. 49.—Approaching Full Throttle

With a small throttle opening the slow-running jet and main jet only are in action (fig. 48). As full throttle is approached the power-jet valve opens and the auxiliary jet comes into action (fig. 49). For take-off boost with full throttle the rich mixture valve is also opened, as shown in fig. 50.

In fig. 51 the throttle is fully open, but the altitude valve is also open, admitting air to the main jet, and thus weakening the suction on the jets,

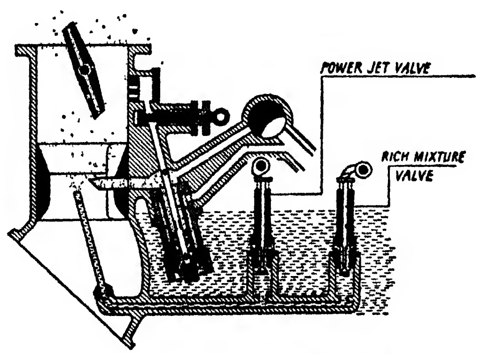


Fig. 50.—Rich Mixture for Take-off Boost

the rich mixture valve being closed. By progressive opening of the altitude valve the suction on the jets can be decreased as the air density decreases, thus keeping the petrol-air ratio by weight reasonably constant. The altitude valve does not, however, affect the slow-running system.

As the suction on the main jet increases (altitude valve closed), sets of holes are uncovered which admit air and cause the fuel to issue in the form of an emulsion which is weaker according to the number of holes uncovered, thus regulating the mixture strength.

In the case of monoplanes the fuel is usually carried in the wings, so that little, if any, gravity head is available for supply to the carburettor, even in level flight. With high-speed seaplanes and flying boats the fuel may have to be lifted through appreciable heights. Changes of attitude during climbing, diving and manœuvring will, of course, correspondingly alter the effective head or lift as the case may be, so that to ensure a steady supply of fuel to the carburettor a fuel pump or pumps must be provided. Most fuel pumps are engine-driven. They must be placed in a reasonably cool position, and some form of control for the delivery pressure must be incorporated in those types which are not inherently self-regulating.

Fuel pumps may be classified as follows:

(*a*) *Centrifugal pumps.* These are used only for engines of small power and are usually windmill-driven.

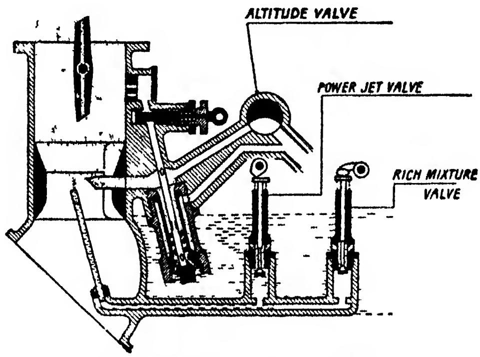


Fig 51 —Mixture-control Operation

(*b*) *Diaphragm pumps,* representative types of which are the Amal, Armstrong-Siddeley, De Havilland and Pobjoy Tecalemit. In these the diaphragm is flexed by means of a cam and returned by means of a spring. In the latest Amal pump the diaphragm is controlled positively throughout its movement.

(*c*) *Vane type pumps,* represented by the Bristol duplex vane pump. The Evans and Romec pumps are used in a number of American machines.

(*d*) *Plunger pumps,* represented by the Hobson pump, which is driven by compressed air.

(*e*) *Gear pumps,* used by Messrs Napier and Sons and by Rolls-Royce, Ltd.

Much information on the construction and operation of fuel pumps for aircraft is given in a paper by W. C. Clothier, published in Vol. XXXI (1936–7) of the *Proceedings of the Institution of Automobile Engineers.*

The following figures, selected from a number of test results supplied by the courtesy of Armstrong-Siddeley Motors, Ltd., illustrate the degree of economy to be expected from the use of weak mixtures. The results quoted are extracted from a series of tests of a Tiger VIII engine with full supercharge, running at 1870 r.p.m.

Minimum fuel consumption in pints per b.h.p. hour:

| | |
|---|---|
| Normal rich mixture | 0·645 |
| ½ per cent h.p. drop | 0·490 |
| 6 per cent h.p. drop | 0·460 |
| 11½ per cent h.p. drop | 0·455 |

The maximum cylinder temperatures in degrees C. with the above mixtures were respectively 180, 200, 195 and 180.

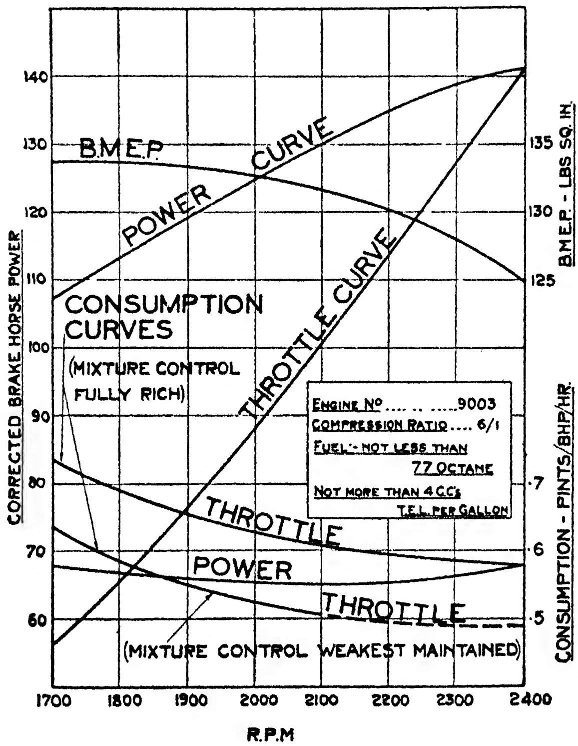


Fig. 52.—Gipsy Major Series II Engine Power and Throttle Curves—Air Ministry Type Test, May, 1937

The load-consumption curves are reasonably flat, as shown by the figures of 0·71 pint at 350 b.h.p., 0·645 pint at 450 b.p.h. and 0·68 pint at 575 b.h.p. with normal rich mixture at 1870 r.p.m.

Fig. 52 shows power and consumption curves for the De Havilland Gipsy Major Series II engine.

Fig. 53 shows curves of economic cruising consumption for the Napier-Halford Dagger VIII. The effect of engine speed on consumption is clearly shown in this diagram, but it must be remembered that for a given b.h.p. the mean effective pressure is reduced as the speed increases. Curves AB, CD and EF, which have been inserted by the author, are curves of constant m.e.p., from which the change in consumption with change of speed for a given m.e.p. can be observed.

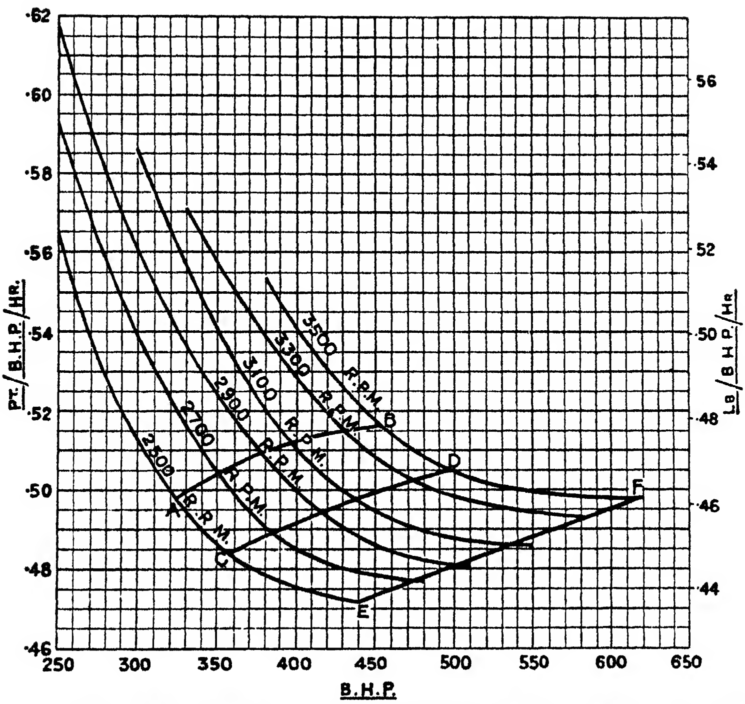


Fig. 53.—Napier-Halford Dagger VIII. Economic Cruising Consumption based on Type Test

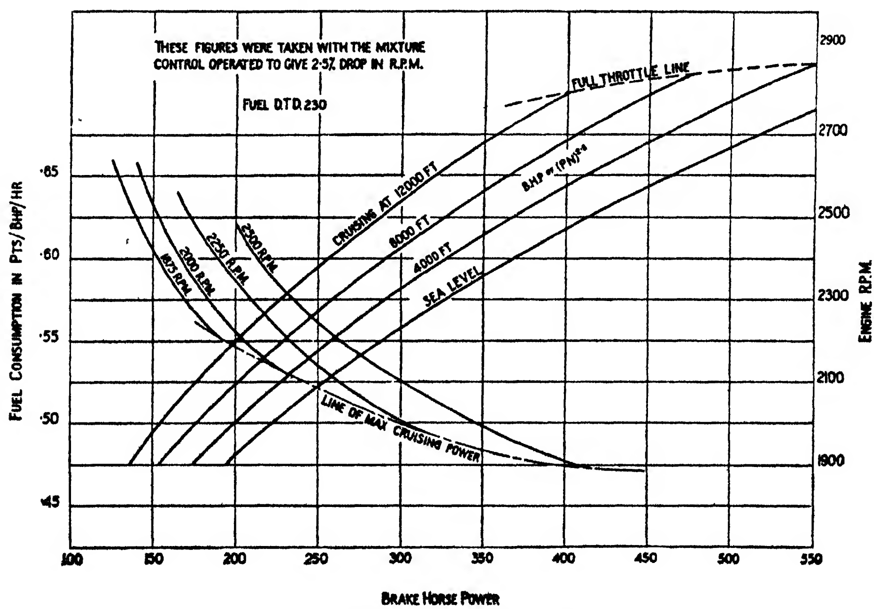


Fig. 54.—Throttled Fuel Consumption Curves (R.-R. Kestrel X)

Fig. 54 shows curves of throttled fuel consumption for the Rolls-Royce Kestrel X. The relation between brake horse-power and engine revolutions for different altitudes is also shown.

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# CHAPTER IX

## Controllable-pitch Airscrews

The thrust of an airscrew depends on the revolution speed, the pitch and the density of the air in which it works. It is not necessarily proportional to these quantities, but a reduction in one of these quantities

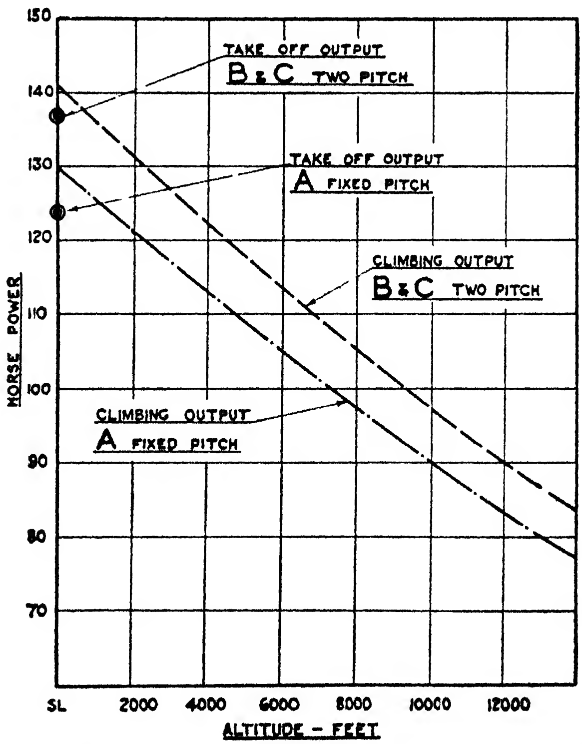


Fig. 55.—Gipsy Major Series II Engines. Comparison of outputs available for take-off and climb when using fixed-pitch or two-pitch controllable airscrews

requires an increase in at least one of the others if the thrust is to be maintained. Since the horse-power developed by the engine at its maximum safe speed depends on the resisting torque due to the airscrew thrust, it will not be possible at altitude to use the same horse-power as at ground level, even with supercharging, unless the pitch of the airscrew is increased in accordance with the reduced air density. This is why the controllable-pitch airscrew has been introduced in order to improve the performance of aircraft both for climbing and for level flight at altitude.

An airscrew which can absorb and utilize the full power of an engine during the take-off and climb and then, after change of pitch, convert the

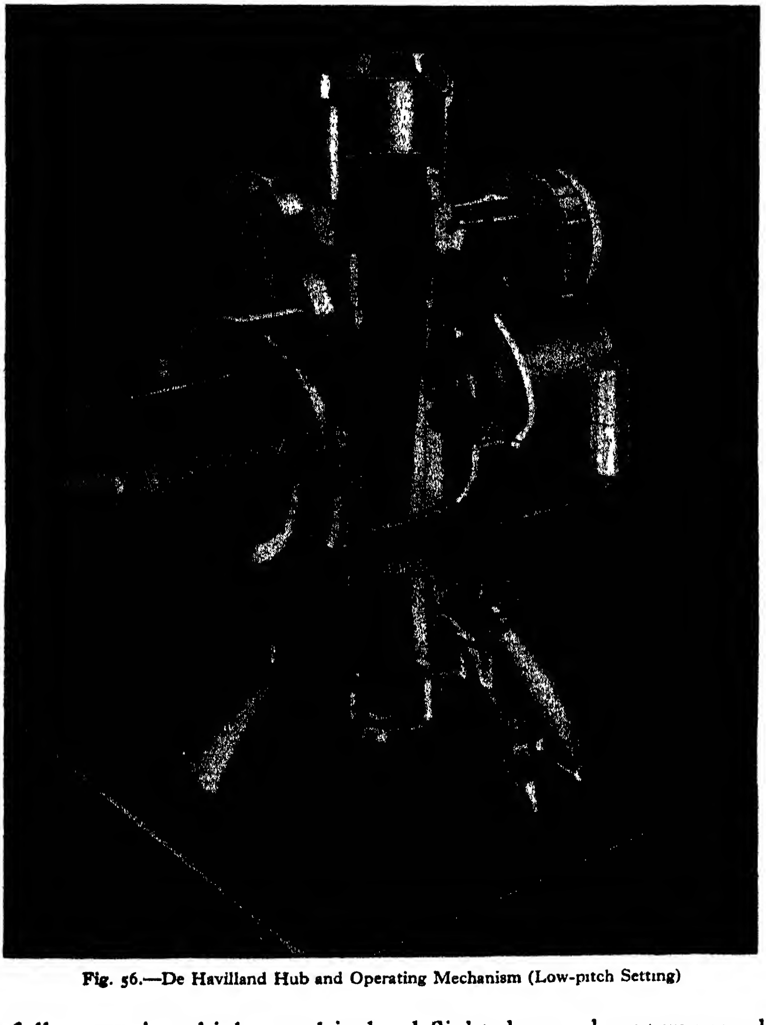

Fig. 56.—De Havilland Hub and Operating Mechanism (Low-pitch Setting)

same full power into high speed in level flight, has such enormous advantages that it has become recognized as a practically indispensable component of the efficient aeroplane of to-day. The advantage gained may be expressed in better all-round performance or in greater payload or in the use of an engine or engines of smaller output, or in any combination of these.

The possible increase in output for take-off and climb by the use of this device is shown in fig. 55, which refers to the performance of the Gipsy Major Series II engine.

Figs. 56 and 57 show cut-away sections of the De Havilland hub and operating mechanism for the controllable-pitch airscrew. Fig. 56 shows the mechanism in low-pitch setting for take-off and climb. Fig. 57 shows the mechanism in high-pitch setting for cruising at maximum level flight. In fig. 57 the oil pressure has been released, allowing the counterweights

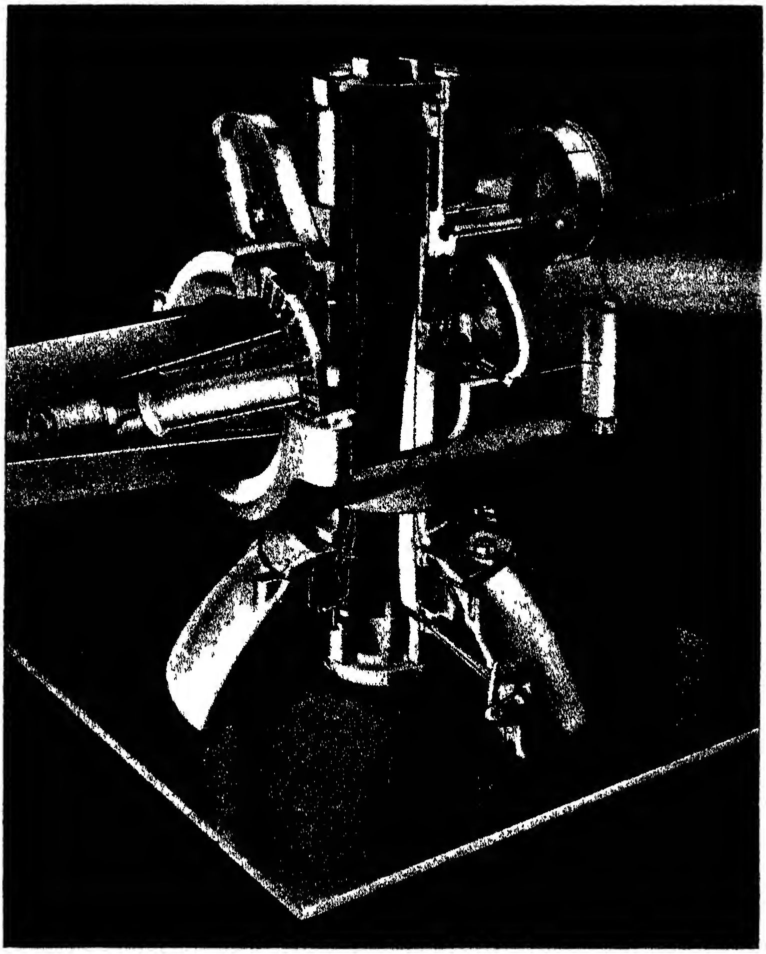

Fig. 57.—De Havilland Hub and Operating Mechanism (High-pitch Setting)

to rotate the blades in their housings until they assume the high-pitch position, which they will hold until oil is again admitted into the cylinder.

The mechanism employed is quite simple and has few working parts. Oil from the engine lubricating system is used to force outward an axial cylinder, external to the airscrew hub, which turns the blades to low pitch by means of roller cams actuating two short levers attached to the blade roots. When the oil pressure is released, centrifugal counterweights on these levers rotate the blades into coarse pitch for level flight. Mechanism within the counterweights permits the blade angles to be adjusted to any two desired pitches within the arc of total movement. The coarse and fine settings may be made independently of each other.

In the event of complete oil-pressure failure the blades remain in, or take up, the high-pitch position under the pull of the counterweights, and the aircraft is landed, in effect, with a fixed-pitch airscrew. The change from high to low pitch, or vice versa, however, occupies an appreciable interval of time, usually from 5 to 8 seconds, and the movement is accompanied by a change of the note of the airscrew, so that even should oil failure occur during a steep climb the pilot is warned and has ample time to flatten out and avoid a stall.

The Portsmouth-Johannesburg Race in September, 1936, was won by Mr. C. W. A. Scott with Gipsy Six Series II engines. Mr. Scott made a fine flight of 6450 miles in 53 hours. Captain S. S. Halse in this race made a record-breaking flight of 5650 miles in 36 hours until forced down by navigational difficulties. He used the same type of engine, and in both cases C.P. airscrews were fitted. Both pilots reported that their engines and airscrews had functioned flawlessly, and subsequent examination showed them to be in perfect condition. In a flight of long stages, the importance of being able, by means of the C.P. airscrew, to take-off from high-altitude tropical aerodromes in machines which are heavily loaded with fuel is strikingly emphasized.

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