DIESEL ENGINE DESIGN

by

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FIFTH EDITION

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PREFACE TO THE FOURTH EDITION

In preparing this Fourth Edition it has been necessary to take account of the rapid development of the Diesel Engine during the last decade in several directions, e.g., airless injection, supercharging, the application of light high speed units to road and rail traction and even aircraft, the use of high-powered double acting, 2 cycle engines for marine service, etc. Whilst there is ample scope for monographs on the various types of Diesel Engines, the Author considered that the purpose of this book would best be served by considering the functions and details of large and small engines side by side. Prominence is therefore given to the principle of similitude which, in a mainly new chapter, has been applied to fundamental problems of design such as heat flow, lubrication, fuel injection, vibration stresses, etc. A new chapter on Mechanical Efficiency follows the same line of thought and charts are given for estimating the mechanical losses of engines of any size or speed.

In the chapter on engine types some representative sectional and outside views of complete engines have been given. and the Author's thanks are due to the engine builders whose names appear against these illustrations ; the Author also has pleasure in acknowledging the courtesy of the Editor of *Engineering* in giving permission to make use in Figs. 39, 151, 152, 179 and 247 of illustrations which have appeared in the above journal.

In spite of the overwhelming preponderance of airless injection in new construction, a chapter dealing with air blast injection has been retained in view of the large number of existing engines in which this system is employed. In other chapters some figures have been retained which represent constructions now seldom used but which it might be unwise to term obsolete, since the history of the Diesel Engine contains many examples of ideas revived after a period of disuse.

The tendency towards higher piston speeds has increased the importance of vibration problems ; accordingly new chapters have been added on Balancing, Torsional and Transverse

Vibration, and Noise. New chapters have also been added on Lubrication, the Cooling Water System and Supercharging.

These and other changes have (in spite of some deletion) necessarily increased the size of the book, which now contains twenty-six chapters as against fifteen of the previous edition.

The Author wishes to make grateful acknowledgment of corrections and fruitful suggestions received from time to time from readers, including reviewers for the technical press. It gives him special pleasure to record his indebtedness to a colleague, Mr. Norman McCallum, for his generous and quite invaluable help in proof reading and correction, and to the Author's wife for compiling the index and lists of contents and illustrations.

H. F. P. P.

Belfast, 1936.

PREFACE TO THE FIFTH EDITION

A LARGE part of the fourth edition shared in a very general destruction by enemy action before reaching the public. In this fifth edition there has been a general revision, parts of four chapters have been rewritten, and many figures have been replaced with new illustrations typical of later practice, including several sectional views of complete engines.

In addition, a new chapter has been added in the form of a general survey. In this chapter the author has reviewed the main problems of design, shown that the performance of a typical modern Diesel engine can be closely accounted for by known physical principles, and attempted to define the place of the Diesel engine in a scheme of classification of known types of the internal combustion engine, with special reference to the various possible heat engine cycles, and the newer possibilities resulting from the recent development of the gas turbine.

Whilst these developments may produce limits in certain directions, there is no doubt of an expanding future for the Diesel engine in its numerous forms for service on land and water. Indeed, now as never, the wind is set fair for the Diesel engine and Diesel engine design.

It gives the author special pleasure to be able, once again, to express his indebtedness to Mr. Norman McCallum, M.A. (Cantab), for his generous help with the proofs.

H. F. P. P.

Belfast, 1947.

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DIESEL ENGINE DESIGN

CHAPTER I

FIRST PRINCIPLES

The Diesel Principle.—The characteristic feature of the Diesel Engine is the injection of oil fuel into air which has been previously compressed by the rising of a piston to a pressure corresponding to a temperature sufficiently high to ensure immediate ignition of the fuel.

In the course of the pioneer experiments by which the commercial practicability of this engine was demonstrated, it was found advantageous to effect the injection of the fuel by a blast of air, and this feature was retained in all Diesel Engines until the lapse of the original patents.

At the present date there exists a class of high-compression oil engines operating on the Diesel principle, in which the injection of oil is effected by mechanical means without the assistance of an air blast. These engines have been variously termed "Solid Injection Engines," "Cold Starting Heavy Oil Engines," "Airless Injection Engines." For our purpose the term "Airless Injection Diesel Engine" will serve to distinguish this class from that of the true Diesel Engine as defined below. Special features in connection with the design of both types will be considered in separate chapters; the remaining chapters deal for the most part with matters which are equally relevant to either type.

The features which characterise the true Diesel Engine, in the correct use of the term, are now understood to be the following:—

(1) **Compression sufficient to produce the temperature requisite for spontaneous combustion of the fuel.**

(2) **Injection of fuel by a blast of compressed air.**

(3) A maximum cycle pressure (attained during combustion) not greatly exceeding the compression pressure, i.e. absence of pronounced explosive effect.

Item 3 is deliberately worded somewhat broadly as the shape of a Diesel indicator card is subject to considerable variation under different conditions of load, blast air pressure, fuel valve adjustment, etc.

In the earlier days of Diesel Engine construction the square top indicator card, shewing a period of combustion at constant pressure, was considered the ideal to aim at. It has since been found that a card having a more peaked top is usually associated with better fuel consumptions. When tar oil is used as fuel the square top card appears to be almost out of the question.

It should further be remembered that the existence of a period of combustion at constant pressure is no guarantee that all the combustion takes place at that pressure. This ideal is never realised. Combustion probably proceeds slowly well after half stroke, even under the most favourable conditions.

Compression Pressure.—The height to which compression is carried is governed by the following considerations :—

(1) The attainment of the requisite temperature.
(2) The attainment of a desirable degree of efficiency.
(3) Mechanical considerations.

Considerations of temperature for ignition fix the lower limit of compression at somewhere in the neighbourhood of 400 lb. per sq. in. The temperature actually attained depends on the initial temperature of the intaken air and the heat lost to the jacket during compression, so it is clear that the temperature attained on the first few strokes of the engine will be considerably lower than the value it assumes after the engine has been firing consecutively for some time.

As regards efficiency, it is well known that increasing the degree of compression beyond certain limits does not very materially increase even the theoretical efficiency.

In practice the compression most usually adopted is about 500–550 lb. per sq. in. for four stroke engines. For two stroke engines the compression is sometimes in the neighbourhood of 600 lb. per sq. in., owing to the fact that the charge of air delivered by the scavenge pump may itself be at a pressure slightly above atmospheric. The compression ratio in the working cylinder itself needs to be sufficient to attain ignition temperature since at starting up the scavenge pressure is nearly atmospheric on account of the small pressure required to pass the charge through the ports or valves in the time available. The mechanical considerations which limit the compression are numerous, and some are mentioned below.

Higher compression involves :—

(1) Heavier load per sq. in. of the piston and necessitates massive construction of all the main parts.

(2) More highly compressed air for injection and consequently increased trouble with the air compressor, and its valves particularly.

(3) Increased wear of cylinder liners due to increased pressure behind the piston rings.

Compression Temperature.—With a compression of 500 lb. per sq. in. in a fair-sized four cycle cylinder working under full load conditions the compression temperature is about 1200° F. On starting the engine from a cold state the compression pressure and temperature are considerably lower owing to the cold state of the cylinder walls and the piston crown.

In addition to this the injection of cold blast air with the fuel in the proportion of about 1 lb. of blast air to 12 lb. of suction air still further reduces the temperature apart from the probability that the blast air has momentarily a local cooling effect in the zone of combustion.

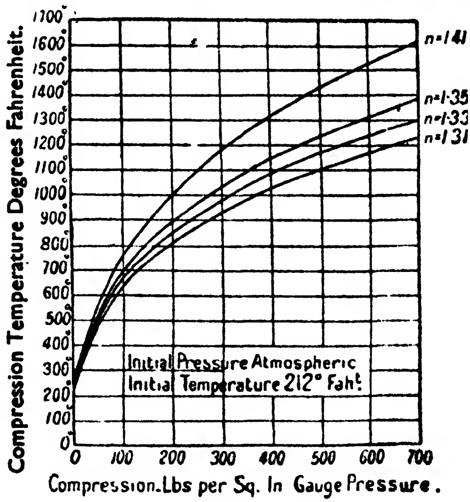


Fig. 1.

The middle curve (Fig. 1) shews graphically the connection between the compression temperature and compression pressure on the assumptions that :—

(1) The initial temperature of the intaken air is 212° F.

(2) That the exponent in the equation PV^n=const. is 1·35.

These assumptions correspond approximately to the conditions obtaining with a heavily loaded engine of fair size—say an 18″ cylinder with uncooled piston.

The noteworthy point about this curve is the slowing down of the rate of increase of temperature with pressure as the latter increases. Expressed mathematically $\frac{dT}{dP}$ diminishes as P increases.

Airless Injection Diesel Engines are characterised by:—

(1) Compression sufficient to produce the temperature requisite for spontaneous combustion of the fuel.

(2) Injection of fuel by mechanical means in the neighbourhood of top dead centre.

The combustion may approximate either to the constant pressure or constant volume cycle. Good results are often obtained by so-called "mixed" combustion consisting of a period of combustion at constant volume followed by a period of combustion at nearly constant pressure.

Owing to the absence of the cooling effect of an air blast, airless injection engines can safely be designed with lower compression pressure than air blast engines without detriment to their ability to start readily from cold. The compression pressure is commonly 400 to 500 lbs. per sq. in.

The Four Stroke Cycle.—The well-known four stroke cycle consists briefly of:—

(1) The Suction Stroke.
(2) The Compression Stroke.
(3) The Combustion and Expansion Stroke.
(4) The Exhaust Stroke.

These are considered in detail below.

Suction Stroke.—If the engine crank is considered to be at its inner dead centre and just about to begin the suction stroke, the suction valve is already slightly open. In steam engine parlance it has a slight lead. At the same time the exhaust valve, which has been previously closing on the exhaust stroke, has not yet come on its seat. The result of this state of affairs is that the rapidly moving exhaust gases create a partial vacuum in the combustion space and induce a flow of air through the suction valve, thus tending to scavenge out exhaust gases which would otherwise remain in the cylinder.

As the piston descends its velocity increases and reaches a maximum in the neighbourhood of half stroke. At the same time the suction valve is being lifted further off its seat and attains its maximum opening also in the neighbourhood of half stroke. The lower half of the suction stroke is accompanied by a more or less gradual closing of the suction valve, which, however, is not allowed to come on its seat until the crank has passed the lower dead centre by about 30°. At the moment when the crank is passing the lower dead centre the induced air is passing through the restricted opening of the rapidly closing suction valve with considerable velocity and an appreciable duration of time must elapse before the upward movement of the piston can effect a reversal of the direction of flow through the suction valve. It will be clear from the above that owing to the effect of inertia more air will be taken into the cylinder in the manner described than by allowing the suction valve to come on its seat exactly at the bottom dead centre. The exact point at which the suction valve should close is doubtless capable of approximate calculation, but is usually fixed in accordance with current practice or test-bed experiments.

Compression Stroke.—The piston now rises on its up stroke and compresses the air to about 500 lb. per sq. in., the clearance volume necessary for this compression being about 8% of the stroke volume. During the compression the temperature rises and a certain amount of heat is lost to the cylinder walls and cylinder cover. The final compression temperature is in the neighbourhood of 900° F. to 1200° F.

Combustion and Expansion Stroke.—At the upper dead centre, or slightly previous thereto, the injection valve opens and fuel oil is driven into the cylinder and starts burning immediately. The actual point at which the fuel enters the cylinder is not quite certain, as there is inevitably some lag between the opening of the injection valve and the entrance of fuel. The point at which the fuel valve of an air injection engine starts to open, as determined by a method described below, varies from about 3° (slow speed engines) to 14° (high speed engines). The method of determining the point of opening of the fuel valve is as follows :—

With the engine at rest, air at about 100 lb. pressure is turned on to the injection valve and then communication with the blast air-bottle is cut off to prevent unnecessary waste of air and the possibility of the engine turning under the impulse of the air which is subsequently admitted to the cylinder. The indicator cock is now opened and the engine slowly barred round by hand until the air is heard to enter the cylinder by placing the ear to the indicator cock. The position of the engine when this occurs is the nearest possible approximation to the true point of opening, assuming the operation has been carefully done.

The duration of the fuel valve opening is usually about 48°, and in the majority of engines is fixed for all loads.. It is evident that at light load the opening is longer than necessary, and in some designs the duration of opening is regulated by the governor in accordance with the load.

The fuel valve timing of airless injection engines varies rather widely; it depends on the piston speed of the engine and the shape of indicator card which it is desired to obtain.

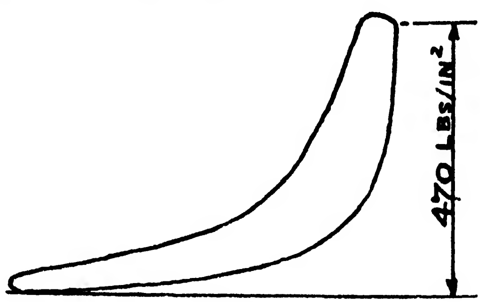


Fig. 2.

The duration of fuel valve opening at full load varies in different instances from about 14 to 30 crank-shaft degrees. Injection may begin as late as 5° or as early as 30° or more before top dead centre.

The combustion is by no means complete when the fuel valve closes, and usually continues in some measure well past the half stroke of the engine. This is known as "after burning," and takes place with the very best engines in the best state of adjustment. Exaggerated after burning is the surest sign of maladjustment, and makes itself apparent by abnormally high terminal pressure at the point at which the exhaust valve opens, and is readily detected on an indicator card by comparison with that taken from an engine in good adjustment.

As will be shewn later, the presence of "after burning" is most clearly seen on an Entropy Diagram.

Expansion continues accompanied by loss of heat to the cylinder walls until the exhaust valve opens.

Exhaust Stroke.—The exhaust valve opens about 50° before the bottom dead centre, in order that the exhaust gases may effect a rapid escape and reduce the back pressure on the exhaust stroke. The pressure in the cylinder when the exhaust valve starts to open is about 40 lb. per sq. in. with an engine working with a mean indicated pressure of 100 lb. per sq. in. The temperature of the exhaust gases at this point is somewhere in the neighbourhood of about 1600° F. and the velocity is consequently very high.

The pressure falls nearly to atmospheric shortly after the bottom dead centre has been passed and the back pressure during the remainder of the exhaust stroke should not be more than about 1 lb. per sq. in., or less. Excessive back pressure may arise from :—

(1) Insufficient diameter or lift or late opening of exhaust valve.
(2) Exhaust pipe too small in diameter.
(3) Obstructions or sharp bends in the exhaust pipe or silencer.
(4) Interference by another cylinder exhausting into the same pipe.

It is interesting to note that owing to the higher velocity of air at high temperature per unit pressure difference the back pressure is more at light load than at full load.

Indicator Cards.—Figs. 2 and 3 shew typical indicator cards taken with a heavy and a light spring respectively.

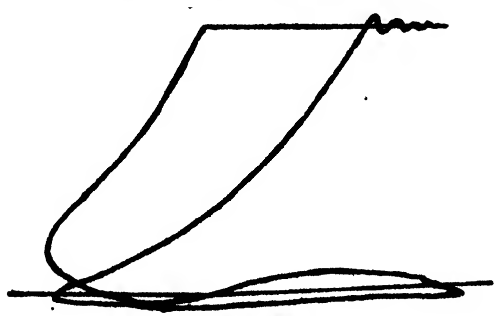


FIG. 3.

The latter is particularly useful for investigating the processes of suction and exhaust.

It is to be observed that in Fig. 3 the compression is seen to start at a point which is indistinguishable from the bottom dead centre, thus indicating a volumetric efficiency of practically 100%. This is to be regarded as a normal state of affairs, obtainable with both high speed and low speed engines. The volumetric efficiencies of internal combustion engines are frequently quoted at figures varying between about 95% for slow speed engines to 80% for high speed engines. The former figure is reasonable, but the latter can only be due either to imperfect design (or adjustment) of the engine or to erroneous indicator cards. The use of too weak a spring in the indicator may lead to a diagram shewing not more than 60% volumetric efficiency, owing to the inertia of the indicator piston, etc. Consequently, fairly stiff springs are to be preferred.

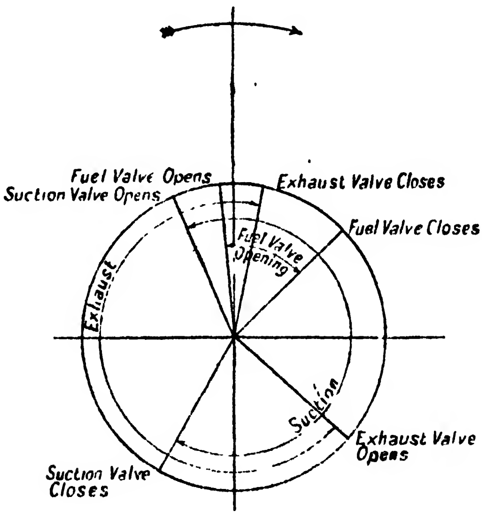


FIG. 4.

It may be worth remarking here that the term "volumetric efficiency" is often used to denote the ratio of the weight of air drawn into the cylinder to the weight of air at normal temperature and pressure (°C and 760 m/m. mercury) which would fill the swept volume of the cylinder. According to another usage the actual temperature and pressure of th surrounding air is used instead of N.T.P. On either of these systems the " volumetric efficiency " is always materially below 100% unless supercharging is used.

Valve Setting Diagram.—Fig. 4 is a typical valve setting diagram for a four stroke engine, and shews the points relative to the dead centres at which the various valves open and close.

Representative valve setting figures are given under :—

Inlet valve opens.	20° before T.D.P.
,, ,, closes.	30° after B.D.P.
Exhaust valve opens.	50° before B.D.P.
,, ,, closes.	15° after T.D.P.

The fuel valve opening shewn in Fig. 4 refers to air blast injection.

It will be noted that the inlet and exhaust valve periods overlap to the extent of about 35 crank-shaft degrees. With supercharged engines with exhaust driven turbo-blowers the overlap is increased to about 120° in order to scavenge the clearance space with fresh air and to effect a certain amount of air cooling of the walls of the combustion space.

The Two Stroke Cycle.—As its name implies, the two stroke cycle is completed in one revolution of the engine. The revolution may roughly be divided into three nearly equal parts :—

(1) Combustion and Expansion.
(2) Exhaust and Scavenge.
(3) Compression.

The exhaust and scavenge take place when the piston is near the bottom dead centre, and consequently only very small portions of the expansion and compression strokes are lost, in spite of the fact that nearly 120° of the crank revolution are occupied with exhaust and scavenge. This point is clearly seen on reference to Fig. 5.

Exhaust Period.—The exhaust starts when the piston uncovers slots in the cylinder wall or exhaust valves begin to open. The point at which this happens is different in different designs of engine, an average being about 20% of the stroke before bottom dead centre. The exhaust ports are usually of large area, and consequently the pressure falls to atmospheric very rapidly. The period required for this process naturally depends on the port area and the piston speed, and average figures are about 20° to 30°.

It is well to dwell carefully on the state of affairs at this point.

During exhaust the cylinder pressure has fallen from about 55 to about 15 lb. per sq. in. absolute, and there is no reason

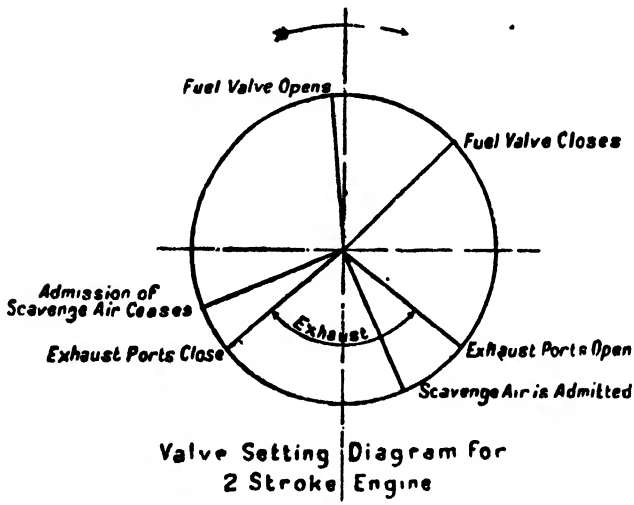


Fig. 5.

to suppose that the remaining exhaust gases have fallen greatly in temperature. (Given adiabatic expansion, the fall in absolute temperature is less than 20%.) The conclusions are, therefore :—

(1) Something like 50% by weight of the gases have effected their escape.

(2) The remaining gases are rarefied compared with atmospheric air.

Scavenge Period.—The scavenge air is admitted by ports or valves (or both), and the instant at which admission starts is timed to coincide with that at which the cylinder contents attain appreciably the same pressure as the scavenge air, or a trifle less. The incoming scavenge air is supposed to sweep the remaining exhaust gas before it and so fill the cylinder with a charge of pure air by the time the piston has covered the exhaust slots on the up stroke. Actually certain processes take place which do not enter into the ideal programme. Some of these are :—

(1) A certain amount of mixing between the incoming scavenge air and the retreating exhaust gases.
(2) Short circuiting of scavenge air to the exhaust pipe before all the exhaust gas has been expelled.

The effects of both these processes are minimised by providing a large excess of scavenge air. The figure adopted for the ratio of scavenger volume to cylinder volume is about 1·4 in modern designs, securing good stratification and avoiding undue loss of the fresh charge.

There are a number of different systems in use for controlling the release of the exhaust gases and the admission of scavenge air, and some of these are discussed below with reference to Fig. 6.

Simple Port Scavenge.—In this system the scavenge air is admitted by means of ports in the cylinder liner opposite a row of similar ports for the exhaust (see Fig. 6A), the piston top being provided with a projection to deflect the scavenge air to the top of the cylinder. This system is simple but possesses some disadvantages which are enumerated below.

(1) The scavenge air slots have to be made shorter than the exhaust slots in order that the cylinder pressure may fall to the same value as the scavenge air pressure before the piston begins to uncover the scavenge slots. This entails the latter being covered by the piston on its upward stroke before the exhaust ports are covered, and consequently the pressure at the beginning of compression can barely exceed the pressure in the exhaust pipe. There is also a possibility of exhaust gases working back into the cylinder.

(2) The projection on the top of the piston necessitates a specially shaped cylinder cover in order to provide a suitable shape for the combustion space.

Engines provided with this system of scavenge are only suited for a relatively low mean indicated pressure of about 80 lb. per sq. in.

Fig. 6B shews a variation of the above arrangement, in which no lip is provided on the piston crown, but the scavenge air inlet ports are given a steep slope upwards. Fig. 6C shows an arrangement in which the scavenge air ports and the exhaust ports are arranged on the same side of the cylinder. This arrangement is used on engines built by the M.A.N.

Valve Controlled Port Scavenge.—This system, associated

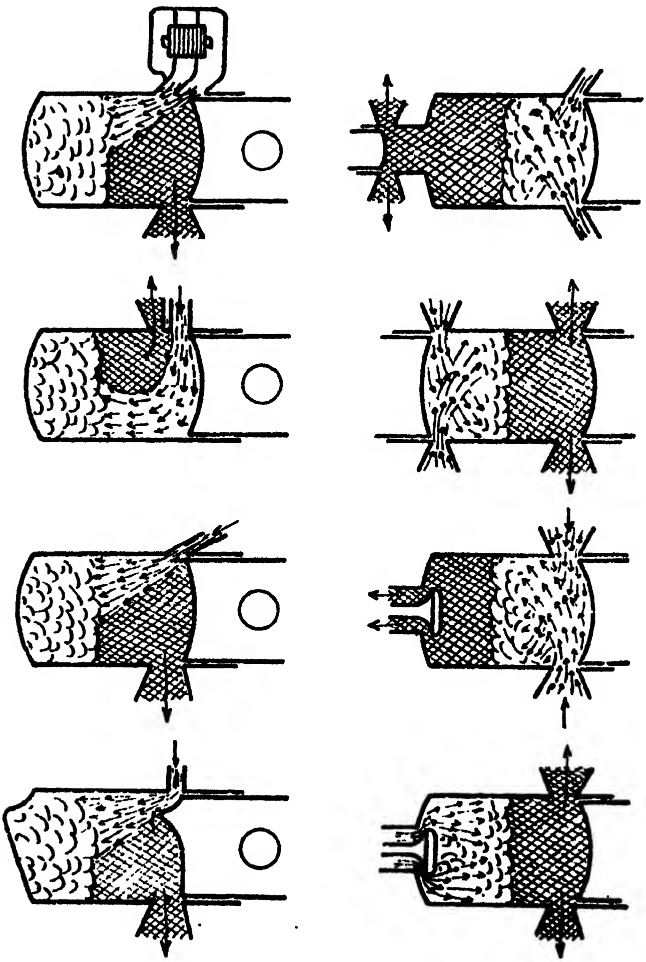


Fig. 6.

with the names of Sulzer, Fiat and others, is shown diagrammatically in Fig. 6D.

The air ports, or a certain number of the air ports, are so situated that they are uncovered before or simultaneously with the exhaust ports, but are controlled by a valve of the non-return, rotary, or other type, in such a way that communication does not exist between the cylinder and the scavenge pipe until the exhaust ports have been uncovered for a sufficient period to allow the cylinder pressure to fall to or below the scavenge pressure. On the upward stroke the controlling valve remains open, so that the cylinder is in communication with the scavenge pipe until the scavenge slots are covered by the piston. With this arrangement the cylinder is filled with air at approximately the pressure of the scavenge pipe at the instant at which the piston covers the scavenge ports; consequently the effective volumetric efficiency of the cylinder may be 100% or over and higher mean pressures are practicable than with the simple port type.

Cylinder Cover Valve Scavenge.—In this system the scavenge air is admitted by means of one to four valves located in the cylinder cover, and avoids some of the disadvantages of the simple port scavenge. (Fig. 6E.)

By allowing the scavenge valves to close after the exhaust ports have been covered by the piston, the cylinder may become filled with air at scavenge pressure before compression starts, and consequently such a cylinder is capable of developing a higher mean effective pressure. The greatest drawback to this system is the complication of the cylinder cover, and this appears to be rather serious. It is not an easy matter to design a cover to accommodate several valves and passages and at the same time secure the necessary conditions for durability under exposure to the strong heat flux which the two stroke cycle involves when high mean pressures are used.

Certain precautions are necessary to obtain good scavenging efficiency. There is a tendency for streams of scavenge air from each valve to bore through the gases in the cylinder and escape through the ports. This tendency can be largely overcome by attention to details of the valve orifices. One method is to provide each valve head with a lip projecting beyond the valve face.

An arrangement similar to that shown in Fig. 6E can be used with the flow reversed, i.e. with scavenge air entering at the ports and the exhaust leaving by the valves in the head. One advantage of such an arrangement is the large area available for the supply of scavenge air. This is an important point when high piston speeds are required. This system has been extensively used by several builders.

Opposed Piston Engines.—Two pistons reciprocate in opposition in a single cylinder with the combustion space between the pistons. One piston uncovers the exhaust ports and the other piston the scavenge air ports about 20 to 30 crank-shaft degrees later. This scheme as shewn in Fig. 6G is associated with the names Junker, Doxford and Fullagar.

If the engine is required to run in one direction only, it is possible so to arrange the angular disposition of the cranks which operate the two pistons, that although the exhaust ports are uncovered before the scavenge air ports, the latter are closed on the return stroke later than or simultaneously with the exhaust ports.

It is usual so to direct the scavenger air ports that the air in the cylinder receives a whirling or vortex motion, which helps to distribute the fuel throughout the air charge during the process of injection.

In one of the Burmeister & Wain's two stroke designs shown diagrammatically in Fig. 6H, the piston which controls the exhaust ports has a diameter about one half and a stroke rather less than half that of the main piston which controls the scavenge air ports. The auxiliary piston is operated either by cranks on a lay shaft or by eccentrics on the crank-shaft. The angular settings are such that the exhaust ports are uncovered the necessary amount in advance of the air ports.

In later designs of B. and W. two cycle engines both single acting and double acting, the exhaust pistons are made the same diameter as the main pistons and operated by means of eccentrics on the crankshaft (see Fig. 8).

Other Types of Scavenging.—A number of other methods or variations of the methods described above are in use or have been proposed, for example designs involving sliding or rotating sleeves arranged between the piston and the cylinder in the manner familiar in connection with four stroke petrol engines; designs incorporating a rotating or oscillatory valve in the exhaust discharge port; designs in which the piston is provided with a concentric extension sleeve passing through a gland in the cylinder head and provided with slots for the escape of the exhaust.

Literature.—Chalkley, A. P., "Diesel Engines."—Constable.
Bird, A. L., "Oil Engines."—Methuen.
Körner, K., "Der Bau des Dieselmotors."—Springer.
Sass, F., "Kompressorlose Dieselmaschinen."—Springer.
Zeman, J., "Zweitakt-Dieselmaschinen."—Springer.
Shepherd, "Diesel Engine Design."—Wiley.
Dicksee, C. B., "The High Speed Compression Ignition Engine."—Blackie.

CHAPTER II

TYPES OF DIESEL ENGINES

Classification.—Existing types of Diesel Engines may be divided into groups in various ways, according as they are :—

(1) Stationary, Marine, Locomotive, Automobile, Aircraft type, etc.
(2) Four Cycle or two Cycle (4 c. or 2 c.).
(3) Single Acting or Double Acting (S.A. or D.A.).
(4) Vertical, Horizontal, Vee, Radial, etc.
(5) Slow Speed or High Speed.
(6) Air Injection or Airless Injection.

High Speed and Slow Speed.—Piston speed, Vp, is often used as a measure of speed of rotation in relation to size. If S = stroke in inches and n = revolutions per minute then,

$$Vp = \frac{n\,S}{6} \quad \text{ft./min.} \ldots\ldots\ldots\ldots\ldots\ldots\ldots (1)$$

A criterion which has some advantages over the above is to substitute for S the value of $\sqrt[3]{B^2S}$ where B = Bore in ins. The speed found thus may be called the "cylinder speed," Vc ; then :—

$$Vc = Vp\,\frac{\sqrt[3]{B^2S}}{S} \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (2)$$

Values of Vc/Vp are tabulated under for various values of S/B, the stroke bore ratio :—

$\frac{S}{B}$ = 1	1·2	1·5	1·7	2	2·5	3
$\frac{Vc}{Vp}$ = 1	0·885	0·763	0·707	0·634	0·545	0·481

The powers of similar engines running at the same piston or cylinder speed and working with the same mean pressure vary inversely as the (R.P.M.)[2]. This gives another useful measure of rotational speed in relation to power, which we may call "horse power speed," defined as follows :—

$$n_1 = n\sqrt{\text{B.H.P. per cylinder}} \quad \ldots\ldots\ldots\ldots (3)$$

The quantity n_1 is the R.P.M. of a one B.H.P. cylinder similar to the engine considered and running at the same piston speed or cylinder speed and the same brake mean pressure.

Four Stroke Cycle Stationary Engines.—Fig. 7. The earlier types of stationary Diesel engine had independent cylinders and ring lubricated bearings. The stroke bore ratio was usually about 1·4 and piston speeds ranged from 600 to 900 ft./min. The values of n_1 were about 1200 to 2000. For example, the 40 B.H.P. cylinder ran at 200 R.P.M. and the 165 B.H.P. at 150 R.P.M. Such engines are still made but with rather higher values of n_1.

Present practice for vertical engines is in favour of a totally enclosed crank-case and forced lubrication. Values of n_1 range from about 2000 to 5000, with piston speeds from about 800 to 1600 ft./min. A few typical examples are tabulated under.

B.H.P. per cylinder.	Bore B ins.	Stroke S ins.	R.P.M. n.	Vp ft./min.	Vc ft./min.	n_1
5	$4\frac{1}{8}''$	6″	800	800	710	1800
20	6″	8″	900	1200	1090	4000
50	$7\frac{1}{2}''$	12″	750	1500	1280	5300
100	$15\frac{1}{2}''$	17″	350	1000	970	3500
125	15″	20″	375	1250	1130	4190
150	18″	24″	250	1000	910	3060
250	21·7″	39·4″	175	1150	950	2770
280	$21\frac{1}{2}''$	22″	300	1100	1090	5000

For driving electric generators fairly high values of Vc and n_1 are preferred on account of saving in space and reduced cost of dynamo. This is admissible when the engines will operate under skilled supervision. For many industrial purposes simple robust and relatively foolproof power units are desirable and for such engines relatively low values of Vc and n_1 are suitable. Engines designed for such low values are not necessarily expensive per B.H.P. since cheaper materials and simpler constructions are used than would be allowable with higher values.

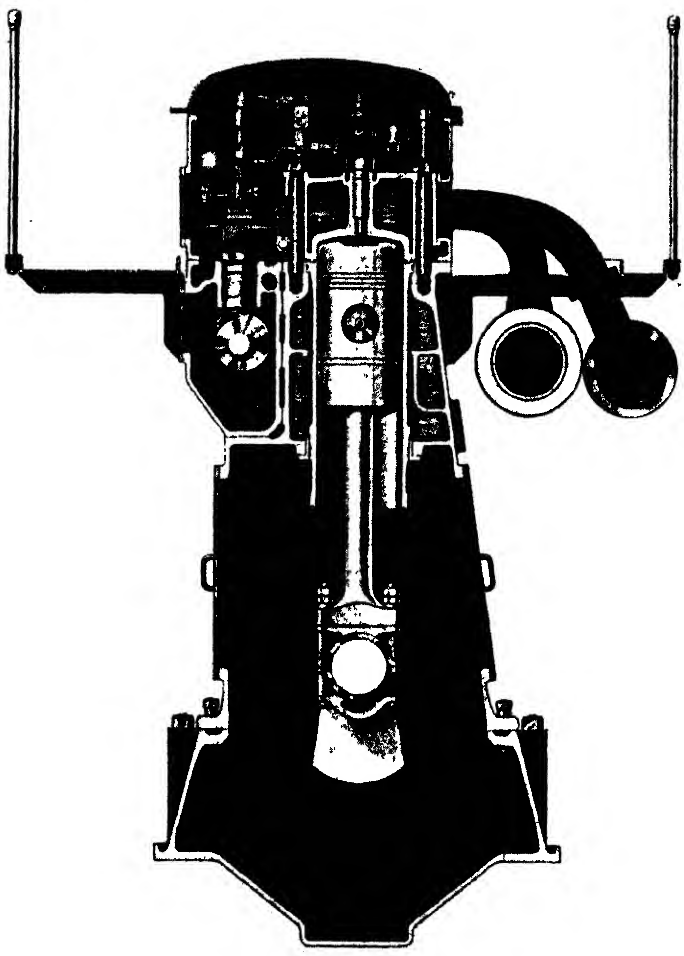


Fig. 7a.
4 Stroke Trunk Engine.
(Harland and Wolff.)

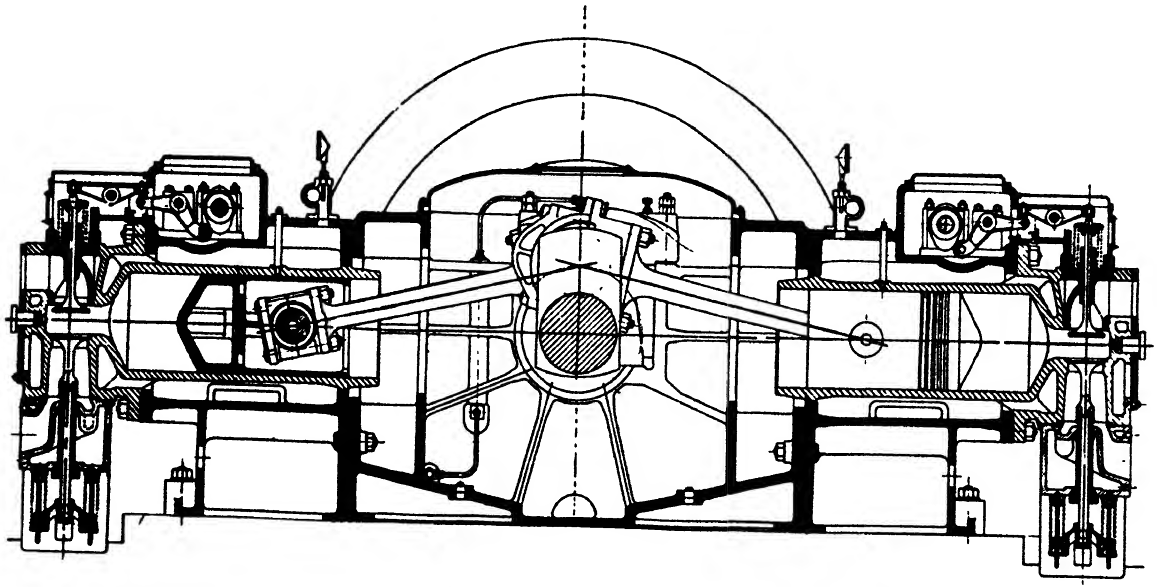


Fig. 7b.
Crossley-Premier Totally Enclosed Vis-a-Vis Type Horizontal 4 Stroke Oil Engine.

The above table refers to normal, i.e. unsupercharged engines. With supercharging the n_1 values may be increased by about 15% to 30% or more.

Large numbers of single cylinder units of about 5 to 20 B.H.P. are made for industrial purposes, particularly for export to undeveloped countries where a demand exists for a simple power unit capable of being looked after by unskilled operators. Two, three and four cylinder engines of powers from 10 to 200 B.H.P. are useful for a large number of industrial uses at home and abroad.

For driving electric generators of say 200 KW. and upwards six and eight cylinder engines are preferred on account of good balance and freedom from vibration. Five and seven cylinder engines are also used.

All powers from 5 to about 2500 B.H.P. can be satisfactorily met by standardised four cycle trunk Diesel engines.

Horizontal Engines—Four Stroke Cycle.—Where headroom and overhauling facilities are limited horizontal engines are sometimes preferred. Powers range from about 10 to 220 B.H.P. per cylinder. Some typical sizes and speeds are tabulated under :—

B.H.P. per cylinder .	10	20	50	90	130
R.P.M. . . . . n .	390	290	240	220	185
n_1 .	1230	1300	1700	2100	2110

The stroke bore ratio is commonly 1·5 to 1·75.

Large numbers of single cylinder engines are manufactured but multicylinder horizontal engines having two to sixteen cylinders are made either side by side (up to six cylinders) or *vis-à-vis* (up to sixteen cylinders).

Airless injection is almost invariably used and the general design usually owes a good deal to previous gas engine or hot bulb oil engine experience. The relatively low n_1 value enables a considerable degree of simplicity and accessibility to be achieved without detriment to performance. The power range covered by horizontal airless injection Diesel engines is about 10 to 3500 B.H.P.

Two Stroke Cycle Stationary Engines.—Two stroke engines cover the entire range of powers attempted with the Diesel engine from about 5 B.H.P. to 22,000 B.H.P. The larger sizes from about 2000 B.H.P. upwards are usually based on designs developed for marine purposes, but run at rather higher revolutions. Typical revolutions for high power double acting two stroke cylinders for stationary purposes are tabu lated under :—

B.H.P. per cylinder .	600	800	1100	2500
Bore in ins. . . . . B .	20·5	23·6	27·6	33·0
Stroke in ins. . . . . . S .	27·6	35·5	47·2	59·1
R.P.M. . . . . n .	215	160	125	115
Piston Speed . . . . . Vp .	990	950	980	1135
n_1 .	5300	4500	4200	5750

Single acting two stroke engines developing from 5 to 600 B.H.P. per cylinder are made in a wide variety of types. From 100 B.H.P. and upwards the cross head type of piston has been very commonly used, but the whole range can be adequately covered by engines of the trunk type. The larger engines are always fitted with some form of blower, whether reciprocating, positive rotary, or centrifugal type, capable of delivering an excess of air for scavenging purposes, thus enabling a full load indicated mean pressure of 80 to 100 lb./in.^{2} to be developed. The following table refers to engines of this general type :—

B.H.P. per cylinder .	50	100	200	400	600
R.P.M. . . . . n .	500	400	300	200	167
n_1 .	3530	4000	4230	4000	4080

With loop scavenge the stroke bore ratio is usually 1·3 to 1·75 ; with uniflow scavenge the ratio may be 1·8 or over.

For engines of small power crank-case scavenge is sometimes used when simplicity is the first consideration. Such engines develop relatively low mean indicated pressures of the order 50 lb./in.^{2} and the piston speeds are low. Typical figures are tabulated below :—

B.H.P. per cylinder .	5	12	18	30	50
R.P.M. . . . . n .	600	525	450	340	300
n_1 .	1340	1820	1910	1860	2120
B/S . . .	—	—	—	10½/12″	13½/15″

Revolutions of Marine Engines.—The highest revolutions available for a given power are usually too high to give a reasonable propeller efficiency if directly coupled. Sometimes gearing is used, but more usually the engine is directly coupled and run at revolutions which result in a reasonable propeller efficiency and a very durable engine.

Neglecting subsidiary factors, propeller efficiency is a function of

$$x = \frac{V_1^{2.5}}{n\sqrt{T.H.P.}} \dots\dots\dots\dots\dots\dots (1)$$

where V_1=speed of advance of propeller (knots)
T.H.P.=thrust horse power.

The T.H.P. may be taken as approximately equal to the B.H.P. x (propeller efficiency—0·1). The following table shows the efficiency obtainable from a normal design of 3 bladed propeller having a developed surface ratio of 0·45. The figures for efficiency should be regarded as relative rather than absolute.

X	0·04	0·08	0·12	0·16	0·20
Propeller efficiency ηp	0·61	0·66	0·70	0·72	0·73
Optimum pitch ratio	0·65	0·9	1·05	1·20	1·35
T.H.P./B.H.P.	0·51	0·56	0·60	0·62	0·63

The tabulated efficiencies are based on the assumption that the propeller diameter is not limited below its optimum diameter by draft.

There is little to be gained by an X value exceeding 0·12, but an appreciable sacrifice is incurred by using X=0·04.

Assume for the moment X=0·12 then equation 1 becomes

$$n = \frac{V_1^{2.5}}{0{\cdot}12\sqrt{0{\cdot}6\ B.H.P.}} \dots\dots\dots\dots\dots (2)$$

leading to the following values :—

Revolutions per min. Corresponding to x=0·12

B.H.P.	V_1 = knots. 5	10	15	20	25	30
10	190	1070	2960	6100	10,600	16,800
50	85	480	1230	2720	4750	7500
100	60	340	940	1920	3260	5300
500	27	150	420	860	1500	2380
1000	—	107	296	610	1060	1680
2000	—	76	210	430	750	1190
4000	—	54	148	304	530	840
10,000	—	34	94	192	336	530
20,000	—	—	66	136	238	375
40,000	—	—	47	96	168	265

The use of $X=0{\cdot}12$ would result in certain instances (e.g. slow single screw vessels) in impossibly low revolutions; multiplying the tabulated revolutions by 3 reduces the propeller efficiency by about 15%.

For single screw vessels V_1 is commonly about 0·6 to 0·7 of the speed of the vessel; for twin screw vessels about 0·8 to 0·9.

For large twin screw or quadruple screw installations x is usually about 0·12. For single screw ships of slow speed X may be as low as $X=0{\cdot}04$ with corresponding sacrifice in propeller efficiency, in order to obtain reasonable revolutions.

Four Stroke Marine Engines.—Double acting four stroke engines have been built up to 10,000 B.H.P. in 10 cylinders running at 100 R.P.M. (White Star liners *Britannic* and *Georgic*, 20,000 B.H.P. on twin screws), but the two stroke double acting engine, with its advantages in respect of weight, space and cost, has arrested the development of this type.

Single acting crosshead engines of the four stroke type have been built up to 6000 B.H.P. in 10 cylinders running at 120 R.P.M. with supercharging. Similar engines from about 200 B.H.P. per cylinder upwards with or without supercharging are built to run at revolutions varying from about 80 to 140 R.P.M.

Four stroke trunk engines are built for marine purposes in all sizes from about 10 to 400 B.H.P. per cylinder (diameters 4″ to 25″) running at 1500 or more to 130 R.P.M.

For twenty years the four stroke engine held a commanding position amongst Diesel Engines for high power marine service. Since 1934 the tide has set rather strongly in favour of the two stroke engine on account of saving in weight and space.

Two Stroke Marine Engines.—Fig. 8. For mercantile marine service any power from about 4000 to 30,000 B.H.P. per shaft can be satisfactorily dealt with by double acting two cycle engines of existing design which have passed beyond the experimental stage. For ship speeds from 15 to 20 knots the revolutions generally range from 80 to 100 for single screw and 100 to 120 for twin screw vessels. Even for war vessels the double acting two cycle engine in conjunction with gearing has shown itself capable of conforming to the exacting requirements of weight and headroom in certain classes of vessel.

For engines of the uniflow scavenge type the stroke bore ratio may be as high as 2·25; for loop scavenge the limit is about 1·75 and in certain types is commonly as low as 1·5 or less.

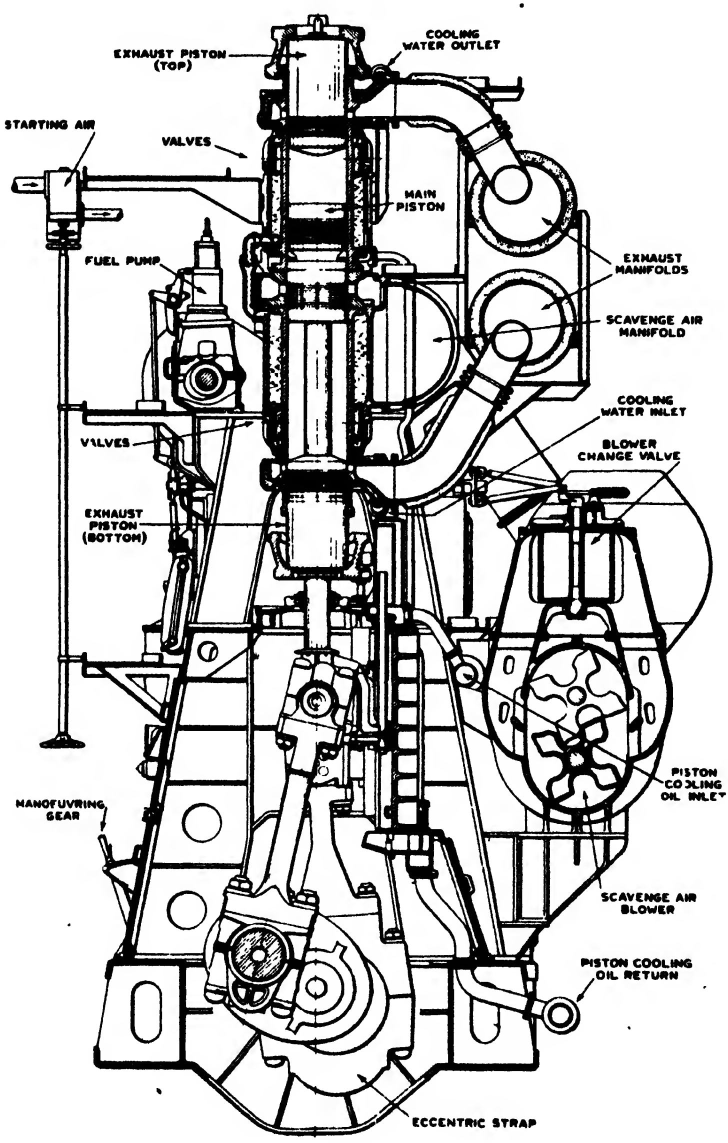


Fig. 8.

B. & W. Double Acting 2 Cycle Engine.
(Burmeister and Wain, Harland and Wolff, and other B. & W. licencees.)

The piston speeds for double acting two stroke engines of the large slow speed type range from about 700 ft./min. to 900 ft./min. in single screw installations up to 1200 ft./min. in twin screw installations. There is no serious obstacle in the way of piston speeds of 1500 ft./min., the determining factor usually being permissible revolutions for propeller efficiency.

All powers from the smallest up to about 6000 S.H.P. per shaft can be dealt with by single acting two cycle engines of the trunk or crosshead types. The revolutions depend on the speed power relation referred to above. For example suitable revolutions for a 7000 S.H.P. engine in a 17 knot twin screw vessel of 14,000 total S.H.P. would be about 115 R.P.M. and 6 to 8 relatively long stroke cylinders would be suitable. For the same power in a 21 knot vessel 250 R.P.M. would be allowable and 12 relatively short stroke cylinders would be appropriate.

Ranges of two cycle engines of all powers from about 8 B.H.P. upwards are available for lower power craft. These may be of simple robust relatively low speed type suitable for fishing boats or of the light high speed type developed along the same lines as locomotive or automobile engines, for the use of lighter higher speed boats.

Locomotive Diesel Engines.—Fig. 9. For locomotive service the first essential after reliability without attention, is lightness. The upper limit of weight is about 30 to 40 lb./B.H.P. for locomotives and about 15 to 25 lb./B.H.P. for railcars.

The engine of a Diesel locomotive with electric or mechanical or hydraulic transmission only contributes a fraction of the total weight (of the order 15% or so), but there are so many difficulties in the way of reducing the other weights that a low engine weight is essential to keep within permissible axle loads without introducing an undesirable number of axles, thus increasing costs. With railcars the effect of engine weight is cumulative. Increase of engine weight increases weight of supporting members, which increases the bolster load, which increases the weight of the bogie to carry it. The necessary degree of lightness is obtained by some of the following methods.

(1) The use of a large number of small cylinders rather than a small number of larger ones (usually 6 to 16).

(2) The adoption of a relatively low stroke/bore ratio (1 to 1·6 according to type).

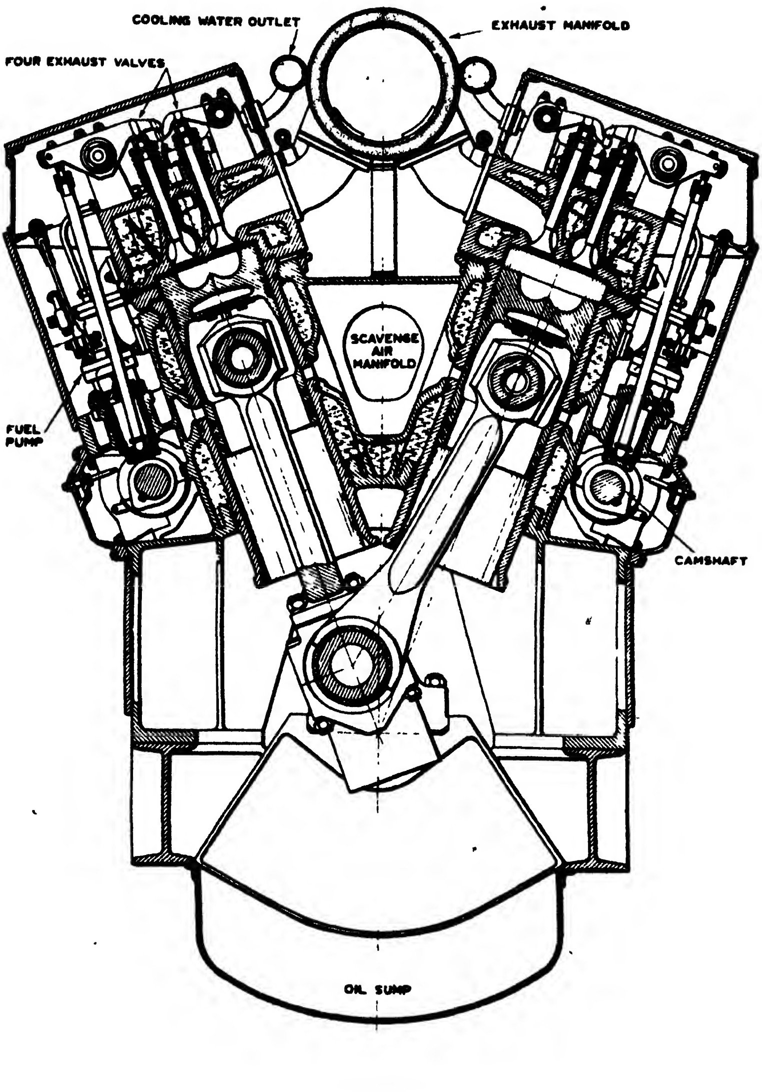


Fig. 9.

2 Cycle "V" Engine for Railway Traction.
(Harland and Wolff.)

C

(3) High revolutions involving high piston and cylinder speeds.

(4) The use of high tensile materials with correspondingly high working stresses.

(5) The use of light alloys or sheet metal for doors, covers, sumps.

(6) Elimination of bedplate by underslinging the crankshaft as in motor car practice.

The piston speeds of locomotive Diesel Engines range from about 1200 ft./min. to 1800 ft./min. With four stroke engines of short stroke bore ratio running at piston speeds of this order, light alloy pistons are usually adopted to minimise inertia effects, in particular, the stress on the big end bolts. With two cycle engines the inertia is cushioned by compression and cast iron pistons can be used to advantage as the inertia load reduces maximum stresses and bearing pressures. For a given B.H.P. per cylinder, stroke and revolutions, a well scavenged two stroke cylinder may be considerably less in diameter than a four stroke cylinder; for this reason any prescribed degree of lightness is more easily obtainable with a high duty two stroke engine than a four stroke. The water circulation system of a locomotive Diesel Engine should be designed for a generous flow resulting in a temperature rise of about 15 to 25° F. in order to reduce the required surface of the radiator. A section of the radiator is often used as an oil cooler.

The governor is frequently arranged for two to four speeds of revolution, by compressing the governor spring in stages by pneumatic or electro-pneumatic relays.

Sometimes the engine is controlled by direct regulation of the fuel pumps. In this case the governor overrides the hand control at top speed and idling speed. A separate emergency overspeed cut out may be fitted.

On account of saving in weight and space, the Vee arrangement of engine (8 to 12 or more cylinders) is useful for locomotive or railcar service.

Diesel Engines for rail traction are available in a great variety of sizes. The smallest sizes for light cars or tractors are the same models as used for road traction or stationary purposes, developing 15 to 20 B.H.P. per cylinder at 1000 to 1800 R.P.M. A typical rail car engine develops 240 B.H.P. in 6 cylinders or 320 B.H.P. in 8 cylinders at 1200 R.P.M. For powerful locomotives one to four units may be used each developing say 2000 B.H.P. in 12 cylinders arranged in Vee formation at 700 R.P.M.

Automobile Diesel Engines.—Fig. 10. A typical Diesel Engine for road traction is a 6 cylinder four stroke engine with cylinders about 4½″ bore by 6″ stroke developing a peak power of 130 B.H.P.@1800 R.P.M. A normal loading for such an engine is about 100 B.H.P.@1600 R.P.M. Engines of this size and type can run up to 2500 R.P.M. or more but with decreased efficiency and increased wear and tear. Similar engines both smaller and larger developing peak powers from about 3 to 30 B.H.P. per cylinder cover a wide range of sizes for commercial road vehicles. The smaller units run up to 4000 R.P.M. and the larger may be governed to a maximum as low as 1200 R.P.M.

Many engines of this class depart from the principle of direct injection of fuel into the main combustion chamber, by employing pre-combustion chambers, air cells and the like, with a view to increasing turbulence and making the combustion less sensitive to fuel pressure and injection timing; the object in view is successful operation over a wide range of revolutions with a relatively low fuel pressure, and a simple fuel valve with a comparatively coarse orifice.

Their general design owes more to petrol engine development than Diesel Engine practice with larger sizes. Compared with petrol engines of the same cylinder dimensions, the pistons, connecting rods and crank-shafts require to be materially stronger to withstand the higher maximum pressures obtained in the cylinders. For the same reason stronger crank-cases and cylinder heads are necessary. Torsional vibration at corresponding orders of critical speed are more severe and effective dampers are desirable unless the revolutions are limited by the governor to exclude from the working range all criticals of orders equal to or less than 6 per revolution (in a 6 cylinder engine).

Similar units with 4 to 8 cylinders running at 900 to 1500 R.P.M. are used with reverse and sometimes reducing gear for the propulsion of small marine craft.

Aircraft Engines.—Fig. 11. The chief advantages of the Diesel Engine over the petrol engine for aircraft are :—

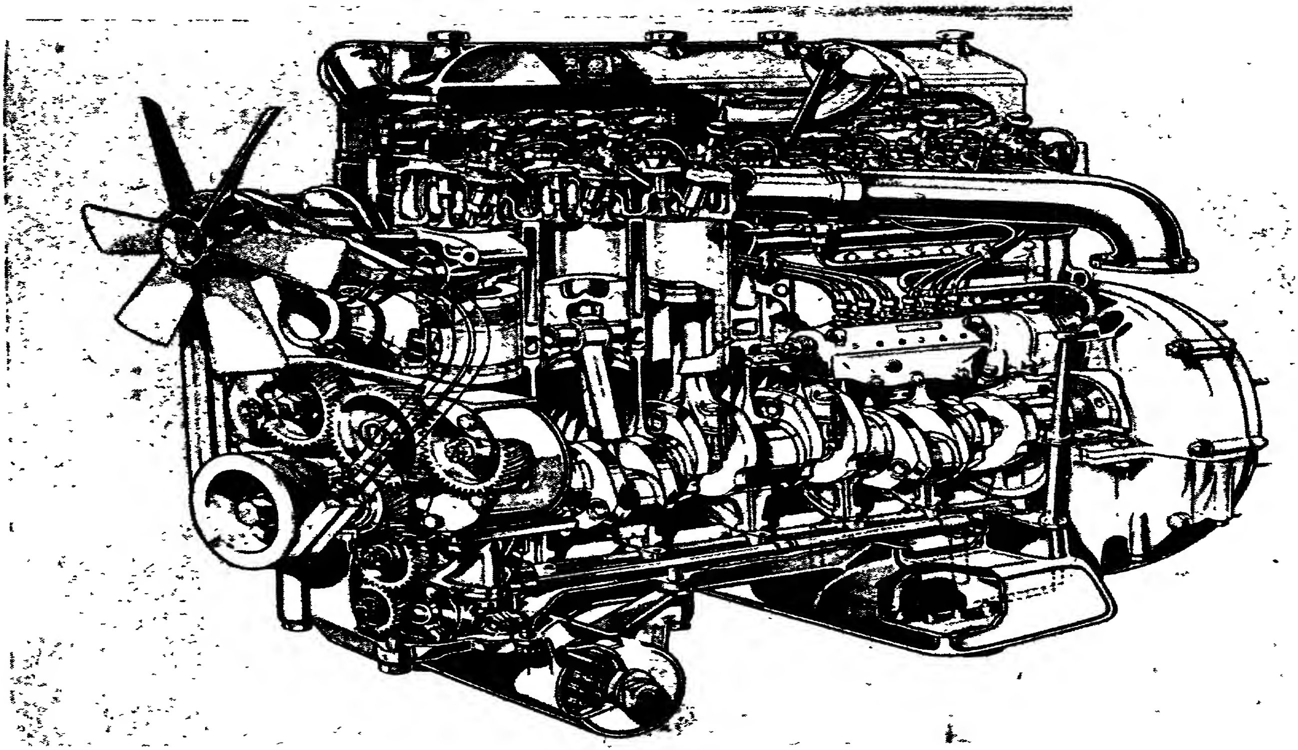


FIG. 10.
9·6 Litre 6 cylinder Direct Injection Oil Engine.
(Associated Equipment Co., Ltd.)

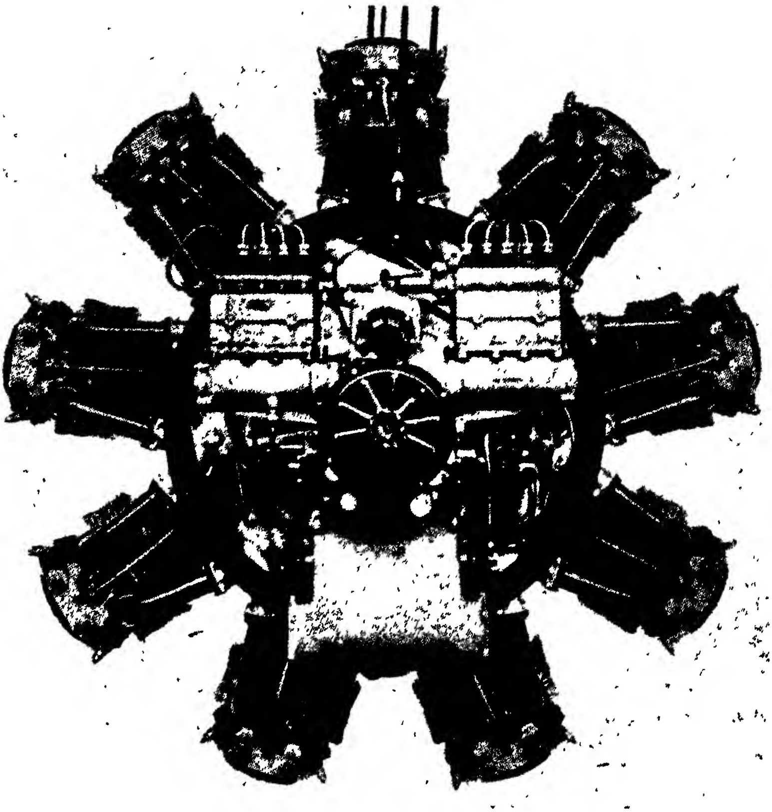


FIG. 11.
"Bristol Phœnix" Compression Ignition Engine for Aircraft. Made by the Bristol Aeroplane Company, Ltd.

(1) The fire risk is reduced.
(2) The fuel cost is less.
(3) The combined weight of engine plus fuel is less for long distance flights.

The weight per B.H.P. of a Diesel Engine is inherently greater than that of a petrol engine of the same general design on account of the greater value of the ratio, maximum pressure ÷ mean pressure. This fact is sufficient to nullify (2) and (3) for short flights. For flights of sufficient length to obtain the condition referred to in (3) the future of the Diesel Engine seemed, a few years ago, to be assured.

With few exceptions aircraft Diesel Engines follow very closely, in their main features, the lines of successful petrol engine practice in this field. Four stroke engines of the 12 cylinder Vee type and 9 cylinder radial arrangement are represented. A multiple fuel pump replaces the magneto and automatic fuel valves replace the plugs. In other respects they are hardly distinguishable from petrol engines externally.

In one important respect the aircraft Diesel Engine was expected to break away from petrol engine traditions. The development of a high duty economical two stroke petrol engine suitable for aircraft still awaits consummation, whereas at least one two stroke cycle Diesel Engine (Junker opposed piston uniflow scavenge engine), was in air service. Others were in course of experimental development. It seemed highly probable that various types of two cycle Diesel Engine sharing the advantages of uniflow scavenge would eventually supersede the four stroke engine in this field.

Two stroke Diesel aircraft engines developing up to about 2000 B.H.P. at revolutions up to 4500 per minute have been built for experimental purposes with weight rates of about 1 lb. per B.H.P.

The development of jet propulsion and the gas turbine has changed the outlook, but it seems possible that the Diesel engine may survive in this field as the first stage of a combination arrangement of a reciprocating engine with an exhaust turbine, in which the power of the Diesel engine is mainly or wholly absorbed in driving its own supercharger.

Supercharged four stroke Diesel engines of "V" type running at speeds of about 2500 R.P.M., originally developed for flight, have been applied to military tanks and light high speed naval craft.

Additional Notes.—Many of the above classes overlap ; for example engines of the horizontal stationary type have been applied to marine propulsion by means of electric drive. By the same means any of the high speed stationary, locomotive or road traction engines may be used for a like purpose. Many high powered stationary Diesel Engines are virtually marine engines adapted to land use.

For aircraft engines the fundamental problem are those of weight reduction and reliability.

For locomotive and road traction uses, considerations of weight and spacc are important but fairly easily met ; the important considerations are reliability and durability. Skilled attention is available in the running shed but not in service. The replacement of worn parts may have an important influence on the total cost of maintenance.

With stationary engines weight matters little and space requirements are not as a rule exacting. A high degree of economy, reliability and durability is expected with comparatively little skilled attention. First cost is important.

For marine service, great reliability and relative ease of adjustment and overhaul take first place. Conditions are favourable to the highest economy in fuel and lubricating oil consumption. Considerations of weight and space have importance but are easily satisfied. First cost is important.

For naval purposes all the above considerations are important except first cost.

Literature.—Judge, A. W., " High Speed Diesel Engines." —Chapman & Hall.

Reed, B., " Diesel Locomotives and Railcars."—Loco. Pub. Co.

Yeates, A. C., " The Development of the Multi-Cylinder Horizontal Oil Engine."—Diesel Users Assoc., Feb., 1931.

Heldt, P. M., " High Speed Diesel Engines."—New York.

Wilkinson, P. H., " Aircraft Diesels."—Pitman.

Pounder, C. C., " Diesel Propelling Engines."—N.E. Coast Inst. of Engineers. and Shipbuilders., Feb., 1945.

Carter, H. D., " The Loop Scavenge Diesel Engine," Inst. Mech. Engs., Mar., 1946.

" Oil Engine Manual."—Temple Press.

" Motor Ship Ref. Book."—Temple Press.

" Diesel Engine Catalog."—Diesel Engines Inc., New York.

" British Diesel Engines."—B.I.C.E.M.A.

CHAPTER III

THERMAL EFFICIENCY

THE earlier part of this chapter (after the first few sections) refers to engines with *Air Blast Injection*. Airless injection engines are considered in the latter part.

The overall thermal efficiency of a heat engine is the ratio of the useful work performed to the mechanical equivalent of the heat supplied during a given period of working.

Problem: What is the thermal efficiency of a Diesel Engine which consumes 0·4 lb. of fuel per brake horse-power hour?

The lower calorific value of different kinds of liquid fuel varies from about 15,300 (Mexican crude) to about 19,300 (Galician crude) British Thermal Units per lb. For calculations and comparison of test results fuel consumptions are usually reduced to their equivalents at a calorific value of 18,000 B.T.U. per lb. One B.T.U. (the amount of heat required to raise the temperature of 1 lb. of water 1° F.) is equivalent to 778 ft. lb. This is Joules' equivalent. Therefore, since 1 H.P. hour = 1,980,000 ft. lb., the required thermal efficiency is equal to :—

$$\frac{1{,}980{,}000}{0{\cdot}4 \times 18{,}000 \times 778} = 0{\cdot}35$$

From this it is seen that roughly one-third of the heat supplied has been converted into useful work. It is within the province of thermodynamics to determine what proportion of the heat loss is theoretically unavoidable, and to what extent the performance of an actual engine approximates to that of an ideal engine working on the same cycle of operations. It is not proposed to give here more than a brief summary of the physical laws relating to the behaviour of air under the influence of pressure and temperature, which forms the basis of thermodynamic investigations.

Pressure, Volume and Temperature of Air.—The relation between these three quantities is expressed by the formula :—

$P.V. = 53{\cdot}2 \times w \times T$—(1) where P=Pressure in lb. per sq. ft. abs.
V=Volume in cubic ft.
w=Weight in lb. of the quantity of air under consideration.
T=Temperature in degrees abs. F.

This relation holds good for any condition of temperature and pressure, and for a specified weight of air, given the values of any two of the quantities represented by capital letters, the third can be calculated.

Example: Find the volume of 1 lb. of air at atmospheric pressure and 60° F. In this case $P=14{\cdot}7$ lb. per sq. in. abs., $T=60+461=521°$ abs. F., and $w=1$.

$$\text{Hence : } V = \frac{53{\cdot}2 \times 521}{14{\cdot}7 \times 144} = 13{\cdot}1 \text{ cub. ft.}$$

Isothermal Expansion and Compression.—If the temperature remains constant during compression or expansion the process is said to be isothermal, and the value of T in equation (1) becomes a constant quantity. Thence for isothermal processes equation (1) becomes : $P.V. = \text{constant}$ . . . .(2)

If 1 lb. of air is under consideration the value of the constant is equal to 53·2 times the absolute temperature.

Work Done during Isothermal Compression.—If P_1 and V_1 represent the pressure and volume before compression, and P_2 and V_2 the same quantities after expansion, then :—

$$\text{Work done} = \text{const.} \log_e \frac{V_1}{V_2} \quad \ldots\ldots\ldots\ldots\ldots\ldots(3)$$

the constant being that of equation (2). It should be borne in mind (though the bare fact only can be stated here) that the internal energy of a gas depends on its temperature only, regardless of the pressure. It therefore follows that all the work done in isothermal compression must pass away as heat through the walls of the containing vessel, and for this reason isothermal processes are not attainable in practice, though they may be approximated to by slow compression in cylinders arranged for rapid conduction of heat.

Specific Heat at Constant Volume.—If 1 lb. of air is heated in a confined space, i.e. at constant volume, 0·169 B.T.U. are required to raise the temperature by 1° F. This then is the specific heat of air at constant volume. For many purposes it is near enough to consider the specific heat as constant, though actually its value increases slightly as the temperature increases. The amount by which the internal energy of 1 lb. of air increases as the temperature rises is therefore :—

$0{\cdot}169\ (T_2 - T_1)$ where $(T_2 - T_1)$ = the increase of temperature.

Specific Heat at Constant Pressure.—In this case, on the other hand, work is done by expansion if heat is being added, and by compression if heat is being discharged ; consequently the specific heat at constant pressure exceeds that at constant volume by the equivalent of the work done. The specific heat at constant pressure is 0·238 B.T.U. per lb. per degree Fahrenheit.

Adiabatic Expansion and Compression.—Expansion or compression unaccompanied by the transfer of heat to or from the air is termed Adiabatic. It should be noted that the air may lose heat by doing external work or gain heat by having work done on it by the application of external force during an adiabatic process ; but this heat comes into existence or passes out of existence within the air itself and does not pass through the walls of the containing vessel.

The following relation holds good between the pressure and the volume during an adiabatic process :—

$P\,V^n$ = constant——(4) where n = ratio of the two specific heats, viz.—1·41.

Owing to the conductivity of the cylinder walls adiabatic compression and expansion are not to be obtained in practice, but it is frequently possible to express the actual relation between pressure and volume by means of equation (4) if suitable values are chosen for " n." A little consideration will shew that for compression with some loss of heat the value of " n " will be less than 1·41, and for expansion with some loss of heat greater than 1·41. An expansion or compression in which the relation between P and V is expressed by equation (4) is known as a Polytropic Process.

Work Done on Polytropic Expansion of Air.—The work done is the integral of the pressure with respect to the volume, and if expressed in ft. lb. is given by :—

$$W = \frac{P_1V_1 - P_2V_2}{n-1} = 53{\cdot}2\,\frac{(T_1 - T_2)}{n-1} \quad \ldots\ldots\ldots\ldots (5)$$

for 1 lb. of air, where the suffixes 1 and 2 have their usual significance in denoting the state before and after expansion; the work done during a corresponding compression between the same pressures is of course the same.

Temperature Change during Polytropic Expansion.—By combining equations (1) and (2) the following is obtained :—

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{1-\frac{1}{n}} = \left(\frac{V_1}{V_2}\right)^{n-1} \quad \ldots\ldots\ldots\ldots (6)$$

With the information supplied by the above six equations it is possible to construct the Indicator Diagram corresponding to an Ideal Diesel Engine, in which all defects of combustion and heat losses to the cylinder walls are supposed to be eliminated.

An Ideal Diesel Engine.—Before proceeding further, it is necessary to define what is to be understood by the term ideal engine for the purposes of this investigation. In the first place, frictional losses and all leakage are eliminated and compression and expansion are supposed to take place adiabatically ($n = 1{\cdot}41$). The specific heats are supposed to be constant and the same whether for air or a mixture of air or exhaust gases. On the other hand, the provision of compressed air for the purpose of fuel injection will be recognised, and consequently the machine will have a mechanical efficiency less than unity.

The compression in the air compressor will be regarded as isothermal and the compressor itself free of all mechanical or other losses.

It will also be supposed that all the exhaust gas, including that contained in the clearance space, is ejected on the exhaust stroke and that every suction stroke fills the cylinder with air at 14·7 lb. per sq. in. abs. and 521° abs. F. The stroke volume is taken for convenience of calculation at 100 cub. ft.

The Ideal Indicator Card.—The indicator card corresponding to this ideal engine is now readily constructed, and is shewn in Fig. 12.

Point A denotes a volume of air equal to 100 cub. ft. plus the clearance space, which has to be calculated.

Line AB represents adiabatic compression from 14·7 lb. per sq. in. to 514·7 lb. per sq. in. abs.

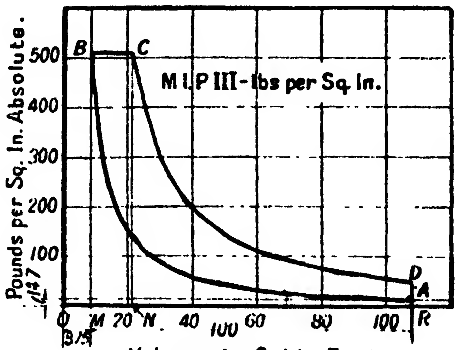


FIG. 12.

Line BC represents increases of volume at constant pressure due to addition of heat by combustion of fuel.

Line CD represents adiabatic expansion of the products of combustion and DA the fall in pressure due to exhaust release. It is not necessary to deal with the exhaust and suction strokes, as these are supposed to take place at atmospheric pressure.

Determination of Clearance Volume.—

From equation (4).

$$\left(\frac{V_A}{V_B}\right)^{1.41} = \frac{P_B}{P_A} = \frac{514{\cdot}7}{14{\cdot}7} = 35$$

$$\therefore \quad \frac{V_A}{V_B} = \left(\frac{1}{35}\right)^{\frac{1}{1\cdot41}} = 12{\cdot}45$$

$$\therefore \quad \frac{100}{V_B} = 11{\cdot}45 \quad \therefore \quad V_B = 8{\cdot}75 \text{ cub. ft.}$$

This determines Points B and A.

Construction of Line AB.—This is effected by calculating the pressure at various points of the stroke in accordance with the following schedule :—

1	2	3	4	5	6	7
Per cent of stroke.	V = cubic feet.	$\frac{V_a}{V}$	Log $\frac{V_a}{V}$	(4) × 1·41.	Antilog (5)	P = (6) × 14·7 lb. per sq. in.
0	108·75	1·00	0·000	0·000	1·00	14·7
20	88·75	1·23	0·099	0·127	1·34	19·7
40	68·75	1·58	0·199	0·280	1·90	27·9
60	48·75	2·23	0·348	0·490	3·09	45·5
80	28·75	3·78	0·579	0·815	6·53	95·5
90	18·75	5·79	0·763	1·075	11·89	174·5
95	13·75	7·90	0·898	1·265	18·41	270·6
100	8·75	12·45	1·095	1·545	35·08	514·7

This simple calculation is given in detail, as it is typical.

Determination of Point C.—The value of V_c depends of course on the amount of heat added. The effect of the oil itself in increasing the weight of the working fluid will be ignored.

The blast air, however, will be taken into consideration and assumed to be equivalent to 8 cub. ft. of free air.

The weight of air concerned has now to be calculated.

(1) Suction air.

$$\text{From equation (1) } w = \frac{P \times V}{53{\cdot}2 \times T} = 144 \times \frac{14{\cdot}7 \times 108{\cdot}75}{53{\cdot}2 \times 521} = 8{\cdot}3 \text{ lb.}$$

(2) Blast air.

$$\text{Weight of blast air} = \frac{8{\cdot}3 \times 8}{108{\cdot}75} = 0{\cdot}61 \text{ lb.}$$

„ „ suction air + blast air = 8·3 + 0·61 = 8·91 lb.

The effect of the heat released by the combustion of the fuel is to increase the temperature of :—

(1) The suction air (8·3 lb.),
(2) The blast air (0·61 lb.),

at a constant pressure of 514·7 lb. per sq. in. abs.

The temperature of the blast air is taken to be 521° F. abs., and that of the adiabatically compressed air is found from equation (6) as follows :—

$$\frac{T_B}{T_A} = \left(\frac{P_B}{P_A}\right)^{1-\frac{1}{1.41}} = \left(\frac{514{\cdot}7}{14{\cdot}7}\right)^{\cdot 29} = 2{\cdot}805$$

Therefore $T_B = 521 \times 2{\cdot}805 = 1460°$ F. abs.

Now let H = heat added in B.T.U.

Then since the pressure remains constant,

$$H = \cdot 238\ [8\cdot 3\ (T_c - 1460) + \cdot 61\ (T_c - 521)\ ]$$
$$= 2\cdot 12\ T_c - 2961$$

from which $$T_c = \frac{H + 2961}{2\cdot 12} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots (7)$$

and from equation (1)

$$V_c = \frac{T_c \times 53\cdot 2 \times 8\cdot 91}{514\cdot 7 \times 144} = \cdot 0064\ T_c \quad \ldots\ldots\ldots\ldots (8)$$

Now suppose that 0·2 lb. of fuel is added, having a calorific value of 18,000 B.T.U. per lb., the resulting temperature will be :—

$$\frac{0\cdot 2 \times 18{,}000 + 2961}{2\cdot 12} = 3100^\circ \text{ F. abs.}$$

and $V_c = \cdot 0064 \times 3100 = 19\cdot 8$ cub. ft.

The expansion line CD can now be constructed in the same way as the compression line, both being adiabatics. The value of the terminal pressure P_d is required for calculating the work done, and is found by means of equation (4) as follows :—

$$\frac{P_d}{P_c} = \left(\frac{V_c}{V_d}\right)^{1\cdot 41} = \left(\frac{19\cdot 8}{108\cdot 75}\right)^{14\cdot 1} = \cdot 091$$

$$\therefore\ P_d = \cdot 091 \times 514\cdot 7 = 46\cdot 8 \text{ lb. per sq. in. abs.}$$

Calculation of Work Done.—Referring to Fig. 12, the work done is clearly equal to the areas BCNM plus CDRN less BARM. Using equation (5) for the two latter, we have :—

Area $BCNM = 144 \times 514\cdot 7 \times (19\cdot 8 - 8\cdot 75) = 819{,}000$ ft. lb.

" $\dfrac{CDRN = 144(514\cdot 7 \times 19\cdot 8 - 46\cdot 8 \times 108\cdot 75)}{\cdot 41} = 1{,}800{,}000$ ft. lb.

By addition 2,619,000 " "

" $\dfrac{BARM = 144(514\cdot 7 \times 8\cdot 75 - 14\cdot 7 \times 108\cdot 75)}{\cdot 41} = 1{,}020{,}000$ " "

Work done by difference = 1,599,000 " "

Since 1 H.P. hour = 1,980,000 ft. lb.

the fuel consumption is :—

$$\frac{0\cdot 2 \times 1{,}980{,}000}{1{,}599{,}000} = \cdot 248 \text{ lb. per I.H.P. hour.}$$

From equation (3) the work done in compressing the blast air is given by

$$8 \times 14 \cdot 7 \times 144 \times \log_e \frac{914 \cdot 7}{14 \cdot 7} \text{ (assuming blast pressure of 900 lb. per sq. in.)}$$

$$= 70{,}000 \text{ ft. lb.}$$

so the nett work done is

$$1{,}599{,}000 - 70{,}000 = 1{,}529{,}000 \text{ ft. lb.}$$

and the consumption of fuel per B.H.P. hour is :—

$$\frac{0 \cdot 2 \times 1{,}980{,}000}{1{,}529{,}000} = \cdot 259 \text{ lb. per B.H.P. hour.}$$

The mechanical efficiency being $\frac{1{,}529{,}000}{1{,}599{,}000} = 95 \cdot 6\%$

The M.I.P. of the diagram is equal to the indicated work divided by the stroke volume.

$$\frac{1{,}599{,}000}{100} = 15{,}990 \text{ lb. per sq. ft.}$$

$$= 111 \text{ lb. per sq. in.}$$

This is about 5 to 20% higher than the M.I.P. usually indicated at full load in an actual commercial engine.

The brake thermal efficiency is given by :—

$$\frac{1{,}980{,}000}{0 \cdot 259 \times 18{,}000 \times 778} = 0 \cdot 548$$

A good figure for the brake thermal efficiency of an actual engine at full load is 0·35; comparing this with the above figure, it is seen that the actual engine attains 64% of the efficiency attributed to the ideal. The indicated thermal efficiency of the ideal engine is :—

$$\frac{1{,}980{,}000}{0 \cdot 247 \times 18{,}000 \times 778} = 0 \cdot 573$$

That of an actual engine at full power is about 0·472, the ratio of actual to ideal being about 82%. From the above it will be seen that so far as the thermal actions within the cylinder are concerned, an actual Diesel Engine in good order leaves comparatively little room for improvement as long as the accepted cycle is adhered to. As a matter of fact, slight deviations from the constant pressure cycle are frequently made, and improved efficiencies obtained thereby. Most high speed Diesels, for example, are arranged to give an indicator card shewing a certain amount of explosive effect, i.e. combustion at constant volume, causing the pressure at the dead centre to rise to a figure which may be anything up to about 100 lb. above the compression pressure. This is found to have a beneficial effect on the fuel consumption, which is readily explained on theoretical grounds; but it is obvious that considerations of strength must limit the extent to which this principle is used.

Fuel Consumption at Various Loads.—If the foregoing calculations for the fuel consumption of the ideal engine are

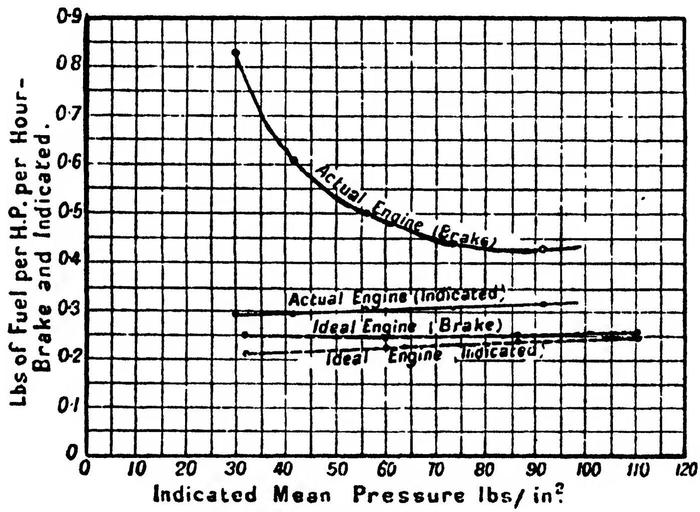


FIG. 13.

repeated for various values of the quantity of fuel admitted results will be obtained which are shewn graphically in Fig. 13, together with test results of an actual engine. The fuel consumptions per I.H.P. and B.H.P. hour are plotted on a basis of M.I.P.; the actual engine to which the test results refer was of the four stroke type developing 100 B.H.P. per cylinder at full load. There are two facts to be noticed :—

(1) The fuel consumption per I.H.P. hour increases as the M.I.P. increases.

(2) The fuel consumption per B.H.P. hour attains a minimum value at about 90 lb. M.I.P. in the case of the actual engine and about 60 lb. in the case of the ideal. The reason for (1) will be evident on comparing theoretical diagrams for various values of the M.I.P., and the case is quite comparable to that of a steam engine working with an early cut-off.

The point of maximum brake efficiency depends upon two conflicting influences, viz. the indicated efficiency which decreases and the mechanical efficiency which increases as the M.I.P. is augmented.

It is usual, when considering mechanical efficiency, to treat the work done in driving the air compressor as a mechanical loss, so that :--

$$\text{Mechanical efficiency} = \frac{\text{B.H.P.}}{\text{I.H.P. of main cylinders}}$$

Variation of Mechanical Efficiency with Load.—Examination of a large number of Diesel Engine test results reveals the fact that the difference between the I.H.P. and the B.H.P. remains nearly constant as the load is varied. This fact enables one to calculate the mechanical efficiency at any load if the full load efficiency is known.

Example : What is the mechanical efficiency at three-quarter, half and quarter load if that at full load is 72% ? Let the full load I.H.P. be represented by 100, then the following tabulated figures hold good :—

	B.H.P.	I.H.P.	Const. Diff.	Mech. Efficy.
Full load . . .	72	100	28	·72
Three-quarter load .	54	82	28	·66
Half load . . .	36	64	28	·56
Quarter load . .	18	46	28	·39

This method yields sufficiently accurate results for many estimating purposes.

Mechanical Losses.—The work corresponding to the difference between the I.H.P. and the B.H.P. is approximately accounted for in the two following tables, which apply to medium-sized engines of good design, and of the four stroke and two stroke air blast injection types respectively :—

FOUR STROKE ENGINE—Full load Mech. Efficy., 75%

	Per cent.
Brake-work	75·0
Work done on suction and exhaust strokes	3·0
Indicated compressor work	5·8
Compressor friction	1·2
Engine friction, opening valves, etc.	15·0
Work indicated in main cylinders	100·0

TWO STROKE ENGINE—Full load Mech. Efficy., 73·5%

	Per cent.
Brake-work	73·5
Indicated compressor work	6·2
Compressor friction	1·4
Indicated scavenger work	3·3
Scavenger friction	1·6
Engine friction, opening valves, etc.	14·0
Work indicated in main cylinders	100·0

Improvements in bearings and guides on the principle of the well-known Michell bearing or the use of roller bearings for the main journals and big ends suggest possibilities for reducing engine friction which will possibly materialise in the future. The adoption of some form of limit piston ring to prevent excessive pressure on the liners would also help matters in the same direction, besides increasing the life of the liners.

Influence of Size on Mechanical Efficiency and Fuel Consumption.—Piston speed, within the range of present practice, appears to have very little influence on either mechanical efficiency or fuel consumption. This remark does not apply to the abnormally low speeds obtaining, for example, with a marine engine turning at reduced speed.

The cylinder bore is the principal factor in economy, always assuming a reasonable ratio of bore to stroke and good design generally.

The following table is a rough guide to the mechanical efficiency and fuel consumptions to be expected from cylinders of various sizes working at full load with air injection.

Four Stroke				Two Stroke			
Cylinder Diam., in.	Mech. Effcy. %	Fuel per B.H.P. hr. lb.	Fuel per I.H.P. hr. lb.	Cylinder Diam., in.	Mech. Effcy. %	Fuel per B.H.P. hr. lb.	Fuel per I.H.P. hr. lb.
10	·72	·43	·310	10	·70	·45	·314
15	·73	·41	·300	15	·71	·42	·300
20	·75	·40	·300	20	·73	·41	·300
25	·76	·40	·305	25	·74	·41	·304
30	·76	·40	·305	30	·74	·41	·304

The fuel consumption per B.H.P. at loads other than full load is readily found by first calculating the probable mechanical efficiency, as described in the previous article, and then allowing for a fall in the consumption per I.H.P. proportional to that shown on Fig. 13 for a typical engine for the same M.I.P.

Heat Balance Sheet.—An elaborate trial of a Diesel Engine includes the measurement of the quantity of cooling water used, the inlet and outlet temperatures of the water and the temperature of the exhaust gases. Apart from very slight losses, such as radiation, etc., these data usually enable the heat supplies by the fuel to be accounted for.

A typical heat balance is given below :—

	Per cent.
Accounted for by B.H.P.	34
Rejected to cooling water	28
Rejected to exhaust and otherwise dissipated .	38
Total heat supplied . .	100

A striking feature of this balance is the large amount of heat appearing on the cooling water account, which at first sight would appear to indicate very poor utilisation of heat within the cylinder. It has been shewn that so far from this being the case, an actual engine in good order indicates about 80% of the work attributable to an ideal engine.

The explanation of this lies in the fact that a large proportion of the heat received by the cooling water is given out by the exhaust gases after combustion is complete, particularly on their passage through the cylinder cover in the case of a four stroke engine and through the ports in the case of a two cycle. Most of the friction work done by the piston and all the compressor work appear on the cooling water account.

Efficiency of Combustion.—In all actual oil engines there is a considerable amount of "after burning," i.e. gradual burning during the expansion stroke. In a well-tuned Diesel Engine this effect is not sufficiently pronounced to cause smoke even at considerable overload, 120 lb. per sq. in. M.I.P. for example. Exaggerated "after burning" is to be avoided as, in addition to increasing the fuel consumption, it increases the mean temperature of the cycle and of the exhaust stroke particularly, and gives rise to accentuation of all the troubles which arise from the effects of high temperature. The most prominent of these troubles are enumerated below :—

(1) Cracking of piston crowns and cylinder covers.
(2) Pitting of exhaust valves.
(3) Increased difficulty of lubricating the cylinders, resulting in—
  (a) Increased liner wear.
  (b) Sticking of piston rings.
  (c) Liability to seizure of piston.
(4) Increase of temperature of gudgeon pin bearing.

The above formidable list is probably not exhaustive, but is sufficient to shew the desirability of securing the best possible conditions, apart altogether from the question of economy in fuel and lubricating oil consumption. The attainment of good combustion, assuming a good volumetric efficiency of cylinder and good compression, depends more than anything upon small points in connection with the fuel valve, which are easily adjusted on the test bed, provided the design of the fuel valve is satisfactory.

Entropy Diagrams.—Entropy is a convenient mathematical concept which it is difficult, and perhaps impossible, to define in non-mathematical terms. It is sometimes described as that function of the state of the working fluid which remains constant during an adiabatic process. Entropy increases when heat is taken in by the working fluid and decreases when it is rejected. If heat is supplied to the working fluid at constant temperature, then the increase of entropy is equal to the amount of heat so supplied, divided by the absolute temperature. If the temperature is variable during the process of heat absorption, then the increase of entropy is determined by the integration of the equation :—

$$d\phi=\frac{dH}{T}$$ where ϕ=Entropy.
H=Heat taken in or given out.
T=Absolute temperature.

The zero of entropy may be located at any convenient level of temperature except absolute zero. It is convenient for our purposes to consider the entropy to be zero when

$P=14{\cdot}7$ lb. per sq. in. abs.
$T=521°$ F.

The value of a diagram connecting T and ϕ during the working cycle of an internal combustion engine depends upon the following properties :—

(1) Increasing and diminishing values of ϕ denote heat supplied and heat rejected respectively.
(2) For a complete cycle the diagram is a closed figure, the area of which is proportional to the quantity of heat which has been converted into work during the cycle.

The area of the diagram is given by :—

$$\int T\,d\phi=\int\frac{TdH}{T}=H_1-H_2$$ the difference between the heat supplied and that rejected.

Construction of an Entropy Chart for Air (see Fig. 14).—The axis of T is drawn vertically, and includes temperatures from 0 to 3000° F. abs. The axis of ϕ is horizontal, and is graduated from 0 to 0·30. It will be clear that the axis of T is an adiabatic line of zero entropy and that any selected temperature on this line corresponds to a definite pressure, which can be calculated by means of equation (6). Actually it is more convenient to tabulate a series of pressures from 0 to, say, 700 lb. per sq. in. abs., and tabulate the corresponding temperatures. Points on the axis of T found in this manner are the starting points of constant pressure lines.

Constant Pressure Line. $P=14{\cdot}7$ lb. per sq. in.—A constant pressure line is a curve connecting T and ϕ when P remains constant. It is usual to consider 1 lb. of air, and if the specific heat at constant pressure is assumed to be 0·25

Then $dH=0{\cdot}25\,dT$

$$\text{And } d\phi=\frac{dH}{T}={\cdot}25\frac{dT}{T}$$

Therefore $\phi={\cdot}25\log_e\frac{T_2}{T_1}$, ($T_1$ being 521° F. abs.)

Selecting various increasing values of T_2 corresponding values of ϕ are calculated and plotted on the chart, a fair curve passing through the points being the required constant pressure line. One such line having been constructed, lines corresponding to other pressures are readily drawn, since their ordinates

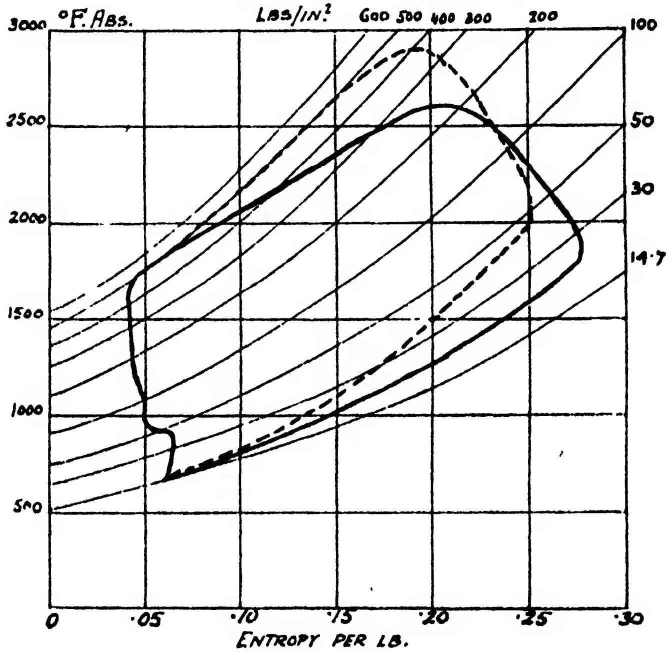


Fig. 14.

are proportional to the temperatures indicated by their starting points.

Constant Volume Lines.—These can be constructed in a similar manner, using the specific heat at constant volume instead of that of constant pressure. On Fig. 14 only one constant volume line is shewn, viz. that corresponding to

$$P = 14{\cdot}7 \text{ lb. per sq. in.}$$
$$T = 673^\circ \text{ F. abs.}$$

as this is required to complete the diagram by representing the rejection of heat at constant volume at the end of the stroke. This process is of course a scientific fiction, as the pressure in an actual cylinder is reduced to approximately atmospheric pressure by the discharge of exhaust gases and not by cooling the latter. The reason for using the value 0·25 for the specific heat must now be explained. Although the specific heat of pure dry air is 0·238, that of the gases present in the cylinder of a Diesel Engine is a variable quantity for the following reasons :—

(1) The composition of the working fluid is altered by the addition of the fuel.
(2) The specific heat of the exhaust gases increases slightly with increase of temperature.

If these variations were rigidly taken into account the construction of the entropy diagram would be a very laborious business, and the work is greatly curtailed by adopting certain approximations which will be described. In the first place the specific heat is assumed to be constant and equal to the calculated specific heat of the exhaust gases at 60° F. The variation in the weight of the working fluid is dealt with by first treating the diagrams as though the weight were constant and then correcting the diagram (entropy) by increasing the entropy and decreasing the absolute temperature of points on the expansion line in the same proportion by which the fuel and blast air increase the weight of the charge.

Use of the Entropy Chart.—A method of constructing the entropy diagram, corresponding to an indicator card, will now be described. The clearance volume of the engine must first be ascertained and the card accurately calibrated. Points on the indicator diagram are then marked, corresponding to every 15 or 30° or other convenient division of revolution of the crank past the top dead centre. Each of the 12 or more points so marked is given a reference number and the absolute pressure in lb. per sq. in., and also the volume in any arbitrary units (hundredths of an inch on the diagram, for instance) corresponding to each point is read off and tabulated. The apparent temperature corresponding to each point is then calculated by means of equation (1) on the assumption that the initial temperature just before compression is, say, 673° F. abs.

Points on the entropy diagram corresponding to the selected points on the indicator diagram are now found by following the appropriate constant pressure lines until the calculated temperatures are reached, thus obtaining the apparent entropy diagram, which requires to be corrected in accordance with the preceding article.

The entropy diagram shewn in Fig. 14 has been constructed in the manner described, and the data are given below :—

M.I.P. shewn by indicator diagram, 82 lb. per sq. in. (see Fig. 15).

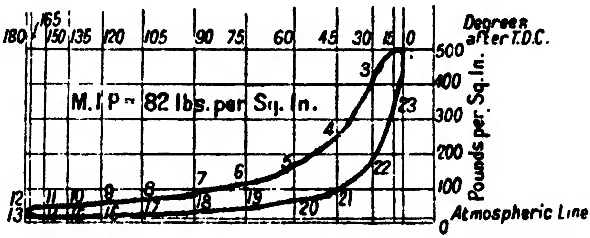


Fig. 15.

Initial temperature of suction air before compression assumed to be 673° F. abs.

Absolute temperature (apparent) calculated from PV=kT where P=Pressure in lb. per sq. in. abs. (scaled off diagram)

V=Volume measured in linear inches off diagram

T=Temperature in degrees F. abs.

Initial conditions are P=14·7, V=6·38, T=373.

Whence k=·1395.

Figures are given in the table below :—

Ref. No.	Degree after firing centre.	V in in.	P in lb. per sq. in.	Apparent Temperature.
1	0	0·48	490	1690
2	15	0·63	498	2260
3	30	0·95	421	2880
4	45	1·53	251	2760
5	60	2·17	163	2540
6	75	2·94	114	2400
7	90	3·75	84	2260
8	105	4·56	65	2130
13	180	6·38	14·7	673
19	265	2·94	44	930
20	300	2·17	61	950
21	315	1·53	99	1090
22	330	0·95	184	1250
23	345	0·63	331	1500

The apparent entropy diagram was plotted from the above values of P and T, and the corrected diagram, constructed in accordance with the preceding article, is shewn in Fig. 14.

Area of corrected entropy diagram 9·90 sq. in.

Temperature scale . . .	1 in. = 500°.
Entropy scale	1 in. = ·05°.

Therefore work done per lb. of suction air is given by :—

$$9{\cdot}90 \times 500 \times 0{\cdot}05 = 248 \text{ B.T.U.}$$

1 lb. of suction air @ 673° F. abs. occupies 16·9 cub. ft. Since clearance volume = 8% of stroke volume, therefore corresponding stroke volume = 16·9 ÷ 1·08 = 15·65 cub. ft. Therefore work done by suction air, according to the indicator card, @ 82 lb. per sq. in. M.I.P. is given by :—

$$\frac{82 \times 144 \times 15{\cdot}65}{778} = 238 \text{ B.T.U.}$$

This figure is about 4% less than that shewn by the entropy chart, and suggests that perhaps the assumed temperature of the suction air before compression (viz. 673° F. abs.) is too high, which is very probable.

Note.—The low pressures at the beginning of the compression stroke and the end of the expansion stroke have not been used in the construction of the entropy chart for the following reasons :—

(1) Low pressures are very difficult to scale off the indicator card with any degree of accuracy.

(2) The low pressures indicated on the card are invariably erroneous, unless the greatest care has been exercised in taking the card, and that with a suitable indicator in perfect order. Many of the reputable makes of indicator, though quite suitable for steam engine work, give very inaccurate results when applied to internal combustion engines.

The following points should be noticed :—

(1) The compression line deviates from the true adiabatic in a manner which indicates that heat has been lost to the cylinder walls.

(2) The expansion does not become adiabatic until well after half stroke, shewing that there is a certain amount of "after burning."

Area of Entropy Diagram.—If the diagram has been carefully done the area of the diagram in heat units should correspond fairly closely to the work shewn on the indicator diagram. Deviations from equality may be due to :—

(1) Variation in the specific heat at high temperatures.
(2) Incorrect assumption of the initial temperature of the induced air.

Owing to the approximations referred to above there is generally a discrepancy of a small percentage between the amounts of work shewn by the indicator card and the entropy diagram. Also the total area under the upper boundary of the entropy diagram should correspond to the total heat supplied per lb. of suction air less the heat discarded to the water-jacket on the expansion stroke. Investigation of the sort described seems to indicate that not more than about 10% to 15% of the total heat supplied by the fuel is lost in this way at 100 lb. M.I.P.

Specific Heat of Exhaust Gases.—The specific heat of a mixture of gases is the sum of the products of the specific heats of the constituents into the proportion by weight in which the constituents are present.

The specific heats (at constant pressure) of the constituent gases present in the exhaust of an oil engine are given below :—

Water vapour . .	0·480	(Varies considerably with temperature)
Nitrogen . . .	0·247	
Oxygen . . .	0·217	
CO_2	0·210	(Varies considerably with temperature)
CO	0·240	

The average composition of air and the specific heat derived from that of its constituents are given below :—

N_2 . .	75·7%	× 0·247	= ·187
O_2 . .	22·7	× 0·217	= ·049
H_2O . .	1·5	× 0·480	= ·007
Total .	100		·243

This is about 3% higher than the accepted value for pure dry air.

The approximate composition of exhaust gases assuming complete combustion is readily calculated as follows :—

Data M.I.P.—100 lb. per sq. in.

Fuel per I.H.P. hour—0·31 lb.

Composition of fuel (assumed) $\begin{cases}\text{Carbon, 86\%}\\ \text{Hydrogen, 13\%}\\ \text{Oxygen, 1\%}\end{cases}$ by weight.

One H.P. hour=1,980,000 ft. lb.

$\therefore$ Volume swept by piston per I.H.P. hour$=\dfrac{1,980,000}{100\times 144}$

=137·5 cub. ft.

Clearance volume, say, 8% = 11·0 „ „

Total volume of suction air =148·5 „ „ (assuming perfect scavenge of clearance space).

$$\text{Weight of suction air}=\frac{148\cdot5\times14\cdot7\times144}{53\cdot2\times673}=8\cdot78\text{ lb. @ }212^\circ\text{ F.}$$

Free volume of blast air @ 8% of stroke volume @ 60° F.
=·08×137·5=11 cub. ft.

$$\text{Weight of blast air}=\frac{11\times14\cdot7\times144}{53\cdot2\times521}=0\cdot83\text{ lb.}$$

$\therefore$ Total air=8·78+0·83	=9·61 lb. =	97%
Weight of fuel .	=0·31 „ =	3%
Total mixture .	9·92 „	100%

Composition of mixture before combustion is given by :—

Air	N_2 . .	·76×97	=73·7%
	O_2 (22·7×97)+(·01×3)		=22·0%
	H_2O . .	·015×97	= 1·5%
Fuel	C . .	·86×3	= 2·5%
	H_2 . .	·13×3	= 0·4%
			100·1%

Combustion of carbon and hydrogen takes place in the proportions given by :—

$$C+O_2=CO_2$$
12+32=44 by weight

And

$$H_2+O=H_2O$$
2+16=18 by weight

The following table shews the composition after combustion and the specific heat (constant pressure) of the mixture :—

	Per cent.		Specific heat.		Product.
N_2	73·7	×	·247	=	·1820
CO_2 $2{\cdot}5 \times 44 \div 12$ . .	9·2	×	·217	=	·0191
O_2 $22-(32 \times 2{\cdot}5 \div 12)$ $-(16 \times 0{\cdot}4 \div 2)$.	12·1	×	·210	=	·0254
H_2O $1{\cdot}5+(18 \times 0{\cdot}4 \div 2)$.	5·1	×	·480	=	·0245
	100·1				

Specific heat of exhaust gases = ·2510

Repeating the above calculation for different values of the M.I.P., the following figures are obtained :—

M.I.P. lb. per sq. in.	0	30	45	75	105	130	155
Specific heat . .	·245	·247	·248	·250	·252	·254	·256

Airless Injection.—The use of mechanical means in place of air blast injection increases the mechanical efficiency by nearly 10%, and the change over is usually accompanied by a reduction of fuel consumption per B.H.P. of about this amount ; this indicates that the fuel consumption per I.H.P. has remained unchanged. This result is partly to be attributed to a higher

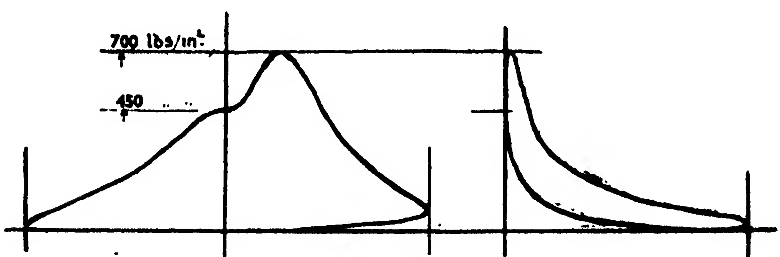


FIG. 16.

maximum pressure with airless injection. For the same maximum pressure in each case the fuel consumption per I.H.P. is less with air blast injection. Satisfactory combustion with airless injection requires earlier injection and a decided rise in pressure above compression pressure during the combustion period. A typical airless injection indicator diagram is shewn in Fig. 16.

Large or medium size relatively slow running engines usually shew a period of combustion at constant volume followed by a period of combustion at approximately constant pressure (dual cycle). Very high speed engines give indicator diagrams approximating to the constant volume cycle.

The efficiencies of these cycles have been calculated for various conditions by W. J. Walker (I. Mech. E., Dec., 1920). By means of an entropy diagram it is easy to shew that the Diesel cycle is theoretically slightly more efficient than a constant volume cycle or a dual cycle having the same *maximum* pressure, provided the expansion stroke is the same length as the compression stroke in each case. In practice, however, the dual cycle in which part of the combustion takes place at constant volume appears to give a better fuel consumption than the Diesel cycle with Airless Injection Engines. This is probably due to the fact that the constant volume combustion is brought about by earlier injection of the fuel resulting in better distribution of the fuel in the combustion space and less after burning.

Effect of Variable Specific Heat.—The specific heats of air are not constant (as heretofore assumed in this chapter) but increase with the temperature. The effect of this increase is to lower the efficiency theoretically obtainable under any proposed conditions. Figures shewing the variation of specific heats are given under :—

Temperature ° C. .	0	500	1000	1500	2000	2500
,, ° F. .	32	932	1832	2732	3632	4532
C_v=specific heat const. volume .	0·171	0·172	0·178	0·204	·236	·254
C_p=const. pressure .	0·240	0·241	0·247	0·273	0·305	0·323
$C_p \div C_v$ . . .	1·41	1·41	1·39	1·34	1·29	1·27

The effects of these variations on the efficiency of various internal combustion engine cycles have been worked out by the Heat Engine Trials Committee of the Inst. C. E. (see Report, 1927). The efficiencies are given in terms of the Compression ratio (r) and the Heat supply per standard cubic foot, mean lb. calories (I). One standard cubic ft. refers to a pressure of 14·7 lb. per sq. inch and a temperature of 32° F. (0° C.) and corresponds to 0·081 lb. of air. Now one lb. of average Diesel fuel requires for theoretical complete combustion 14·2 lb. of air so that the quantity I is related to the excess air percentage and percentage of CO_2 after complete combustion, as tabulated under :—

Excess air % . .	0	50	100	150	200
CO_2 % . . .	15·6	10·2	7·5	6·0	5·0
O_2 % . . .	0	7·25	10·7	12·8	14·2
A=air per lb. of fuel in lb. . .	14·2	21·3	28·4	35·5	42·6
I=lb. calories per standard ft.^{3} of air.	57·0	38·1	28·5	22·8	19·0

$I = 10{,}000 \times 0{\cdot}081 \div A$ where 10,000=assumed calorific value of oil in lb. calories per lb.=18,000 B.T.U. per lb.

Efficiencies of Various Cycles.

In the Report referred to above efficiency curves are given for the following cycles, viz.:—

(1) Constant Volume cycle.
(2) Constant Pressure cycle with compression carried down to atmospheric pressure by making expansion stroke longer than compression stroke.
(3) Diesel cycle.
(4) Atkinson cycle, i.e. combustion at constant volume and expansion carried down to atmospheric pressure.

A selection of the results has been read off these curves and retabulated under in such a way as to give a comparison of the various cycles.

Theoretical Efficiencies, I=30.

Compression Ratio	10	12	14	16	18	20
Constant Volume Cycle . .	57·2	60·0	62·0	63·7	65·1	66·4%
Diesel Cycle .	46·0	50·0	53·0	56·0	58·0	60·0 ,,
Atkinson Cycle .	65·8	67·3	69·5	70·5	71·5	72·1 ,,
Constant Pressure Cycle . .	58·5	61·2	63·5	65·2	66·8	68·0 ,,

Theoretical Efficiencies, I=10.

Compression Ratio	10	12	14	16	18	20
Constant Volume Cycle . .	59·0	62·0	64·0	65·5	67·2	68·4%
Diesel Cycle .	57·7	58·0	60·5	62·5	64·3	66·0 ,,
Atkinson Cycle .	63·5	65·2	66·8	68·8	69·8	70·8 ,,
Constant Pressure Cycle . .	59·3	62·2	64·3	66·2	68·0	69·0 ,,

Theoretical Efficiencies, I=50.

Compression Ratio	10	12	14	16	18	20
Constant Volume Cycle . .	54·6	57·3	59·5	61·3	63·0	64·0%
Diesel Cycle .	39·0	43·5	47·5	50·0	52·5	54·5 ,,
Atkinson Cycle .	66·5	68·8	70·0	70·8	71·5	72·1 ,,
Constant Pressure Cycle . .	57·2	60·0	62·2	64·1	65·5	66·9 ,,

The Atkinson cycle and the Constant Pressure cycle, involving as they do expansion strokes longer than the compression strokes, naturally give higher efficiencies than the other two cycles considered. A petrol engine working on a cycle approximately to the Atkinson cycle has been constructed by the Citröen Co. and the published results indicate that the theoretical improvement in fuel consumption as compared with an engine working on the constant volume cycle is realised in practice. The increased size and cost of an engine working with an expansion stroke greatly in excess of the compression stroke are serious disadvantages and there is a danger of the gain in indicated efficiency being counterbalanced by reduced mechanical efficiency. Nevertheless the possibilities of these two cycles should be seriously considered in the future development of heavy oil engines.

Another feature disclosed by the above tables is the superior efficiency of the constant volume cycle as compared with the Diesel cycle, for a given compression ratio; the disadvantage of the constant volume cycle is the high maximum pressure. For a given maximum pressure the Diesel cycle is more efficient than the constant volume cycle. *On account of the interval of time required for inflammation and combustion, the constant volume cycle seems hardly attainable with heavy oil but the present tendency with high speed heavy oil engines is towards the constant volume type of indicator card.* With large slow revolution engines the tendency is to limit the maximum pressures and to obtain indicator cards which represent a compromise between the constant volume and the Diesel cycles.

Fuel Consumption of Airless Injection Engines.—The following table shows the fuel consumptions per B.H.P. hour (reduced to their equivalents with fuel having a nett calorific value of 18,000 B.T.U. per lb.) to be expected with various maximum pressures. Good arrangements for combustion, adequate valve and port areas, and suitable design, in relation to revolutions, to prevent excessive frictional losses are assumed.

Max. cylinder pressure lb./in.^{2}	Cylinder diameters. 5″	10″	15″	20″ up.
500	0·46	0·43	0·41	0·40
600	0·44	0·41	0·395	0·38
700	0·42	0·395	0·38	0·37
800	0·41	0·38	0·365	0·36
900	0·40	0·37	0·355	0·35
1000	0·39	0·365	0·35	0·34

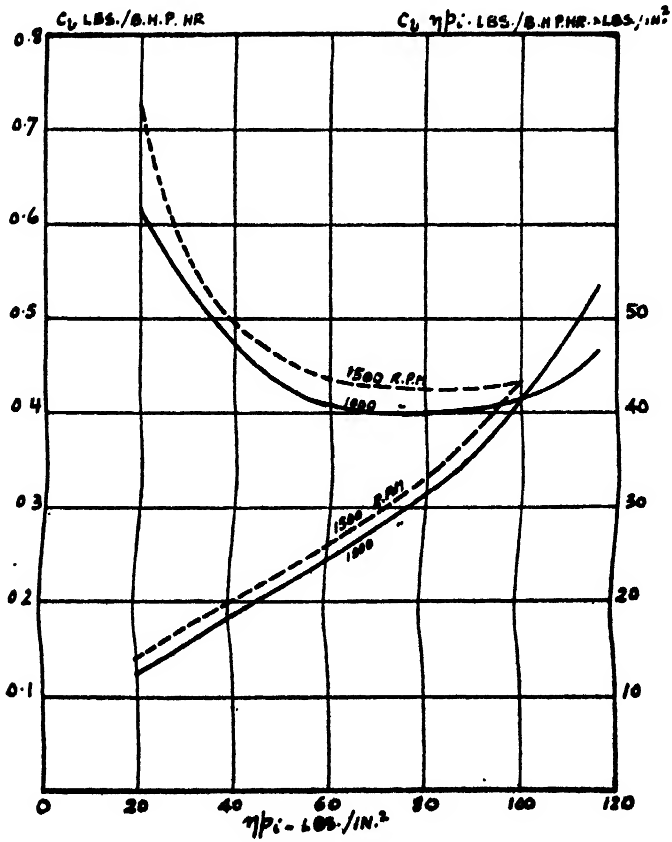


FIG. 17.

Lower values have been recorded but the above table is fairly representative of test bed performances. There is little if any difference in fuel consumption as between the best examples of four stroke and two stroke engines.

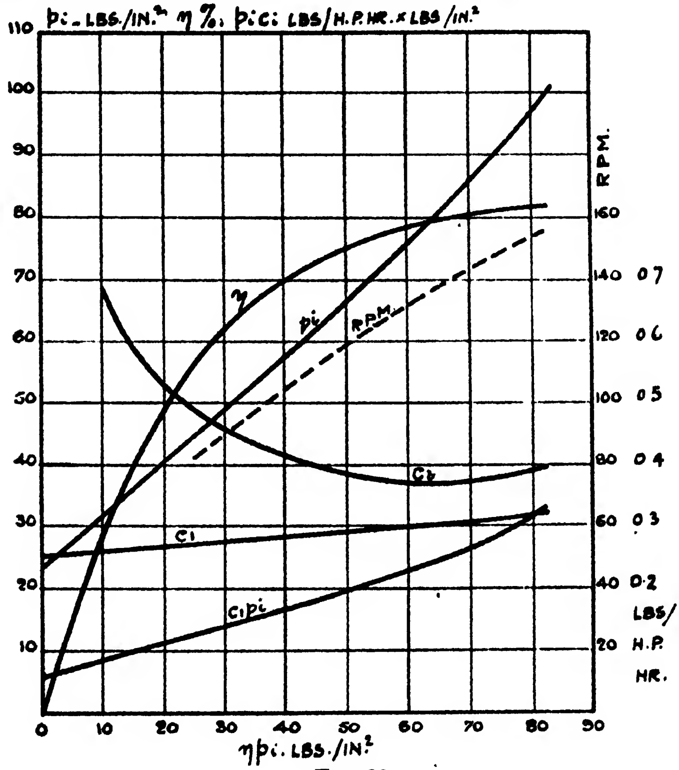


Fig. 18.

Performance Diagram.—It is very convenient for comparative purposes to plot engine performance in a way which eliminates all reference to the dimensions of the engine. Such diagrams are shewn in Figs. 17 and 18.

On a basis of brake mean pressure (ηp_i) the following quantities are plotted, viz. :—

p_i=indicated mean pressure, lb./in.^{2}

η=mechanical efficiency.

c_b=fuel consumption ; lb./B.H.P. hour.

$c_i=\eta c_b$=consumption ; lb./I.H.P. hour.

$c_ip_i=c_b\eta p_i \propto$ total fuel per hour.

The following may also be added, viz. :—

$(1-\eta)pi$ = lost mean pressure,

and exhaust temperature.

Exhaust Temperatures.—On all but the smaller engines it is usual to fit a thermometer at the exhaust outlet for each cylinder. These instruments give useful indications as to satisfactory combustion and equality of distribution between the cylinders. The exhaust temperature depends fundamentally on the mean pressure but it is influenced by so many other factors that it is difficult to formulate rules for its normal value to cover all cases. The value registered by the thermometer is some sort of average related to a variable rate of flow and may be considerably less than the value recorded by a thermometer placed some distance along the common exhaust pipe. Electrical pyrometers usually record lower values than mercury thermometers. Some of the other factors which influence the readings (mean pressure being supposed constant) are as follows :—

(1) Revolutions or piston speed or cylinder speed.
(2) Absolute size.
(3) Back pressure in exhaust.
(4) Throttling of air suction.
(5) Degree of cooling surfaces near the thermometer.
(6) Supercharging, and scavenging effects.
(7) Excess air coefficient of two stroke engines.

Fig. 19 shews three curves of exhaust temperatures plotted against mean indicated pressure for a four stroke engine with normal induction, the same supercharged, and a two stroke engine with pump or blower scavenging.

A sharp upward inflection of the curve indicates the approaching limit of good combustion.

Conditions for High Efficiency.—Some of the chief conditions which are conducive to low fuel consumption of airless injection engines are enumerated under.

(1) Compact combustion chamber. This condition is avoided in certain designs involving precombustion chambers, air cells and the like, in which violent turbulence compensates for subdivision of the combustion space. Without such turbulence, a low degree of compactness is fatal to efficiency.

(2) Correct timing of the fuel injection. This should be easily adjustable on the test bed.

(3) Correct rate of fuel injection. If the rate of injection is too slow combustion proceeds too late in the stroke. If too fast the maximum pressure is too high. It is important that the rate of injection should be sustained and not trail off towards the end. If the fuel pump plunger diameter and the rate of rise of the fuel cam are correct, the rate of injection may yet be incorrect by reason of excessive plunger leakage or sluggish action of the fuel valve.

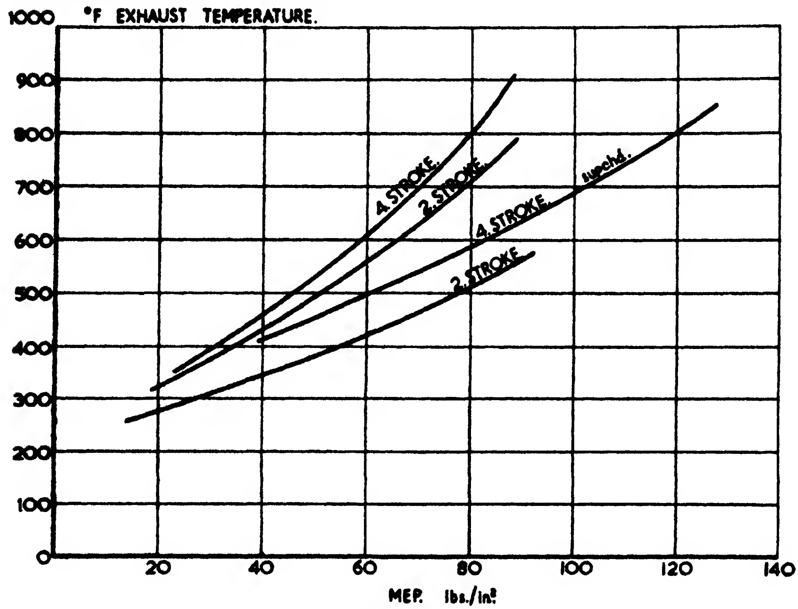


Fig. 19.

(4) Even distribution of fuel in the combustion space. This is sometimes achieved by gas turbulence as referred to above. Where this condition is absent, correct direction of nozzle orifices and correct size of orifices to give sufficient but not too much penetration are important.

(5) Sufficient compression to produce the charge temperature required for the satisfactory initiation of combustion.

(6) A sufficient replacement of the spent charge with pure air, in relation to the mean pressure desired. A poorly scavenged two cycle engine will develop only a low mean pressure with economy. A thoroughly scavenged two cycle engine or a four stroke engine with good volumetric efficiency

will develop a substantially higher mean pressure with better efficiency. By supercharging the mean pressure can be still further increased, without loss and sometimes even with gain in efficiency.

(7) A high mechanical efficiency (see Chapter IV).

Literature.—Dalby, W. E., " Power and the Internal Combustion Engine."—Arnold.

Pye, D. R., " The Internal Combustion Engine."—Oxford Univ. Press.

Ricardo, H. R., " Engines of High Output."—Macdonald and Evans.

Inst. Civil Engrs., " Report on Heat Engine Trials." 1927.—Clowes.

CHAPTER IV

MECHANICAL EFFICIENCY

Introduction.—According to one definition, the mechanical efficiency η of an internal combustion engine is the ratio of the so-called brake mean effective pressure ηp to the mean pressure p as measured from the positive loop of the indicator diagram. The difference $p(1-\eta)$ may be termed the "lost mean pressure." It includes, according to the above definition, in addition to frictional losses, all deductions due to the indicated work of air compressors, scavenge pumps, fuel pumps, lubricating oil pumps, water pumps and also the mean pressure due to the negative loop of the indicator diagram, occasioned by the pumping losses in the case of a four stroke engine. It is worth remark that the negative loop of an indicator diagram shewing compression and re-expansion in a cylinder not firing is not included in the lost mean pressure as above defined. This loss has the status of a thermodynamic loss and not a mechanical loss.

Indicator cards are not infallible records of pressure variation in the cylinder and a more general definition agreeing in principle with the above, defines mechanical efficiency as the ratio of brake mean pressure to the nett mean pressure in the cylinder, during the compression and expansion strokes, averaged on a piston displacement basis, leaving it an open question as to how this mean pressure is to be measured in practice.

A third definition defines mechanical efficiency as the ratio of the brake mean pressure to the brake mean pressure plus the mean pressure due to the sources of loss enumerated above. The great difficulty of applying this definition arises in measuring the friction losses under working conditions. Three methods have been used, viz. :—

(a) Measuring the power required to run the engine (not firing) with an electric motor.

(b) Measuring the retardation of a flywheel of known moment of inertia when fuel is cut off.

(c) Measuring the drop of B.H.P. when fuel is cut off each cylinder in turn.

All suffer from the defects that the pressure conditions are altered when firing ceases and the loss so measured includes a compression-expansion loss which, as already explained, has not the status of a mechanical loss.

The most reliable measurements of the mechanical efficiency of Diesel Engines are those obtained by means of an indicator of a type suited to the revolutions of the engine, worked by a well designed and properly adjusted indicator gear. Other methods are chiefly useful in estimating the subdivision of the lost mean pressure so deduced.

Typical Values of Mechanical Efficiency.—Before attempting to analyse the lost mean pressure into its contributory parts, a list of typical values is given on p. 58; the higher figures refer in general to larger sizes of engines, but quite small engines may yield high efficiencies if rubbing surfaces are curtailed.

The above figures must be regarded as rough guides as there are considerable differences in rated full load brake mean pressure in different designs.

Subsidiary Losses.—The discharge of a cooling water pump seldom exceeds 12 gallons per B.H.P. hr. against a head of 15 lb. per in.^{2} Assuming a pump efficiency of 70% the percentage loss amounts to

$$100\frac{12\times 10\times 15\times 2{\cdot}3}{0{\cdot}7\times 1{,}980{,}000}=0{\cdot}3\%$$

If the brake mean pressure is 75 lb./in.^{2} the lost mean pressure due to the cooling water pumps amounts to $0{\cdot}3\times 75\div 100=$ about *0·2 lb./in.*^{2}

When lubricating oil is used for piston cooling the discharge of the oil pump may amount to about $\frac{2}{3}$ of the above at possibly 50% higher pressure giving the same loss of about *0·2 lb./in.*^{2} The loss is commonly much less than this figure.

An airless injection fuel pump may discharge against a pressure of 6000 lb./in.^{2} If the fuel consumption is 0·4 lb. per B.H.P. hr. and the fuel density 55 lb./ft.^{3}, and a pump efficiency of 70% is assumed, the percentage loss is :—

$$100\,\frac{0{\cdot}4\times 6000\times 144}{0{\cdot}7\times 55\times 1{,}980{,}000}=0{\cdot}4\%$$

Type of Engine.	ηp Brake mean pressure lb./in.²	$p(1-\eta)$ lost mean pressure lb./in.²	η Mechanical Efficiency %
Four stroke trunk type—air injection . .	75	35 to 25	68 to 75
Four stroke crosshead type—air injection . .	75	25	75
Four stroke double acting type—air injection .	70	25	74
Two stroke trunk type—air injection . .	63	31 to 22	67 to 74
Two stroke crosshead type—air injection . .	63	22	74
Two stroke double acting type—air injection .	60	22	73
Four stroke trunk type—airless injection . .	75	25 to 17	75 to 82
Four stroke crosshead type—airless injection .	75	17	82
Two stroke crank-case compression type—airless injection . . .	40	13	75
Two stroke trunk type—airless injection . .	60 to 75	25 to 16	70 to 83
Two stroke crosshead type—airless injection .	60 to 75	14	81 to 84
Two stroke double acting type—airless injection .	60 to 75	12	83 to 87

and the lost mean pressure due to the fuel pump amounts to about *0·3 lb./in.*^{2}.

In a four stroke engine, the biggest item in valve gear operation is the lifting of the exhaust valve against a pressure of about 45 lb./in.^{2}. If the exhaust valve diameter is 0·4 of the cylinder diameter and the lift 0·1 of the stroke, the lost mean pressure must be less than

$$45 \times 0{\cdot}1 \times 0{\cdot}4^2 = \text{about } 0{\cdot}7 \textit{ lb./sq. in.}$$

which is probably sufficient to cover the complete loss due to valve gear and cam-shaft operation.

These rough calculations shew that the subsidiary losses referred to are covered by about $\frac{1}{2}$ to $1\frac{1}{2}$ lb./in.^{2} according to the type of engine.

Fly-wheel Windage.—According to a formula due to Stodola the horse power expended in the windage of a circular disc is given by

$$\text{H.P.} = \beta \times 10^{-6} \left(\frac{\text{D}}{100}\right)^2 \left(\frac{\text{u}}{100}\right)^3 . \gamma . 10^6$$

in which,

β=constant 1·9 to 2·6 average say 2·3.
D=diameter of disc in cms.
u=peripheral speed of disc in cms./sec.
γ=density of fluid surrounding disc in Kg./cc.

The density of air at 760 m/m. of mercury and 15 °C is 0·00123 gr./cc. $\therefore\ \gamma = 0{\cdot}00123 \div 1000$ and $\gamma \times 10^6 = 1{\cdot}23$,

$$\therefore\ \text{H.P.} = \frac{2{\cdot}8}{10^6}\left(\frac{\text{D}}{100}\right)^2\left(\frac{\text{u}}{100}\right)^3$$

Expressing D in inches and u in ft./sec. the constant becomes

$$2{\cdot}8\ (2{\cdot}54)^2(30{\cdot}5)^3 \times 10^{-6} = 0{\cdot}51$$

$$\text{and H.P.} \quad = 0{\cdot}51\left(\frac{\text{D ins.}}{100}\right)^2 . \left(\frac{\text{u ft./sec.}}{100}\right)^3$$

For a disc wheel of conventional shape the approximate variation of the constant is as follows :—

$\frac{\text{Width of wheel}}{\text{Diameter}}$ =	0	0·1	0·2	0·3	0·4
Constant =	0·51	1·02	1·53	2·04	2·55

Example :

Single cylinder four stroke engine 9 in. dia. $\times$ 12 in. stroke; indicated mean pressure 100 lb./in.^{2}, 500 R.P.M.; $u=100$ ft./sec., width of wheel 0·2 diameter.

$$\text{I.H.P.} = \frac{100 \times 1 \times 64 \times 500}{66{,}000} = 48{\cdot}5$$

$$\text{Diameter of wheel} = \frac{100 \times 12 \times 60}{\pi \times 500} = 45{\cdot}5 \text{ in.}$$

$$\text{Windage H.P.} = 1{\cdot}5 \times {\cdot}455^2 \times 1^3 = 0{\cdot}31$$

$\therefore$ lost mean pressure due to windage,

$$= \frac{0{\cdot}31 \times 100}{48{\cdot}5} = \mathit{0{\cdot}64\ lb./in.^2}$$

If the revolutions are only 300 R.P.M. the I.H.P. becomes 29 and the wheel diameter 76 in. and the lost mean pressure works out at 3·0 lb./in.^{2}

An experiment with a single cylinder gas engine gave about 1·5 lb./in.^{2} for the fly-wheel windage loss.

With multicylinder engines the loss as calculated above is to be divided by the number of cylinders.

The loss is evidently greatest in single cylinder engines with large fly-wheels having a high peripheral speed.

Air Compressors for Blast Injection.—The lost mean pressure due to the indicated work of a blast air compressor amounts at full load to about 6lb. per sq. in. falling to about 3 lb./in.^{2} at no load. The mechanical friction of the compressor raises these figures to about *7·5 lb./in.*^{2} and *4·5* respectively. These figures refer to engines developing about 100 to 1000 B.H.P. per cylinder, whether two stroke or four stroke with atmospheric induction and indicated mean pressures from about 80 to 100 lb./in.^{2} Higher mean pressures can be developed under normal full load conditions if larger compressors are fitted. According to published test results of a supercharged four stroke engine, the compressor losses (making an allowance of 1·5 lb./in.^{2} for friction) were as under :—

Supercharge.	M.I.P.	Compressor Loss.
5 lb./in.^{2}	168 lb./in.^{2}	15·5 lb./in.^{2}
3·5 „	157 „	14·5 „
Atmospheric	120 „	10·5 „

With moderately rated engines having a supercharge of about 5lb./in.^{2} and developing a full load M.I.P. of about 130 lb./in.^{2} the compressor loss is commonly about *8·5 lb./in.*^{2}
Example : A large four stroke engine with blast compressor develops an indicated mean pressure of 85 lb./in.^{2} with a mechanical efficiency of 74%. The total lost mean pressure is therefore 85(1–0·74)=22 lb./in.^{2} If the compressor is driven separately there will be a deduction of 7·5 lb./in.^{2} leaving 22—7·5=14·5 lb./in.^{2} and the mechanical efficiency of the engine less compressor will be,

$$(85-14{\cdot}5)\div 85=83\%.$$

Two Stroke Engine Scavengers.—We are not concerned here with separately driven blowers or pumps but only with those scavenge pumps which are directly driven from the engine. These may be :—

(1) Reciprocating Pumps.
(2) Positive Displacement Rotary blowers.
(3) Centrifugal blowers.

The mean pressure absorbed depends on,

(a) The ratio of volume of air delivered by the pump to the cylinder volume.
(b) The scavenge pressure.
(c) The efficiency of the pump.

These factors are very variable and approximate indications only can be given.

In crank-case compression engines the volume of air delivered per stroke is about 60 to 70% of the stroke volume of the engine. The maximum scavenge pressure is about 6 or 7lb./in.^{2} and the mean pressure about *2·75 to 3·0* lb./in.^{2}. As there are no friction losses associated with crank-case scavenge these figures also represent the lost mean pressure due to scavenging.

Designers of engines with positive or centrifugal blowers usually aim at practically complete scavenge of the exhaust products and this involves delivering a volume of scavenge air in excess of the cylinder volume. The percentage excess required to do this depends on the arrangement of the ports or valves for air and exhaust ; in general, arrangements giving a uniflow end-to-end scavenge require the smallest percentage excess of air amounting to about 20 to 30%. Less favourable methods require a greater excess of air up to 40 or 50% or even more.

The scavenge pressure in a given engine varies approximately at the square of the R.P.M.* As between different designs,

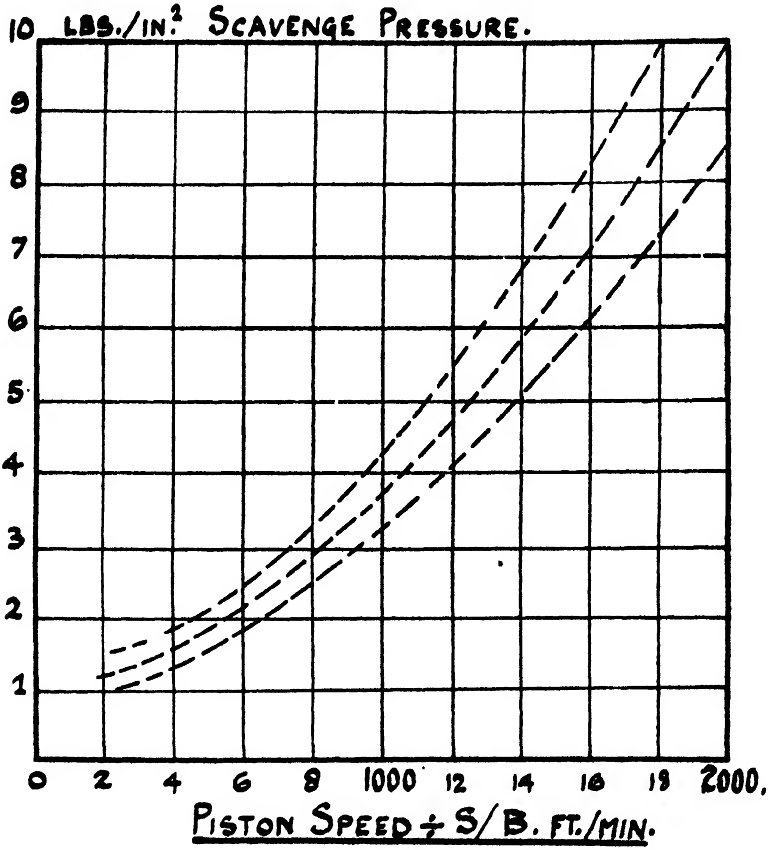


FIG. 20.

the scavenge pressure for the same bore, stroke and R.P.M. depends on the excess air percentage and the time integrals of the areas available for the admission of air and escape of exhaust gas. When both air and exhaust are controlled by

* Unless there is exhaust resonance, in which case the scavenge pressure may be nearly constant over a certain limited range of revolutions.

ports, an increase of stroke (bore and R.P.M. remaining constant) is not necessarily associated with an increase of scavenge pressure, since the length of the ports may be increased in the same ratio as the stroke, without making any increased percentage sacrifice of stroke available for compression and expansion. On this account piston speed *per se* is

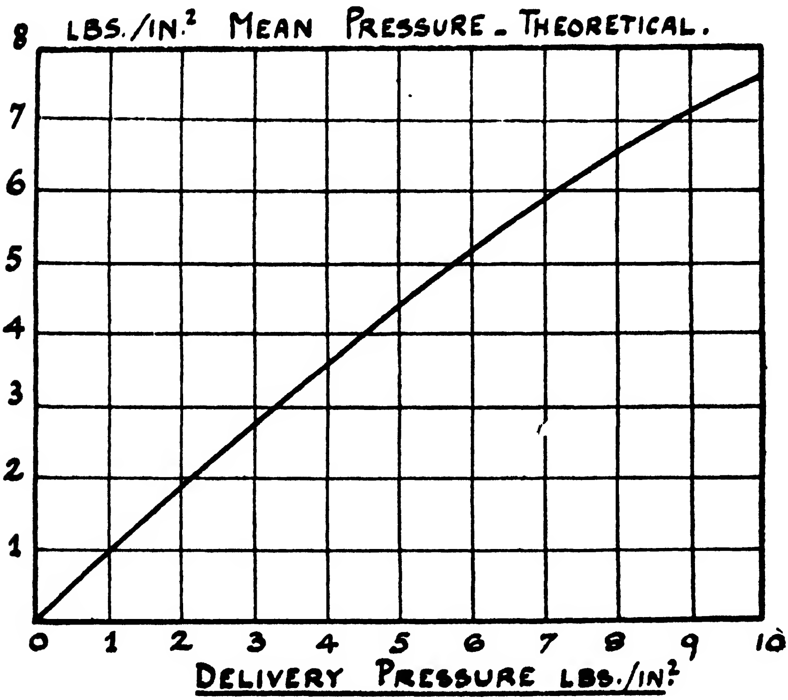


FIG. 21.

not a good criterion of two stroke engine performance. It is easier to obtain a good performance at a high piston speed if the stroke bore ratio is high than if it is low. A more useful criterion is piston speed÷stroke/bore ratio. Fig. 20 shews typical relations between scavenge pressure and this variable.

Fig. 21 shews the mean pressure of a theoretical compression diagram with delivery against various scavenge pressures. The total lost mean pressure due to the scavenger is the mean pressure as given by Fig. 21 multiplied by the excess air co-efficient (one plus a fraction) and divided by the adiabatic efficiency of the scavenger (including mechanical friction).

Large centrifugal blowers have an overall adiabatic efficiency of about 70 to 75%, not including any driving gear. Large positive rotary blowers may have an efficiency as high. Smaller pumps and blowers have in general lower efficiencies down about 50% or even less. In respect of efficiency there seems to be comparatively little to choose as between centrifugal blowers, positive displacement rotary blowers and reciprocating pumps. The efficiency (apart from mechanical losses) of any of these types when direct driven from an engine appears to be nearly independent of revolutions since the scavenge pressure and the pump losses increase together nearly as the revolutions squared. Example: A large two stroke engine has a direct driven blower delivering against a scavenge pressure of 2·2 lb./in.^{2}. The excess air co-efficient is 1·3 and the overall adiabatic efficiency of the blower is 70%. The theoretical mean pressure is 2·0 lb./in. and the lost mean pressure due to the blower,

$$=2{\cdot}0\times1{\cdot}3\div0{\cdot}7=3{\cdot}4 \text{ lb./in.}^2$$

Example: A small high speed two stroke engine has a direct driven blower delivering against a scavenge pressure of

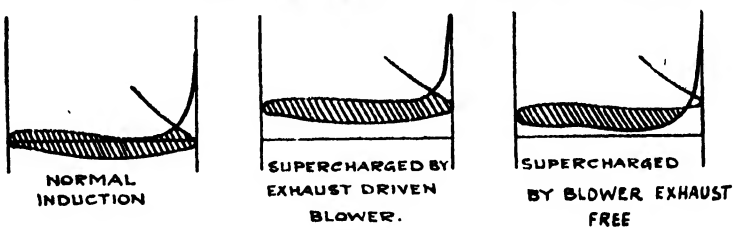


Fig. 22.

5 lb./in.^{2} The excess air co-efficient is 1·5 and the overall efficiency of the blower is 50%. The theoretical mean pressure is 4·15 lb./in.^{2} and the lost mean pressure,

$$=4{\cdot}15\times1{\cdot}5\div0{\cdot}5=12{\cdot}4 \text{ lb./in.}^2$$

In spite of the loss incurred, high scavenge pressure may be justified if sufficient supercharging effect is thereby attained.

Pumping Loss in Four Stroke Engines.—The pumping losses incurred during the exhaust and suction strokes of a four cycle engine are measured by the mean height of the negative loop of a light spring indicator diagram (Fig. 22). The lost mean pressure so incurred depends on the design of the air inlet and exhaust piping arrangements but chiefly on the area of inlet and exhaust valves. Typical values are shewn in Fig. 23 on a basis of gas velocity through the valves. The gas velocity is a nominal figure given by,

$$\text{Piston speed (ft./sec.)} \left(\frac{\text{Diameter of cylinder}}{\text{,, \quad ,, valve}}\right)^2$$

If the engine is supercharged without abnormal restriction of the exhaust, the exhaust-suction loop may be positive

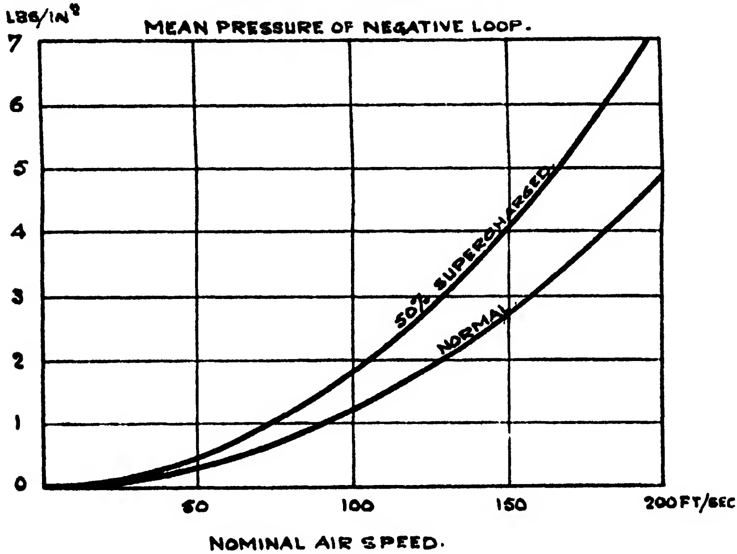


Fig. 23.

instead of negative. The mean pressure so regained is approximately equal to the supercharging pressure (above atmosphere) minus the mean pressure loss to be expected in a normal engine.

If the engine is supercharged by an exhaust driven centrifugal blower the air pressure at full load may exceed the exhaust back pressure in which case the pumping loss is reduced to the extent of the differences.

The losses incurred by engine driven superchargers can be computed as for two stroke engine scavenge pumps (see p. 75).

Friction of Piston Rings.—Worn piston rings and cylinder liners give evidence of the existence of substantial friction between these parts, but numerical data is scarce. A two stroke trunk engine $15\frac{3}{4}$ in. bore $\times 17\frac{1}{2}$ in. stroke with separately driven blast air and scavenge pumps, developing a mean indicated pressure of about 100 lb./in.^{2} at 325 R.P.M. in the course of a full load test, was found towards the end of the test to have improved in mechanical efficiency by about 5% above the normal value observed by frequent indicator cards during

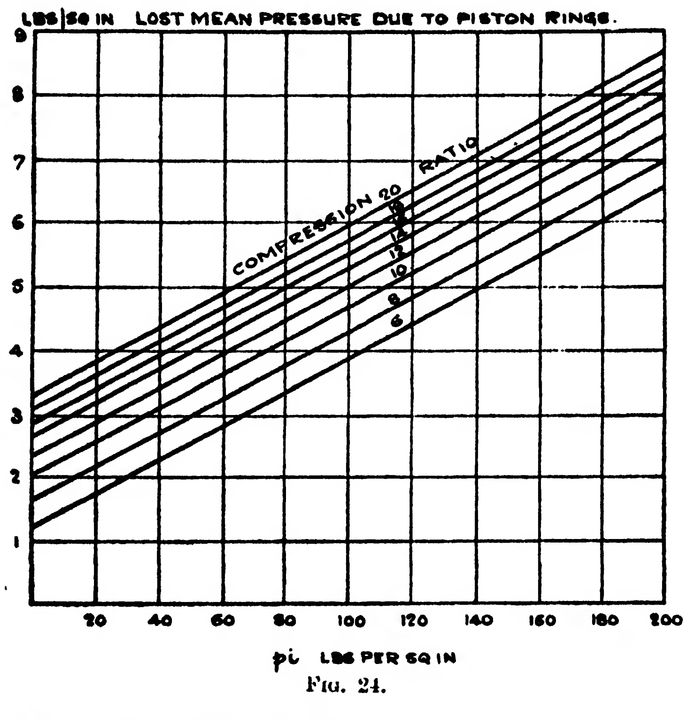


FIG. 24.

the trial. The following day the engine failed to start and on removing the piston all the rings (six in number) were absent. The pieces of the rings were afterwards found on the shop roof. This experience suggests a lost mean pressure in two stroke engines of about 1 lb./in.^{2} per piston ring (not including scrapers). Other experiments with four stroke engines indicate that an improvement of about 2 or 3% results from the removal of 3 rings out of 6. On the assumption that the ring loss is proportional to the mean pressure during compression and expansion, Fig. 24 has been drawn up as an estimate of the probable loss to be expected with six rings in a two cycle engine at various compression ratios and various mean pressures (indicated). It seems reasonable to increase the figures by say 25% for four cycle engines on account of friction on the idle strokes and to make a proportional reduction if fewer rings are used.

Stuffing Box.—The stuffing box of a D.A. engine contains a number of rings (usually six or more) having a diameter about ·25 to ·35 of the cylinder diameter. The loss incurred by friction of these rings applies to the bottom of the cylinder only. The loss may therefore be expected to be about ¼ of the main piston ring loss or say about 0·8 lb./sq. in.

Bearing Friction.—Data regarding engine bearing friction is almost nil and we are thrown back on indirect evidence. The oil cooler of a large crosshead engine having water cooled pistons accounts for only about 2% or so of the total heat supplied, equivalent to a lost mean pressure of about 4 lb./in.^{2} Ostensibly this includes the crosshead friction, but the oil is cooled by radiation losses and (in a marine engine) by contact with the ship's bottom. Small stationary engines run satisfactorily without oil coolers. Evidence of this sort points to the bearing loss being comparatively small, probably less than 4 lb./in.^{2} Some useful guidance is obtainable by direct calculation. As indicated in Chapter V similar bearings with the same oil, the same percentage clearance and the same intensity of pressure have theoretically the same co-efficient of friction at the same R.P.M. regardless of size. This establishes R.P.M. as a more relevant criterion than rubbing speed. A very simple calculation proves the foregoing statement as regards bearings which run so fast that the clearance is nearly the same all round. If pressures remain constant, reduced revolutions result in closer contact at reduced rubbing velocity and eventually a speed is reached at which the friction is a minimum and further reduction of revolutions increases the friction. This effect which applies also to pistons, goes a long way to account for the observed comparative constancy of the mechanical losses of large engines between say [illegible] and 100 R.P.M. With these considerations in mind we may expect a curve of lost mean pressure due to bearing friction on a base of revolutions to be flat at low revolutions. The experiments of Beauchamp Tower and others show this effect quite clearly. An estimate of the friction of highspeed bearings (or pistons) can be obtained by calculating the drag required to shear the oil film, assuming that the clearance remains full of oil and that the journal (or piston) remains central with the bearing (or cylinder). Calculations on these lines will be given later. For bearings running at say 10,000 R.P.M. subject to a pulsating load of say 2000 lb./in.^{2} these assumptions are probably not far from the truth. Even at lower speeds of a 1000 R.P.M. or so the journal would not deviate from the centre of its bearing by more than about 20% or so of the clearance provided the clearance remained full of oil. This, however, is not possible with forced lubrication under moderate pressure. The pulsating load squeezes out the oil until the film reaches such a mean thickness that the amount squeezed out per revolution is balanced by the amount supplied per revolution by pressure or otherwise. At low revolutions therefore the drag is considerably greater than that required to shear a continuous cylindrical oil film of thickness equal to half the diametrical clearance.

On the other hand experiments such as those of Beauchamp Tower on bearings subject to loads of constant direction and intensity are only applicable to very low speeds. A fair approximation to engine conditions may be expected to be given by a curve which approximates to Tower's results at low revolutions and our previous assumption at high revolutions.

Consider a bearing of diameter D, length L, and semi-clearance f; the diametrical clearance as measured by " leads " is 2 f. Let n=R.P.M. and μ=viscosity of oil, then :—

The peripheral speed $= \dfrac{\pi D n}{60}$.

The drag per unit of area $= \dfrac{\mu \pi n}{60} \cdot \dfrac{D}{f}$.

The total surface $= \pi D L$.

The total drag $= \dfrac{\mu \pi n}{60}\left(\dfrac{D}{f}\right)\pi D L$.

The drag work per revolution $= \dfrac{\mu \pi n}{60}\left(\dfrac{D}{f}\right)\pi^2 D^2 L$.

The swept volume of the cylinder $= \dfrac{\pi B^2 S}{4} \cdots \begin{cases} B = \text{Bore.} \\ S = \text{Stroke.} \end{cases}$

The lost mean pressure in a single-acting *2 cycle* engine (or D.A. 4 cycle engine)—one cycle per revolution—due to the friction of such a bearing under the assumed conditions is equal to the drag work per revolution ÷ the cylinder volume, i.e. :—

$$\frac{\mu\pi n}{60}\left(\frac{D}{f}\right)\pi^2 D^2 L \frac{4}{\pi B^2 S}$$

$$\doteq \frac{\pi^2}{15}\mu n\left(\frac{D}{f}\right)\left(\frac{D}{B}\right)^2\left(\frac{L}{S}\right)$$

An average value of μ for a main bearing or big end is 0·5 poises (dyne sec./cm.^{2}) equal to about 220 Redwood No. 1 or 1/140,000 lb. sec./in.^{2}.

Let $n=10{,}000$ R.P.M. and $D/f=2000$ (i.e. diametrical clearance 1/1000 part of diameter which is a fair value for high speed bearings ; then the lost mean pressure

$$=\frac{\pi^2\times 10{,}000\times 2000}{15\times 140{,}000}\left(\frac{D}{B}\right)^2\left(\frac{L}{S}\right)$$

$$=94{\cdot}5 \text{ (or say } 100)\left(\frac{D}{B}\right)^2\left(\frac{L}{S}\right)$$

and *pro rata* for other revolutions.

A typical value of $\left(\frac{D}{B}\right)^2\left(\frac{L}{S}\right)$ is about 0·1, and in order to get the independent variable nearly the same numerically as the revolutions it is convenient to plot the mean pressure loss due to one bearing as a function of $10n\left(\frac{D}{B}\right)^2\left(\frac{L}{S}\right)$, see Fig. 25 dotted line. The full line approaches the dotted line for high values of this function and the lower part of the curve is guided by Tower's bearing experiments as explained below.

Tower's bearing measured 4″ diameter × 6″ long; when running at 150 R.P.M. with oil having a viscosity equal to that assumed above (0·5 poise) and loaded with 600 odd lb./in.^{2} of projected area the drag was 0·81 lb. per in.^{2} of projected area, i.e. $0{\cdot}81\times 24=19{\cdot}5$ lb. Suppose the bearings formed a main bearing of a Diesel Cylinder 6·5″ diameter × 10″ stroke (2 cycle); the lost mean pressure due to the bearing,

$$=\frac{19{\cdot}5\times\pi\times 4}{332}=0{\cdot}73 \text{ lb./in.}^2$$

$$\text{and } 10n\left(\frac{D}{B}\right)^2\left(\frac{L}{S}\right)=1500\times 0{\cdot}615^2\times 0{\cdot}6=340,$$

giving the marked point on the lower part of the curve.

For S.A. four stroke engines the ordinates should be doubled to allow for the fact that the engine makes two strokes per cycle. For D.A. two stroke engines the ordinates should be halved.

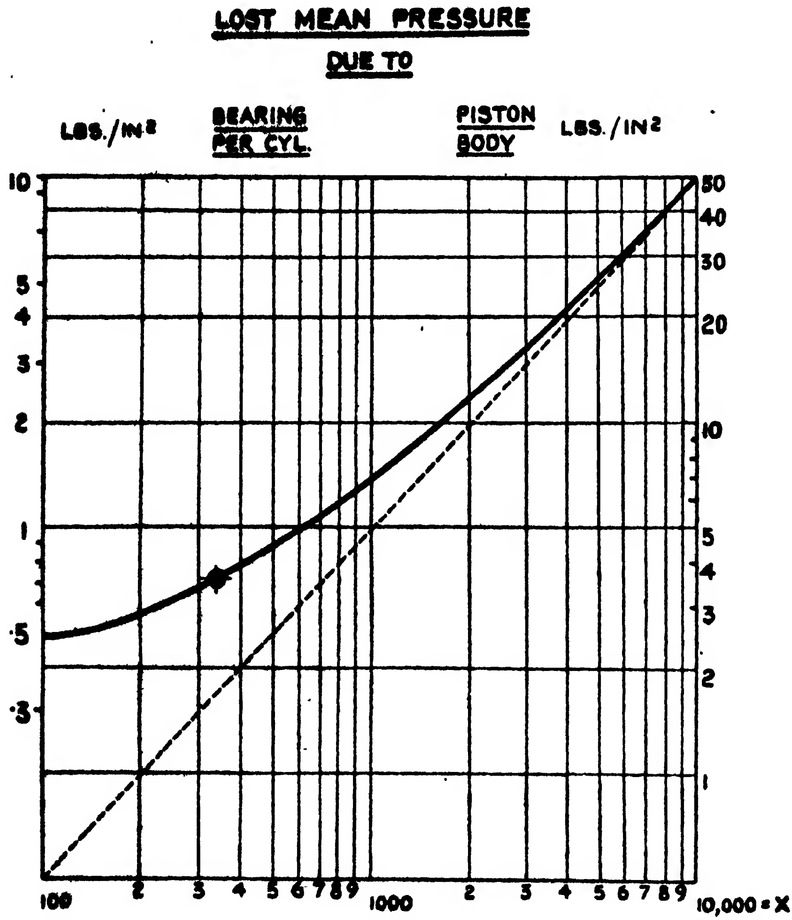


FIG. 25.

To obtain the total crank-shaft loss, account must, of course, be taken of the number of bearings per cylinder; for example, a 6 cylinder engine usually has 6 big end bearings and 7 or 8 main bearings making in all 2·2 or 2·3 bearings, per cylinder.

Also the bearings may not be all of the same diameter and length. In view of the rough character of the estimate it is admissible to average the diameters and lengths.

Piston Friction.—It will be noted that Fig. 25 is also used to estimate piston friction. The assumption made here is that the general behaviour of a piston is similar to that of a bearing journal. The lower the revolutions the closer the contact. As before, consider a high speed engine and assume the piston clearance full of oil. Let T=length of piston and f the semi-clearance.

$$\text{The mean velocity}=\frac{2Sn}{60}$$

$$\text{The drag per unit area}=\frac{Sn\mu}{30f}$$

$$\text{The total drag}=\frac{Sn}{30f}\cdot\pi BT\mu$$

$$\text{The drag work per rev.}=\frac{Sn}{30f}\cdot\pi BT\mu.\,2S$$

$$\text{The cylinder volume}=\frac{\pi B^2S}{4}$$

$$\text{The lost mean pressure}=\frac{\mu Sn}{30f}\cdot\frac{\pi B.T}{\pi B^2S}\cdot 8S$$

$$=0{\cdot}27\mu n\left(\frac{S}{B}\right)\left(\frac{B}{f}\right)\left(\frac{T}{B}\right)$$

A fair value of $\mu=0{\cdot}25$ dyne sec./cm.^{2}=1/280,000 lb. sec./sq. in.
Let n=10,000 and B/f=2000
The lost mean pressure,

$$=\frac{0{\cdot}27\times10{,}000\times2000}{280{,}000}\left(\frac{S}{B}\right)\left(\frac{T}{B}\right)=\text{about }20\left(\frac{S}{B}\right)\left(\frac{T}{B}\right)$$

A typical value of $\left(\frac{S}{B}\right)\left(\frac{T}{B}\right)$ is 2·5, we therefore adopt $\frac{n}{2{\cdot}5}\left(\frac{S}{B}\right)\left(\frac{T}{B}\right)$ as independent variable. When this function is 10,000 the lost mean pressure is $20\times2{\cdot}5=50$ lb./in.^{2}.

The diagram (Fig. 25) is plotted to log. co-ordinates so that the curve for bearings can be used for pistons by a simple shift of scale as shewn on the diagram, assuming, of course, that we are justified in assuming proportionality as between friction of bearings and pistons throughout the whole range of the independent variable.

As before, the plotted curve refers to S.A. 2 cycle or D.A. 4 cycle engines and the ordinates must be multiplied by 2 for S.A. four stroke and divided by 2 for D.A. two stroke engines.

It will be observed that the use of this variable involves the assumption that the lost mean pressure due to piston friction in slow speed engines (see Fig. 25) is as insensitive to stroke bore ratio and piston length as it is to revolutions; this agrees with the author's experience.

If the piston surface is considerably cut away or relieved, some allowance should, no doubt, be made for this in estimating an equivalent length.

Crosshead Friction.—Diesel Engine crosshead shoes usually have a length about equal to the cylinder diameter and an area (in the direction of principal thrust) about equal to the cross sectional area of the cylinder ($\pi B^2/4$). These figures suggest the following rule:

Lost mean pressure due to crosshead friction = half the lost mean pressure due to a trunk piston of length equal to B.

This rule appears to agree fairly well with meagre data on crosshead friction as measured by the heat absorbed by water cooled guides, which however are now seldom used as forced lubrication provides ample cooling in most instances.

Applications of the Foregoing Rules.—To illustrate the principles of this chapter the mechanical efficiencies of a number of typical engines have been worked out by tabulating the several losses as given by the foregoing rules.

Four Stroke Trunk Type—6 cylinders. $4\frac{1}{2}'' \times 5\frac{1}{2}'' \times 1800$ R.P.M.

Item.	Lost mean pressure lb./in.^{2}
Pumping Losses.—Piston speed = 1650 ft./min. = 27·5 ft./sec. Valve dia. = 0·40 cyl. dia. Nominal air speed = $27{\cdot}5 \div 0{\cdot}40^2$ = 172 ft./sec. By Fig. 23 lost mean pressure	3·6
Piston Rings.—Indicated mean pressure about 120 lb./in.^{2}. Compression ratio 14. Loss by Fig. 24 = 5·3 lb./in.^{2}. Increase by 25% for four stroke engine and × by 4/6 for 4 rings instead of 6 . . .	4·8
Piston Body.—S/B = 5·5/4·5 = 1·22. T/B = 1·35; R.P.M. × (S/B) × (T/B) ÷ 2·5 = 1100. Using Fig. 25 and doubling for four stroke the lost mean pressure	16·0

	Lost mean pressure lb./in.^{2}
Bearings.—6 big ends and 7 main bearings$=2 \cdot 17$ bearings per cyldr. $D/B=0 \cdot 68$. $L/S=\cdot 28$ (mean); $10 \times R.P.M. \times (D/B)^2\ (L/S)=2300$. By Fig. 25 lost mean pressure$=2 \cdot 7 \times 2 \times 2 \cdot 17$ . . .	11·6
Subsidiaries . . . . say	1·5
Total	37·5
Brake Mean Pressure say	85·0
Indicated ,, ,, by addition	122·5
Mechanical Efficiency $100 \times 85 \div 122 \cdot 1=$	69·5%

Four Stroke Trunk Type—6 cylinders. $14'' \times 21'' \times 300$ R.P.M.

Item.	Lost mean pressure lb./in.^{2}
Pumping Losses.—Piston speed$=1050$ ft./min.$=17 \cdot 5$ ft./sec. Valve dia.$=0 \cdot 35$ cyl. dia. Nominal air speed$=17 \cdot 5 \times 0 \cdot 35^2=143$ ft./sec. By Fig. 23 lost mean pressure	2·5
Piston Rings.—Indicated mean Pressure about 90 lb./in.^{2}. Compression ratio 12. Loss by Fig. 24$=$ 4·8 lb./in.^{2}. Increase by 25% for four stroke engine and $\times$ 3/6 for 3 rings instead of 6 . . .	3·0
Piston Body.—$S/B=1 \cdot 5$. $T/B=1 \cdot 75$; $R.P.M. \times (S/B) \times (T/B) \div 2 \cdot 5=315$. Using Fig. 25 and doubling for four stroke the lost mean pressure . .	7·6
Bearings.—6 big ends and 7 main bearings$=2 \cdot 17$ bearings per cyldr. $D/B=0 \cdot 60$ (mean). $L/S=0 \cdot 3$ (mean); $10\ R.P.M. \times (D/B)^2\ (L/S)=325$. By Fig. 25 lost mean pressure$=0 \cdot 7 \times 2 \times 2 \cdot 17$ . .	3·0
Subsidiaries . . . . say	1·5
Total	17·6
Brake Mean Pressure say	75·0
Indicated Mean Pressure by addition	92·6
Mechanical Efficiency $100 \times 75 \div 92 \cdot 6=$	81%

Above assumes airless injection. If air blast injection is used with compressor driven off engine there will be an additional loss of about 6 lb./in.^{2} and the mechanical efficiency becomes $75 \div (92\cdot6+6) = 76\%$.

Four Stroke Trunk Type—6 cylinders 25″×48″×130 R.P.M.
(Supercharged by exhaust turbo-blower.)

Item.	Lost mean pressure lb./in.^{2}
Pumping Losses.—Piston speed=1040 ft./min.=17·4 ft./sec. Valve dia.=0·35 cyl. dia. Nominal air speed$=17\cdot4\div0\cdot35^2=142$ ft./sec. Lost mean pressure of normal engine from Fig. 23=2·5 lb./in.^{2} Supercharging pressure 5 lb./in.^{2} and exhaust pressure 4 lb./in. Lost mean pressure$=(2\cdot5\times19\cdot7\div14\cdot7)-(5-4)$	2·4
Piston Rings.—Indicated mean pressure about 130 lb./in.^{2}. Compression ratio 12. Loss by Fig. 24= 5·8 lb./in.^{2}. Increase by 25% for 4 cycle engine and × 4/6 for 4 rings instead of 6	4·8
Piston Body.—S/B=1·92. T/B=2·0; R.P.M.×(S/B) ×(T/B)÷2·5=200. Using Fig. 25 and doubling for four stroke the lost mean pressure . =	6·0
Bearings.—6 big ends and 7 main bearings=2·17 bearings per cyldr. D/B=0·65 (mean) L/S=0·24 (mean); 10 R.P.M.$\times(D/B)^2$ (L/S)=132. By Fig. 25 lost mean pressure=0·53×2·17×2 . .	2·3
Subsidiaries.—(Water and lub. oil pumps driven separately) . . . . say	1·0
Total	16·5
Brake Mean Pressure say	115·0
Indicated Mean Pressure	131·5
Mechanical Efficiency $100\times115\div131\cdot5=$	87·5%

Above assumes airless injection. If air blast injection is used with compressor driven off engine there will be an additional loss of about 8·5 lb./in.^{2} and the mechanical efficiency becomes $115\div(131\cdot5+8\cdot5)=82\%$.

Two Stroke Trunk Type—8 cylinders $10'' \times 15'' \times 700$ R.P.M.
(Airless injection.)

Item.	Lost mean pressure lb./in.^{2}
Scavenger Losses.—Piston speed=1750 ft./min. P.S.×B/S=1170 ft./min. Scavenge pressure 4 lb./in.^{2}. Excess air ratio 1·4. Blower efficiency 60%. Theoretical mean pressure of compression 3·5 lb./in.^{2}. Lost mean pressure $3{\cdot}5 \times 1{\cdot}4 \div 0{\cdot}6$.	8·2
Piston Rings.—Indicated mean pressure about 100 lb./in.^{2}. Effective compression ratio 14. Loss by Fig. 24	5·3
Piston Body.—S/B=1·5. T/B=2·0; R.P.M.×(S/B) (T/B)÷2·5=840. Using Fig. 25 the lost mean pressure	6·0
Bearings.—8 big ends and 10 main bearings=2·25 bearings per cyldr. D/B=0·7, L/S=0·2; 10 R.P.M. (D/B)^2 (L/S)=690. By Fig. 25 lost mean pressure =1·05×2·25	2·4
Subsidiaries . . . . . . say	1·5
Total	23·4
Brake Mean Pressure say	75·0
Indicated Mean Pressure	98·4
Mechanical Efficiency 100×75÷98·4=	76%

Two Stroke Trunk Type—3 cylinders $12'' \times 15'' \times 300$ R.P.M.
(Airless injection, Crank-case scavenge.)

Item.	Lost mean pressure lb./in.^{2}
Scavenger Losses.—Max. scavenge pressure 7 lb./in.^{2}; loss=mean scavenge pressure	3·0
Piston Rings.—Indicated mean pressure about 45 lb./in.^{2}. Effective compression ratio 13. Loss by Fig. 24	3·7
Piston Body.—S/B=1·25, T/B=1·4; R.P.M.×(S/B) ×(T/B)÷2·5=210. Using Fig. 25 lost mean pressure	3·0

	Lost mean pressure lb./in.^{2}
Bearings.—3 big ends, 4 main bearings=2·3 bearings per cylinder. D/B=0·6, L/S=0·5; 10 R.P.M.$\times$(D/B)^2 (L/S)=540. By Fig. 25 lost mean pressure =0·9$\times$2·3	2·1
Subsidiaries . . . . say	1·5
Total	13·3
Brake Mean Pressure . . . . . . say	36·0
Indicated Mean Pressure	49·3
Mechanical Efficiency 100$\times$36·0$\div$49·3=	73%

Two Stroke Double Acting Type—8 cylinders 40″$\times$60″$\times$120 R.P.M.

(Direct Driven Blower.)

Item.	Lost mean pressure lb./in.^{2}
Scavenger Losses.—Piston speed=1200 ft./min. Piston speed$\times$B/S=800 ft./sec. Scavenge pressure 2·5 lb./in.^{2}. Excess air ratio 1·3. Blower efficiency 0·65. Lost mean pressure=2·3$\times$1·3$\div$0·65 .	4·6
Piston Rings.—Indicated mean pressure about 90 lb./in.^{2}. Effective compression ratio 12. 6 piston rings each end of piston. By Fig. 24 lost mean pressure	5·0
Stuffing Box	0·8
Piston Body.—S/B=1·5. T/B=1 after deducting relieved length. R.P.M.$\times$(S/B) (T/B)$\div$2·5=72. By Fig. 25 lost mean pressure	1·2
Bearings.—8 big ends, 10 main bearings=2·25 bearings per cylinder. D/B=0·75, L/S=0·25 (mean). 10 R.P.M.$\times$(D/B)^2 (L/S)=170. By Fig. 25 lost mean pressure=0·26$\times$2·25 . .	0·6
Crosshead Shoes	0·6
Subsidiaries.—(Separate water and oil pumps). say	0·5
Total	13·3
Brake Mean Pressure	75·0
Indicated Mean Pressure	88·3
Mechanical Efficiency 75$\times$100$\div$88·3=	85%

The above refers to airless injection. With air blast injection the efficiency would be

$$75 \times 100 \div (88{\cdot}3 + 6) = 79{\cdot}5\%$$

With separately driven scavenge blower and airless injection or separately driven blast compressor the mechanical efficiency would be

$$75 \times 100 \div (88{\cdot}3 - 4{\cdot}6) = 89{\cdot}5\%$$

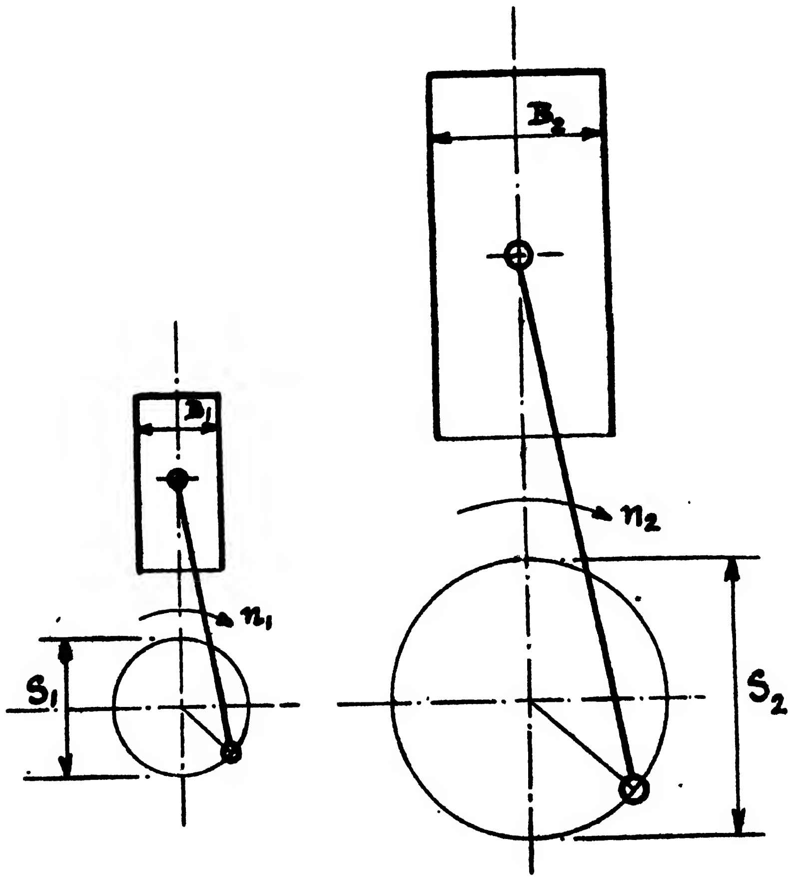
CHAPTER V

THE PRINCIPLE OF SIMILITUDE

The Scope of the Chapter.—The principle of similitude enables a general survey to be made of problems which require for their complete solution elaborate analysis or experimental research. By use of the principle, the findings of practical experience or deliberate research may be applied to cases other than those directly experienced. By applying the principle to all of the relevant physical properties of the materials involved (metal, gas, water, oil) a grasp is gained of fundamental principles of design not easily obtained in any other way. We shall use the principle in relation to such topics as :—

(1) Power output in relation to cylinder dimensions.
(2) Power output in relation to space and weight.
(3) Stresses due to pressure and inertia.
(4) Vibration stresses.
(5) Volumetric, mechanical and thermal efficiencies.
(6) Temperature distribution and temperature stresses.
(7) Lubrication ; bearing problems.
(8) Fuel injection and combustion problems.

The fundamental conception in the line of thought adopted in this chapter is that of a series of similar engines of different sizes. By similar engines we understand engines which are geometrically similar figures, i.e. exact models of each other, with corresponding parts made of the same materials. We shall consider amongst other things the results of running all these engines at the same piston speed. With the conception of a series of engines at the back of the mind it is actually only necessary to consider the relations which exist between any two members of the series distinguished by suffixes 1 and 2, the numeral 2 referring to the larger (Fig. 26). The interesting fact emerges that there is no way of adjusting the revolutions of engine 2 in such a way that its behaviour exactly imitates that of engine 1 in all respects simultaneously. The conclusion is reached that engines of broadly the same type but differing in size should not be made geometrically similar and the discussion indicates the directions in which departures from similarity should be made.



$$\frac{B_2}{B_1} = \frac{n_1}{n_2} = r = \frac{S_2}{S_1}$$

Fig. 26.

Symbols.

Let B = Cylinder bore.
S = Stroke of piston.
n = Revolutions per minute (R.P.M.).
d = Diameter of bearing or pipe or other circular part.
L, l = Any typical linear dimension.
p = Indicated mean pressure.
η = Mechanical efficiency.
ρ = Density of fluid or solid.
μ = Viscosity of fluid.
$v = \mu/\rho$ = kinematic viscosity of fluid.
k = Conductivity of solid or fluid.
s = Specific heat ,, ,,
$h = k/s\rho$ = diffusivity ,, ,,
x, y, z = Rectangular co-ordinates.
u, v, w = Velocities in x, y, z directions.
U, V = Typical velocities.
T = Temperature.
t = Time.
W = Weight.
C = Volume.
$\frac{B_2}{B_1} = r.$

Power Output.—The brake horse power of a single cylinder two stroke engine is given by the formula :—

$$\text{B.H.P.} = \frac{\eta p.\ S.\ \pi/4 B^2 n}{12 \times 33{,}000}$$

$$= \frac{\eta p.\ S.\ B.^2 n}{505{,}000} \quad\ldots\ldots\ldots\ldots\ldots\ldots (1)$$

if B & S are measured in inches and p in lb./in.^{2}. The output of a four stroke cylinder is one-half that given by the above formula (S.A. in each case).

The relative powers of two similar engines of bore B_1 and B_2 respectively are therefore given by :—

$$\frac{\text{B.H.P.}_1}{\text{B.H.P.}_2} = \frac{\eta_1 p_1 S_1 B_1^2 n_1}{\eta_2 p_2 S_2 B_2^2 n_2}$$

If the two engines are run at the same piston speed, $S_1n_1 = S_2n_2$ and then,

$$\frac{B.H.P._1}{B.H.P._2} = \frac{\eta_1 p_1}{\eta_2 p_2}\left(\frac{B_1}{B_2}\right)^2 = \frac{1}{r^2}\frac{\eta_1 p_1}{\eta_2 p_2}$$

It will be shewn later that the mechanical efficiency η_2 (larger engine) may be expected to exceed η_1 (smaller engine) and that the obtainable indicated mean pressure p_1 (smaller engine) may be expected to exceed p_2 (larger engine); assuming these two effect cancel so that $\eta_1 p_1 = \eta_2 p_2$ then,

$$\frac{B.H.P._1}{B.H.P._2} = \frac{1}{r^2} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (2)$$

That is, with similar engines running at the same piston speed, the full power varies as the square of the linear scale or as the piston area. The proportionality of power to piston area (with constant piston speed) also holds good as between engines which have different ratios of stroke to bore and this is the basis of the much abused Treasury rating formula for motor cars, viz.:—

$$B.H.P.\text{ (per cylinder)} = 0{\cdot}4B^2 \quad \ldots\ldots\ldots\ldots (3)$$

which is satisfied by a piston speed of 1000 ft./min. and $\eta p =$ 67 lb./in.^{2} (four stroke engine).

The adjoining table gives the appropriate values of the constant to be substituted for 0·4 in formula (3) for various values of the brake mean pressure ηp and the piston speed.

Power Output in Relation to Weight and Space.—If W_1 and W_2 are the weights of similar engines made of the same materials

$$\frac{W_1}{W_2} = \left(\frac{B_1}{B_2}\right)^3 = \frac{1}{r^3}$$

$$\text{and} \quad \frac{W_1/B.H.P._1}{W_2/B.H.P._2} = \frac{1}{r} \quad \ldots\ldots\ldots\ldots\ldots\ldots (4)$$

The volume and weight per B.H.P. vary directly as the linear scale. If an aircraft engine with 5″ bore cylinders weighs 2 lb. per B.H.P., then a similar engine of 25″ bore developing the same brake mean pressure at the same piston speed would weigh 10 lb. per B.H.P. It has often been maintained that it is not practicable to use the same relative thickness of metal in small engines as in big engines, but this seems to be a fallacy. A 30″ bore cylinder liner of cast iron would normally have a thickness

PISTON SPEEDS—FEET PER MINUTE

M.I.P. lb./in.²	600	650	700	750	800	850	900	950	1000
80	0·572	0·620	0·667	0·715	0·762	0·810	0·857	0·906	0·954
85	0·607	0·658	0·708	0·760	0·810	0·860	0·912	0·962	1·01
90	0·643	0·697	0·750	0·805	0·858	0·912	0·966	1·02	1·07
95	0·680	0·736	0·794	0·850	0·907	0·964	1·02	1·08	1·13
100	0·715	0·775	0·834	0·894	0·953	1·01	1·07	1·13	1·19
105	0·751	0·814	0·876	0·940	1·00	1·06	1·13	1·19	1·25
110	0·786	0·852	0·917	0·983	1·05	1·11	1·18	1·25	1·31

The tabulated figures are values of I.H.P. per in.² of cylinder bore² for two stroke engines. For four stroke engines divide the values by 2. Alternatively the figures give B.H.P. in relation to brake mean pressure.

of 2″ in accordance with a Lloyd's rule for engines with atmospheric induction. A geometrically similar engine 5″ bore would have a liner thickness of 0·33″, a size which presents no difficulty whatever, and similarly for other dimensions. If anything smaller relative thicknesses are admissible with small engines since their iron castings have a higher tensile strength than thick ones and small steel forgings are more homogeneous and finer in grain than larger ones.

By similar reasoning to that used for equation (4), the floor space for B.H.P. is the same for large and small similar engines. This remains true of an installation of several engines, provided the spaces between the engines are determined by the size of engine parts which require handling.

Stresses due to Pressure and Inertia.—The direct stresses on any section due to pressure vary as the pressure × the piston area ÷ the area of the section. If f_1 and f_2 are the stresses on corresponding sections of two similar engines, and A_1 and A_2 are the areas of the sections, then, assuming the pressure P is the same for both,

$$\frac{f_1}{f_2}=\frac{P\pi B_1^2/4}{P\pi B_2^2/4}\times\frac{A_2}{A_1}=\left(\frac{B_1}{B_2}\right)^2\times\left(\frac{B_2}{B_1}\right)^2=1$$

i.e. the stresses are the same in each case.

Similarly in the case of bending or twisting actions; the moments vary as B^3 and the moduli of the sections also vary as B^3 so that the stresses are the same.

Next consider the inertia of the piston. The inertia force of the piston divided by the piston area is sometimes referred to as the inertia pressure. If ρ is the density of the piston material, the relative masses of two pistons are :—

$$\rho_1 B_1^3 : \rho_2 B_2^3$$

Their relative accelerations are,

$$n^2_1 B_1 : n_2^2 B_2$$

The ratio of inertia pressures is therefore :—

$$\frac{\rho_1 B_1^3}{\rho_2 B_2^3}\cdot\frac{n_1^2 B_1}{n_2^2 B_2}\cdot\frac{B_2^2}{B_1^2}=\frac{\rho_1 B_1^2 n_1^2}{\rho_2 B_2^2 n_2^2}=\frac{\rho_1}{\rho_2}\left(\frac{V_1}{V_2}\right)^2 \quad\ldots\ldots(5)$$

which $=1$ if $\rho_1=\rho_2$ and $V_1=V_2$

That is, the inertia pressures and consequently the inertia stresses are the same if the materials are of the same density and the piston speeds are equal.

Vibration Stresses are a particular kind of inertia stresses, but as the amplitudes of the movements are not prescribed by the geometry of the engine, the reasoning used in the preceding section does not suffice.

When a machine is running at a speed n_c which is a severe critical speed with respect to a particular mode of vibration, that particular mode of vibration is exalted out of proportion to other possible modes which may therefore be left out of consideration or treated separately. The system may then for purposes of dynamical calculation be replaced by a suitably chosen system consisting of a single mass M under the control of a spring. The frequency of such a system is given by the formula :—

$$F=\frac{60}{2\pi}\sqrt{\frac{g}{y}}$$

where y is the deflection of the mass due to a force equal to the weight of mass M applied in the direction of motion.

Comparing two such similar systems relating to two similar engines of bore B_1 and B_2, $M_1/M_2=(B_1/B_2)^3$. The relative sectional areas of the springs are as $B_1^2 : B_2^2$, so that the stresses f_1 and f_2 due to the application of the weights of M_1 and M_2 are related as :—

$$\frac{f_1}{f_2}=\left(\frac{B_1}{B_2}\right)^3\cdot\left(\frac{B_2}{B_1}\right)=\frac{B_1}{B_2}$$

Since the lengths of the springs vary as $B_1 : B_2$, the deflections are related by,

$$\frac{y_1}{y_2}=\frac{B_1}{B_2}\cdot\frac{B_1}{B_2}=\left(\frac{B_1}{B_2}\right)^2=\frac{1}{r^2}$$

also $$\frac{F_1}{F_2}=\sqrt{r^2}=r=\frac{n_1}{n_2}\ldots\ldots\ldots\ldots\ldots\ldots(7)$$

The vibration frequencies of similar engines vary inversely as the linear scale and corresponding critical speeds occur at the same piston speed. It follows from the last statement taken in conjunction with the previous article that the forces causing vibration reckoned per unit of piston area are the same for all similar engines. If the damping were solely due to internal friction (hysteresis) of the material, and the materials were the same for all corresponding parts of the similar engines under consideration, it would follow that the vibration stresses would also be the same. The same would hold good for damping due to solid friction. In so far as viscous damping of lubricated surfaces is concerned, further investigation is required. Also it should be kept in mind that the properties of nominally the same material, e.g. 28–32 ton mild steel may vary with size of the forging.

So far as frequencies and the relative seriousness of critical speeds are concerned, a single diagram shewing the various criticals over the whole working range of revolutions can serve for all members of a series of similar engines, by a simple adjustment of the scale of revolutions.

Volumetric Efficiencies.—Similar engines running at the same piston speed have the same nominal gas speeds at all corresponding points in the induction and exhaust systems. If "h" is the loss of head at any constriction, such as a valve, then

$$h=\text{const}\cdot\frac{V^2}{2g}\cdot f\left(\frac{Vd}{v}\right).,$$

the quantity in brackets being Reynolds' number.

If "h" represents the loss of head in a pipe or channel (full),

$$h=\text{const}\cdot\frac{L}{d}\cdot\frac{V^2}{2g}\cdot f\left(\frac{Vd}{v}\right),$$

where L/d is the ratio of length to diameter of the pipe or channel. In either case since L/d is the same for similar engines we have :—

$$\frac{h_1}{h^2}=\frac{f(Vd_1/v)}{f(Vd_2/v)}$$

For flow in smooth pipes f (Vd/v) decreases slowly as Vd/v increases, being approximately halved when Vd/v is increased about ten times. In rough pipes $f(Vd/v)$ is nearly constant. For flow through poppet valves $f(Vd/v)$ appears to be nearly independent of Vd/v over ranges so far investigated. The same appears to be approximately true for ports.

The general conclusion is that similar engines may be expected to have almost identical volumetric efficiencies at the same piston speeds. If any difference is observable the advantage may be expected to rest with the larger sizes on account of lower pipe friction co-efficient at higher values of Reynolds' number.

An influence operating in the opposite direction is the heating of the air charge by contact with the cylinder walls. It will be shewn later that the larger of two similar cylinders will be hotter if both operate under the same conditions of combustion and at the same piston speed.

The Mechanical Efficiency is defined to be

$$\eta=\frac{\text{B.H.P}}{\text{I.H.P}}=\frac{\text{Brake mean pressure}}{\text{Indicated mean pressure}}$$
$$=\frac{p-f}{p}=1-\frac{f}{p}$$

where p=indicated mean pressure and f is the so-called lost mean pressure, including, in addition to friction proper, the work required to drive compressors, scavenge pumps, etc., all referred to the piston displacement in accordance with the conventional formula for horse power.

The co-efficient of solid friction is independent of size; whether this is true of the so-called boundary lubricated friction does not appear. The friction due to the shearing of oil films per unit of area varies inversely as the thickness of the film. If therefore the similar engines have similar piston clearances and bearing clearances, and use oil of the same viscosity under the conditions of running the lost mean pressure due to the shearing of oil films may be expected to vary inversely as the scale. On the whole, therefore, of two similar engines working at the same piston speed and mean pressure the larger engine will tend to have the better mechanical efficiency. This conclusion is confirmed by actual experience of ranges of nearly similar engines. On the other hand, the modern tendency in the design of small Diesel Engines is to reduce the rubbing surfaces (of the pistons in particular) to an extent which has never been tried in large sizes, thereby reducing, if not actually eliminating, the difference in friction loss as between big and small engines.

Thermal Efficiency.—It has frequently been shewn that with an internal combustion engine rejecting at full load 25% of the heat of the fuel to the water jackets, the total suppression of heat losses during the power stroke would not increase the output per unit of fuel supplied by more than about 8%. It will be shewn later that other things being equal the heat lo s during combustion. of a small engine is relatively greater than that of a similar engine of larger size running at the same piston speed. As between cylinders of say 5″ and 30″ diameter the difference is not great but the advantage is with the larger cylinder. In practice small cylinders are commonly given a higher compression ratio than is usual with large cylinders, for the sake of easy starting; also small cylinders usually work with a higher maximum pressure than is considered desirable with large cylinders. As a result quite small cylinders may yield efficiencies as high or higher than those of larger cylinders.

Heat Loss by Gaseous Radiation.—The purely thermal radiation of intensely hot gas is a function of temperature. If gas were perfectly transparent to its own radiation the percentage of the total heat lost in this way would be independent of size and nearly directly proportional to the time occupied by the cycle, i.e. inversity as the revolutions. The heat flux to the walls of the combustion chambers of similar engines running at the same piston speed would then vary as the cylinder bore directly so far as concerns the heat received from this type of radiation.

If gas were perfectly opaque to its own radiation, the rate of emission of heat would be proportional to the area of the combustion chamber (instead of the volume as in the previous supposition). In these circumstances the percentage of heat so lost would again vary inversely as the revolutions, but also inversely as the cylinder bore, so that the percentage of heat lost would be the same for similar engines running at the same piston speed.

Actually gas is neither perfectly opaque nor perfectly transparent, and the intensity of radiation appears to be approximately proportional to the square root of the cylinder diameter over a certain range but tending towards a limit for very large sizes.

Part of the radiation during combustion may be regarded as an inevitable by-product of combustion proportional to the fuel consumed and independent of the time occupied by the process.

So far as heat loss by radiation is concerned the advantage lies with the small engine running at high revolutions. The observed fact that the heat loss from small cylinders is relatively greater than that lost from large ones running at the same piston speed indicates that the loss due to radiation is smaller than that lost by convection and conduction.

Heat Loss by Convection and Conduction.—When a hot fluid (liquid or gas) flows between cold solid boundaries the transmission of heat from the fluid to the boundary is a complex process involving processes of convection and conduction which are perfectly definite at any point and instant but which cannot, even in theory, be separated in the aggregate effect. At any point at any instant there is a flux in a definite direction of convected heat proportional to the velocity, temperature and specific heat of the gas at the point in question. There is also a flux (in possibly a different direction) of conducted heat proportional to the temperature gradient and the conductivity at that point. At a point of zero temperature gradient the flux is purely convective; at a point of zero velocity (i.e. at the boundary) the flux is purely conductive.

Let a be the co-efficient of heat transmission, i.e. the flux at the boundary divided by the difference in temperature between the boundary and the practically uniform temperature near the middle of the mass of fluid. The general theory of the whole process leads to the equation :—

$$\frac{aL}{k}=f\left(\frac{VL}{v},\frac{h}{v}\right)$$

in which k, h and v refer to the properties of the fluid (see list of symbols, p. 80). L is a characteristic linear dimension and V a characteristic velocity.

In flow through tubes the equation is found to take the form :—

$$\frac{ad}{k}=\text{const.}\left(\frac{Vd}{v}\right)^n.\ \phi\left(\frac{h}{v}\right)$$

with n=about 0·8 for air and 0·85 to 1·0 for liquids.

For flow perpendicular to the outside surface of tubes or bundles of tubes, n=about 0·5 to 0·7.

For application to heat flow in engine cylinders we may write the equation :—

$$\frac{aB}{k}=\text{const.}\left(\frac{VB}{v}\right)^n \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (8)$$

in which B=Bore of cylinder and V=piston speed. Then, for two similar engines,

$$\frac{a_1}{a_2}=\frac{B_2}{B_1}\cdot\left(\frac{V_1}{V_2}\cdot\frac{B_1}{B_2}\right)^n=\left(\frac{V_1}{V_2}\right)^n\cdot\left(\frac{}{r}\right)^{1-n}$$

According to the researches of Nüsselt and others n=about 0·6; adopting this value and taking the same piston speed for both engines,

$$\frac{a_1}{a_2}=r^{0\cdot4} \quad\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\quad (9)$$

The heat transmission co-efficient is therefore greater for the smaller engine when both run at the same piston speed. The percentage (%) of heat lost by convection and conduction will vary as the co-efficient of transmission × the cooling surface × time of exposure ÷ the volume of the combustion chamber, so that,

$$\frac{(\%)_1}{(\%)_2}=\frac{a_1}{a_2}\cdot\left(\frac{B_1}{B_2}\right)^2\cdot\frac{n_2}{n_1}\cdot\left(\frac{B_2}{B_1}\right)^3$$
$$=\frac{a_1}{a_2}=r^{0\cdot4}$$

Similarity (but not equality) of heat flow as between two similar engines occur when,

$$\frac{V_1B_1}{v}=\frac{V_2B_2}{v},\ \text{i.e. when } V_1B_1=V_2B_2,$$

i.e. piston speed inversely as cylinder diameter. If this relation holds (see equation 8),

$$a_1B_1=a_2B_2 \text{ and } \frac{a_1}{a_2}=\frac{B_2}{B_1}=\frac{V_1}{V_2} \quad\ldots\ldots\ldots\ldots\ldots\ldots\quad (10)$$

Then also,

$$\frac{(\%)_1}{(\%)_2}=\frac{B_2}{B_1}\cdot\left(\frac{B_1}{B_2}\right)^2\cdot\left(\frac{B_1}{B_2}\right)^2\cdot\left(\frac{B_2}{B_1}\right)^3=1,$$

the same percentage loss in each case.

Let T be the temperature difference between the gas and water sides of the wall and let d be the wall thickness; the condition that T should be the same for two similar engines is:—

$$a_1d_1=a_2d_2,\ \text{i.e. } \frac{a_1}{a_2}=\frac{d_2}{d_1}=\frac{B_2}{B_1}$$

which is the same condition (10) which secures similarity in respect of convection and conduction in the gas.

For example, consider an engine 30″ bore by 60″ stroke running at 100 R.P.M., i.e. a piston speed of 1000 feet per minute. A model of this engine to a scale of 1 : 5, i.e. 6″ bore by 12″ stroke would attain the same distribution of temperature and temperature stress at a piston speed of 5000 ft./min., which corresponds to 2500 R.P.M. Obviously the principle must not be pushed so far that the smaller engine reaches such high piston speed that the processes of induction, exhaust, etc., are impaired. The principle is chiefly useful as indicating the comparative immunity of small engines from heat troubles as compared with large engines and the necessity of adopting in large engines special constructions such as oil cooled pistons, etc., to deal with conditions which do not arise in small engines.

Fluctuating Temperatures.—The general equation of the conduction of heat in the interior of a solid body is :—

$$\frac{\delta T}{\delta t}=h\left\{\frac{\delta^2 T}{\delta x^2}+\frac{\delta^2 T}{\delta y^2}+\frac{\delta^2 T}{\delta z^2}\right\}$$

If T is to remain unaltered when x, y and z are increased r times, then t must be increased r^2 times

$$\text{i.e.}\ \frac{\text{R.P.M.}_1}{\text{R.P.M.}_2}=\left(\frac{B_2}{B_1}\right)^2 \therefore \frac{V_1}{V_2}=\frac{B_2}{B_1}$$

The same condition as above for steady temperature distribution.

Lubrication of Cylinders and Bearings.—The surfaces of pistons and bearings are subject to rapidly fluctuating pressures to which no one simple theory of lubrication is directly applicable. According to the Tower-Reynolds theory of the lubrication of bearings subject to steady loads, the surfaces are kept apart by a wedge-shaped film of oil dragged between the surfaces by relative motion of the surfaces so separated. The application of this principle to the case of a Michel pad in which the film thicknesses at inlet and outlet bear a fixed ratio leads to the result that the pressure per unit area (P) is related to the rubbing velocity U, the oil viscosity μ, the film thickness h and the linear size L of the pad by the formula :—

$$P=\text{const.}\frac{\mu UL.}{h^2} \quad \ldots\ldots\ldots\ldots\ldots\ldots(1)$$

Comparing two similar bearings with film thicknesses h proportional to L, the condition that P shall be the same for both is,

$$\frac{U_1}{U_2}=\frac{L_1}{L_2}\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(2)$$

That is, the rubbing speed must be proportional to the linear dimensions.

Putting $h \propto L$ in (1) and bearing in mind that $V \propto nL$ where n=revolutions per minute, equation (1) becomes,

$$P = \text{const. } \mu n \qquad (3)$$

It can be shewn that (3) applies also to circular and semi-circular bearings working with film lubrication.

Where alternating loads are concerned there is another theory of lubrication independent of relative sliding between the surfaces. Consider, for example, a piston slapping from side to side in the cylinder clearance which we take to be proportional to the cylinder bore B. Assume also that the clearance always remains full of oil. The transverse velocity of the piston in its clearance varies as Bn; so also does the velocity with which oil is squeezed out in the process. The shear per unit area of piston surface due to the squeezing out of the oil varies as the velocity and viscosity divided by the film thickness, i.e. as $\mu nB/B$, i.e. is μn. The pressure developed is proportional to the shear and therefore also varies as μn. Hence,

$$P = \text{const. } \mu n \text{ as in (3).}$$

We have therefore the remarkable result that similar engines subject to the same pressures and using the same lubricant at the same temperature have similar conditions with regard to lubrication when they all run at the same revolutions regardless of size. This conclusion is vitiated in practice by the fact that large engines running at high revolutions develop inertia loads which may be, and in fact usually are, more important in respect of lubrication, than the loads due to gas pressure. Nevertheless it is important to realise that high revolutions *per se* are favourable to good lubrication, and any unfavourable influence on bearings of high revolutions is due to the increase of inertia loading and not to the increased velocity of rubbing. If two similar engines are run at the same piston speed, equation (1) shews that the ratio h/L will be greater for the smaller engine but h absolutely greater for the larger engine in the ratio $\sqrt{L_2/L_1}$, assuming the same viscosity in each case.

Suppose we have a successful 4 cycle engine say 20″ dia. × 24″ stroke running at 200 R.P.M. and it is desired to design a new engine of the same dimensions running at say 400 R.P.M.

Considering inertia pressure only the same film thickness will be obtained if the moving parts are reduced in weight by one-half. Double the heat will now be generated in the bearings and correspondingly larger oil coolers will be required. All the inertia bearing pressures will be doubled and the most serious problem may quite likely be that of preventing cracking of the bearing metal, which is a separate consideration.

Returning to the consideration of similar engines the deductions from equation (1) may conveniently be summarised in a table under the alternative assumptions $U_1/U_2=1$, i.e. same piston speed and $U_1/U_2=L_1/L_2$, i.e. same revolutions.

Condition of Comparison.	$\frac{U_1}{U_2}=1$	$\frac{U_1}{U_2}=\frac{L_1}{L_2}$
Film thickness (h) . .	$\frac{h_1}{h_2}=\left(\frac{L_1}{L_2}\right)^{\frac{1}{2}}$	$\frac{h_1}{h_2}=\frac{L_1}{L_2}$
Intensity of shear (S) $\propto\frac{U}{h}$.	$\frac{S_1}{S_2}=\left(\frac{L_2}{L_1}\right)^{\frac{1}{2}}$	$\frac{S_1}{S_2}=1$
Total heat generated per unit of time $Q \propto SUL^2$ . .	$\frac{Q_1}{Q_2}=\left(\frac{L_1}{L_2}\right)^{1\frac{1}{2}}$	$\frac{Q_1}{Q_2}=\left(\frac{L_1}{L_2}\right)^{3}$
Quantity of oil drawn in per unit of time $O \propto ULh$.	$\frac{O_1}{O_2}=\left(\frac{L_1}{L_2}\right)^{1\frac{1}{2}}$	$\frac{O_1}{O_2}=\left(\frac{L_1}{L_2}\right)^{3}$
Rise of temperature $T \propto \frac{Q}{O}$.	$\frac{T_1}{T_2}=1$	$\frac{T_1}{T_2}=1$

The interesting conclusion from this table is that the rise of temperature of the oil passing through the bearings (assuming viscosity constant) is a function of bearing pressure and independent of revolutions.

So far as any individual engine is concerned, inertia pressures vary as the revolutions squared so that increase of revolution is generally associated with an increase of lubricating oil temperature. Furthermore, equation (1) is not strictly applicable to journal bearings or piston surfaces. In spite of these limitations the table is instructive in suggesting the comparative unimportance of rubbing speed and the fundamental importance of pressure intensity.

The comparison of the intensity of shear shews that of two similar engines running at equal piston speeds the smaller engine will be subject to greater coefficients of friction and will consequently have the lower mechanical efficiency if the design is generally similar.

This conclusion is borne out by actual tests of engines of similar design of different size. The lost mean pressure of similar engines running at the same piston speed appears to vary roughly directly as the cube root of the revolutions. The following table shews the results of this rule as applied to two pairs of similar four stroke engines.

Injection.	Airless.		Air Blast.	
B=Bore	4·0″	20″	14″	18″
S=Stroke	4·0″	20″	21″	27″
n=R.P.M.	2000	400	250	195
f=lost mean pressure lb./in.^{2}	34	20	27*	25*
ηp=broke mean pressure lb./in.^{2}	80	80	73	73
η=mechanical efficiency.	0·70	0·80	0·73	0·75
Piston	Aluminium short		Cast iron long	

The variation of lost mean pressure with the revolutions of one and the same engine is quite another matter discussed in Chapter IV. Above certain revolutions depending on the area of valves, etc., the lost mean pressure increases rapidly with revolutions.

Mechanical Injection of Fuel.—With the usual arrangement of the so-called jerk pump system the velocity of the fuel pump plunger is nearly constant from the beginning to the end of the

* Includes indicated work of compressor.

injection period. The opening of the spring loaded injection valve is preceded by the passage of a pressure wave from the pump to the valve and in some instances the pressure wave may be reflected from each end of the system more than once before the injection valve opens. A similar surge may take place after the valve has closed. This process has been studied both theoretically and experimentally (see Chapter XX). Fundamental factors in the process are the velocity of the fuel in the connecting pipe between pump and valve at the pump end at the instant at which compression of the fuel begins, and the constant velocity of about 5000 ft./sec. with which disturbances in the form of compression and expansion are propagated in the pipe. The first of these velocities determines the intensity of the pressure wave.

Apart from such phenomena of wave propagation and certain scale effects of fluid friction, it is easily shewn that similar hydraulic systems preserve similarity at all speeds and that pressures at corresponding points and instants vary as the velocity squared independently of size. When the velocities concerned are sufficiently high, wave propagation plays an important part and the pressure distribution in space and time of a given hydraulic system no longer remains similar to itself if the speed is altered.

Comparing two similar systems of different size operating with the same velocities, the pressures at corresponding points at corresponding instants are the same including all effects of wave propagation. This arises as a result of the velocity of wave propagation being independent of size.

We have therefore the important result that similar engines running at the same piston speed have the same pressure characteristics of the fuel pump system, apart from a small correction due to the deviation of friction effects from the velocity^2 law. The fuel injection system of a 4″ cylinder running at 2000 R.P.M. presents exactly the same problem as a similar 20″ cylinder running at 400 R.P.M. provided both can be got to work satisfactorily with the same period of injection measured in crank-shaft degrees and with the same injection pressure. This statement ignores any special difficulties which may arise in the manufacture of the parts of the smaller system to the required degree of accuracy; also the undesirability of making nozzle holes below a certain small diameter and such considerations.

Consider the injection of fuel into the combustion chambers of two similar engines of different sizes each provided with the same number of injection orifices of a diameter proportional to the cylinder bore. With the same injection pressure in each case the conditions of dynamic similarity are satisfied apart from effects of viscosity and surface tension if the globules in each case have a diameter proportional to the cylinder diameter. If the resistance to motion through the gases varies as the velocity squared, it can be shewn that the penetration of the jets by the time any percentage (say 90%) of the velocity is lost, will vary as the diameter of the cylinder, which is a necessary condition for equally effective distribution of fuel in two similar engines working at the same piston speed.

If it is further assumed that combustion proceeds at a constant rate per unit of surface of the oil droplets, then the time required for any percentage (say 95%) of the fuel to become burnt will vary inversely as the diameter of the droplets, i.e. inversely as the cylinder bore. This is the condition for equally good combustion in two similar engines running at the same piston speed. This is brought out by the subjoined table.

COMBUSTION OF FUEL

Similar engines running at same piston speed.

Diameter and R.P.M.	$\frac{B_1}{B_2}=\frac{n_2}{n_1}=\frac{1}{r}$
d=dia. of droplets . . δ=dia. of orifices . .	$\frac{d_1}{d_2}=\frac{\delta_1}{\delta_2}=\frac{B_1}{B_2}=\frac{1}{r}$
S=surface of droplet . .	$\frac{S_1}{S_2}=\left(\frac{d_1}{d_2}\right)^2=\frac{1}{r^2}$
R=rate of combustion weight per unit time . .	$\frac{R_1}{R_2}=\frac{S_1}{S_2}=\frac{1}{r^2}$
V=volume of droplet . .	$\frac{V_1}{V_2}=\left(\frac{d_1}{d_2}\right)^3=\frac{1}{r^3}$
t=time required for combustion of say 95% of droplet . . .	$\frac{t_1}{t_2}=\frac{V_1}{V_2}\Big/\frac{R_1}{R_2}=\frac{r^2}{r^3}=\frac{1}{r}=\frac{n_2}{n_1}$

∴ same % of fuel is consumed in a given crank angle in each case.

The above is admittedly a rough and rather hypothetical sketch of the processes involved, but it may be useful as a theoretical datum for the comparison of experimental results.

Conclusions Regarding Similar Engines.—Geometrically similar engines made of the same materials and running at the same piston speed behave similarly in respect of :—

(1) All stresses due to pressure and inertia but not dead weight. Vibration stresses are similar in character, but their actual intensity may vary somewhat with size on account of different degrees of damping.

(2) Volumetric efficiencies of four stroke engines and charging efficiencies of two stroke engines except in so far as as these are influenced by smooth pipe friction, heating effects, and difficulties associated with fine clearances of small scavenge blowers, etc.

(3) Pressure effects in the fuel system, including wave propagation (surging).

Probably also in respect of :—

(4) Distribution and combustion of fuel except in so far as they are influenced by effects of viscosity and surface tension.

They behave differently in respect of :—

(5) Lubrication ; large engines tend to have *relatively* closer contact of rubbing surfaces, but *absolutely* less close contact and lower coefficients of friction than small engines, assuming equal viscosity of lubricants.

(6) Heat flow and temperature stresses ; small engines reject a slightly greater percentage of heat to the water jackets and are subject to a correspondingly higher flux of heat than larger engines. Nevertheless their temperature differences and temperature stresses are smaller.

These considerations clearly indicate that strict geometrical similarity in design is only justified over a relatively small range of size. A few examples are given below of the kind of changes which are involved in an increase of scale.

Large engines require *relatively* finer running clearances than small engines to avoid shock at bearings and pistons and lubricating oil coolers are necessary on account of relatively less conduction of heat from the bedplate to the foundation.

They also require relatively stiffer construction of bearings (particularly big ends) to avoid distortion, since the rubbing contact is relatively closer.

Large engines require more elaborate provision for unequal expansion of all parts exposed to flame or combustion products, and the pistons require oil or water cooling, whereas small pistons maintain a moderate temperature by pure conduction from the crown to the skirt and thence to the cylinder walls.

In small engines effective use can be made of certain high tensile heat treated alloy steels which can hardly be used in the construction of large engines, not only on the score of expense, but on account of fundamental difficulties associated with the heat treatment of large forgings. Similar considerations apply with regard to the use of light alloys of aluminium.

Questions of theoretical point contact of tappets and line contact of cams and gear teeth, etc., require more careful consideration in large engines than in small on account of the relatively greater extrusion of the intervening oil film due to the longer duration of pressure.

Even after allowance has been made for all these and analogous considerations it is probably correct to say that there is a tendency to over-complication in designing large engines. This tendency should be resisted and a careful study of small engine designs may help to correct it.

There appears to be a most promising future for comparatively large high revolution engines provided adequate means are forthcoming for dealing with inertia and temperature stresses.

Literature.—Johansen, F. C., "Research in Mechanical Engineering by small-scale Apparatus."—I. Mech. E., March, 1929.

Stanton, T. E., "The Principle of Similitude in Engineering Design."—Brit. Assoc., 1916; *Engineering*, September 15, 1916.

Porter, A. W., "The Method of Dimensions."—Methuen.

Boswall, R. O., "The Theory of Film Lubrication."—Longmans.

Fishenden, M., and Saunders, O. A., "Heat Transmission."—H.M. Stationery Office.

CHAPTER VI

EXHAUST SUCTION AND SCAVENGE

Renewal of the Charge.—In the theoretical study of heat engines the charge of working fluid is supposed to remain enclosed in a working cylinder and to undergo a cycle of physical changes due to the introduction of heat from, and discharge of heat to, external sources by conduction through the walls. In actual engines of the internal combustion type one constituent of the charge, viz. the oxygen, takes an active part in the chemical processes which constitute the source of energy. In such engines, therefore, the air charge must be renewed periodically. In all existing types of oil engine the charge is renewed as completely as possible for each cycle of thermal changes.

The way in which this is done in four stroke and two stroke Diesel Engines has been described in general terms in Chapter I. In the present chapter it is proposed to discuss the questions of the discharge of exhaust gases and the introduction of a new air charge from the quantitative point of view, apart from the consideration of the mechanical details, such as valves, cams, etc., of which these processes involve the use.

At the outset it will be necessary to state in a form convenient for application, the laws governing the flow of gases through orifices.

Flow of Gases through Orifices.—Fig. 27 is intended to represent a chamber containing gas at a pressure and temperature maintained constant at the values P_1 and T_1, and from which a gaseous stream is issuing through an orifice into the surrounding space, which is filled with gas at a constant pressure P_2.

In the first instance, it will be supposed that P_1 is very much greater than P_2. Then, according to the elementary theory of the flow of gases, it can be shewn that if the gas composing the stream expands adiabatically and all the work done is expended in increasing the kinetic energy of the stream, then the velocity of the latter will increase as expansion proceeds and attain a maximum value given by :—

$$V=\sqrt{2g\,J\,K_p\,(T_1-T_2)}=109{\cdot}5\sqrt{(T_1-T_2)}\ldots\ldots(1)$$

where :—

V=maximum velocity of stream in ft./sec.
g=32·2 ft./sec.^{2} (acceleration due to gravity).
J=Joules' equivalent (778 ft. lb./B.T.U.).
K_p=Specific heat at constant pressure (for air K_p=·238 B.T.U./lb. deg. F.).
T_2=Temperature attained by the stream after adiabatic expansion from P_1 to P_2, and is given (for air) by :—

$$T_2=T_1\left(\frac{P_2}{P_1}\right)^{0{\cdot}285}\ldots\ldots\ldots\ldots\ldots\ldots(2)$$

In equation (1) the velocity in the chamber is supposed to be negligibly small compared with V. For values of P_1 up to

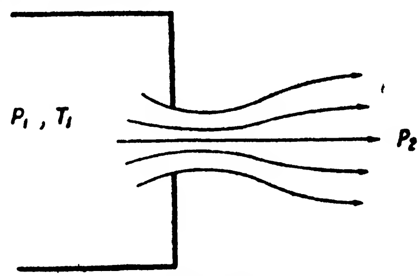


Fig. 27.

70×144 lb. per sq. ft., P_2 being atmospheric, equation (1) has been found to agree with experiment within two or three per cent, which is ample accuracy for our purpose.

It may, however, be mentioned in passing that experiments at high pressures, and particularly with steam, indicate that the elementary theory on which equation (1) is based stands in need of correction.

It is to be observed that equation (1) is equally valid for any stage of the expansion of the stream, so that if any pressure value P_a be selected lying between P_1 and P_2, then the velocity at this stage will be given by equation (1) with P_a substituted for P_2. If values of the velocity V_a be calculated for various values of P_A and the specific volumes v_A corresponding to these values, then the ratios $\frac{v_A}{V_A}$ will be proportional to the areas of the stream at the several stages of expansion. On making such a calculation it will be found that the area of the stream at first contracts and afterwards converges to a final value corresponding to the maximum velocity. In other words, the stream has a neck or throat, as shown in Fig. 27. The pressure P_c at the throat is known as the critical pressure, and for air is equal to $0{\cdot}53\ P_1$.

Furthermore, it is evident that if P_2 is equal to P_c (a contingency which was ruled out at the beginning of the discussion), then the stream will converge to its throat area and remain parallel instead of diverging.

Supposing again that $P_2 < P_c$, it is evident that for a given size of orifice the discharge will be a maximum if the throat of the stream occurs at the orifice, and in this case the discharge in lb. per sec. will be given by :—

$$Q = A\,V_c\,W_c \ldots\ldots\ldots\ldots\ldots\ldots (3)$$

where Q = discharge in lb. per sec.
A = area of orifice—ft.^{2}.
V_c = throat velocity—ft./sec.
W_c = weight in lb. per cub. ft. of the air at the conditions of temperature and pressure obtaining at the throat.

So that the discharge is independent of the back pressure P_2 so long as $P_2 < P_c$.

On the other hand, there is no guarantee that the throat of the stream will coincide with the orifice in every case, so that in general the discharge given by equation (3) has to be multiplied by a discharge coefficient less than unity, in order to give the discharge observed by experiment.

For air and exhaust gases the following figures may be used :—

Discharge coefficient for sharp-edged orifices or ports—0·65
,, ,, ,, mushroom valves . . 0·70

The velocity calculated from equation (1), multiplied by the appropriate coefficient, may conveniently be called the apparent velocity referred to the actual area of the orifice.

The Suction Stroke.—By way of application of the preceding formulæ, we may consider the suction stroke of a Diesel Engine. The retreating piston creates a partial vacuum and air passes in from the atmosphere via the inlet valve in consequence. Supposing it is desired to limit the pressure difference to 1 lb. per sq. in., then—

$$P_1 = 14{\cdot}7 \times 144 \quad P_2 = 13{\cdot}7 \times 144 \quad T_1 = \text{say } 520^\circ \text{ F. abs.}$$

$$\text{Then } T_2 = 520\left(\frac{13{\cdot}7}{14{\cdot}7}\right)^{0{\cdot}285} = 509{\cdot}7^\circ \text{ F. abs.}$$

$$\text{And } V = 109{\cdot}5\sqrt{520 - 509{\cdot}7} = 351 \text{ ft./sec.}$$

Using a coefficient of 0·7 for the value, the apparent velocity referred to the valve area is

$$351 \times 0{\cdot}7 = 246 \text{ ft./sec.}$$

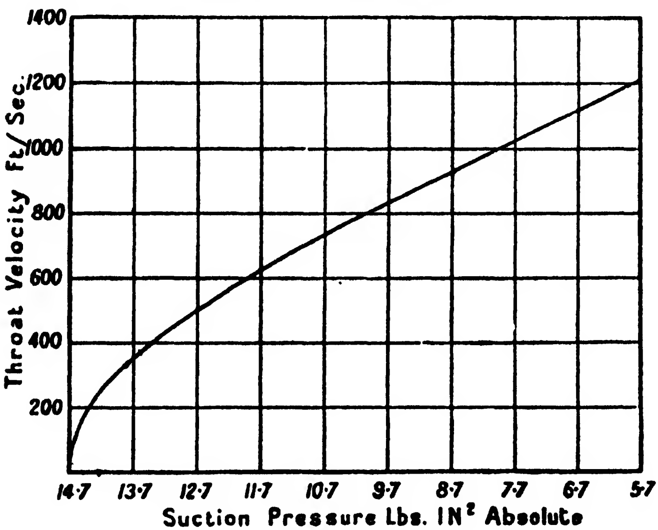


Fig. 28.

If the pressure difference is to be constant at 1 lb./in.² then the valve and its operating cam must be so designed that

$$\frac{\text{Instantaneous piston speed}}{246}\text{ (ft./sec.)} = \frac{\text{Instantaneous valve area}}{\text{Piston area}}$$

As indicated in Chapter I, it is better to allow the inlet valve to open before the inner dead centre, and in order to obtain a maximum charge of air the valve should not seat until 20° or 30° after the outer dead centre has been passed.

Fig. 28, which is a curve connecting calculated velocity and suction pressure, is the result of repeating the above for different values of P_2 on the assumption that

$$P_1 = 14{\cdot}7 \times 144 \text{ and } T_1 = 520^\circ \text{ F. abs.}$$

Scavenging of Two Stroke Cylinders.—Before the scavenge air supply is put into communication with the cylinder whether by valves or ports, the exhaust slots should have been sufficiently uncovered for the cylinder pressure to have fallen to practically atmospheric pressure. As will be shewn later, this takes place with great rapidity owing to the high temperature of the gases. On this account the introduction of scavenge air may be assumed to take place against a pressure differing but very slightly from atmospheric. This, however, does not apply to the later stage of the process in those valve scavenged or controlled port scavenge engines in which scavenge air is

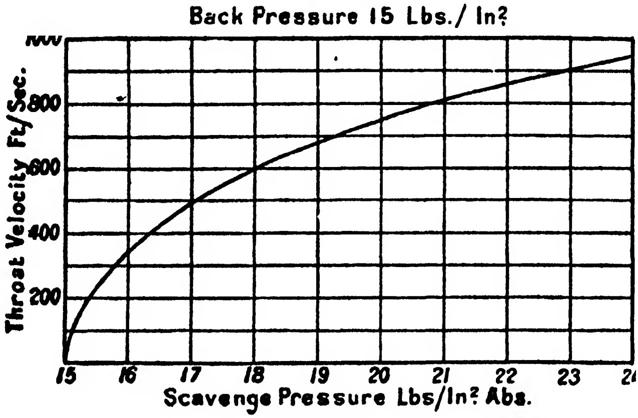


Fig. 29.

forced into the cylinder after the exhaust ports have been covered. During this latter stage the cylinder pressure continually increases until either the scavenge valves (or ports) close and compression begins, or until the cylinder contents and the scavenge air supply are in equilibrium.

The first stage of the process may be dealt with in a similar manner to that indicated in the preceding article, and Fig. 29 shews the calculated throat velocity of the scavenge air plotted against the absolute scavenge pressure, on the assumption that the back pressure is 15 lb. per sq. in. abs. and the temperature of the scavenge air supply is 130° F. (591° F. abs.), this being a good average figure found in practice.

Before applying the above to a concrete example some observations will be made on port and valve areas.

If a fluid flows for a time " t " through an orifice of constant area "A," with a velocity "V," then the volume "v" discharged is evidently given by :—

$$v = V\,(A \times t)$$

If on the other hand A is variable, then

$$v = V \int A \cdot dt$$

The quantity $\int A \cdot dt$ known as the time integral of the area is readily found by plotting A as ordinate against "t" as abscissæ, as in Fig. 30, which exhibits the opening area of a valve from the time it lifts ($t = o$) to the time it seats.

The time integral of the valve area for the whole interval is the area under the curve. For any other interval from, say, $t = t_1$ to $t = t_2$ the time integral of the area is the area bounded by curve, the axis of t and the two ordinates which define

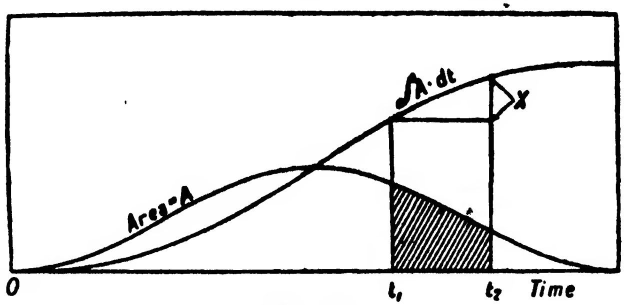


FIG. 30.

t_1 and t_2. This area is shewn shaded on the diagram for particular values of t_1 and t_2.

It is often convenient to take as the unit for "t" $\frac{1}{100}$th sec. or $\frac{1}{1000}$th sec., or even 1 degree of revolution of the crank-shaft.

Also it is very convenient to plot on a "t" base a curve whose ordinates represent $\int A \cdot dt$ from $t = o$.

This has been done on Fig. 30. It will be noticed that between the instants $t = t_1$ and $t = t_2$ the value of $\int A \cdot dt$ has increased by an amount X. The volume discharged between these instants would therefore be $(X \cdot V)$.

Example of Scavenge Calculation.—Data for two stroke engine :—

Bore of cylinder	10″
Stroke	15″
Revolutions per minute . . .	300

Two scavenge valves $3\frac{1}{2}''$ diam., maximum lift $1''$, opening 25° before bottom dead centre and closing 60° after. Lift curve harmonic, i.e. the lift plotted on a time base is a sine curve. Exhaust ports occupy 60% of the circumference of the cylinder bore and become uncovered by the piston 15% before the end of the stroke. The connecting rod is 5 cranks long. Fig. 31 has been drawn to shew the relation between percentage of

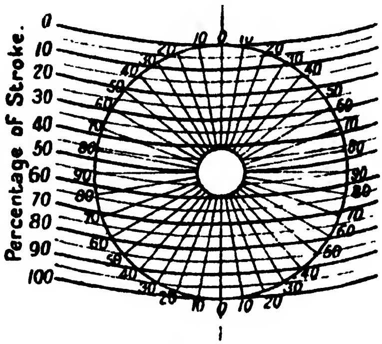


FIG. 31.

stroke and the number of degrees between the crank position and the bottom dead centre. From this diagram it will be seen that the ports are uncovered 50° before B.D.C. and covered again 50° after B.D.C. Compression space 7% of the stroke volume. Free air capacity of scavenger 50% in excess of stroke volume of impulse cylinders.

Stroke volume $= \dfrac{\cdot 785 \times 10^2 \times 15}{1728}$ $= 0{\cdot}68$ ft.^{3}

Compression space $= 0{\cdot}07 \times 0{\cdot}68$ $= 0{\cdot}05$ „

Stroke volume + compression space $= 0{\cdot}73$ „

Volume of scavenge air (at atmospheric pressure and temperature) delivered per cylinder per revolution $= 0{\cdot}68 \times 1{\cdot}5$ $= 1{\cdot}02$ „

Maximum exhaust port area

$$= \frac{0{\cdot}6 \times \pi \times 10 \times 0{\cdot}15 \times 15}{144} = 0{\cdot}295 \text{ „}$$

Maximum scavenge valve area

$$= \frac{2 \times \pi \times 3{\cdot}5 \times 1}{144} = 0{\cdot}153 \text{ ft.}^2$$

Intermediate values of the exhaust port and scavenge valve areas have been plotted on a crank angle base on Fig. 32.

Number of seconds corresponding to 1 degree of revolution of the crank-shaft

$$=\frac{60}{300}\times\frac{1}{360}=0{\cdot}000556 \text{ sec.}$$

In Fig. 32 values of $\int A{\cdot}dt$ for both exhaust and scavenge areas have been plotted in accordance with the preceding article. In particular, since the scavenge valve lift is harmonic,

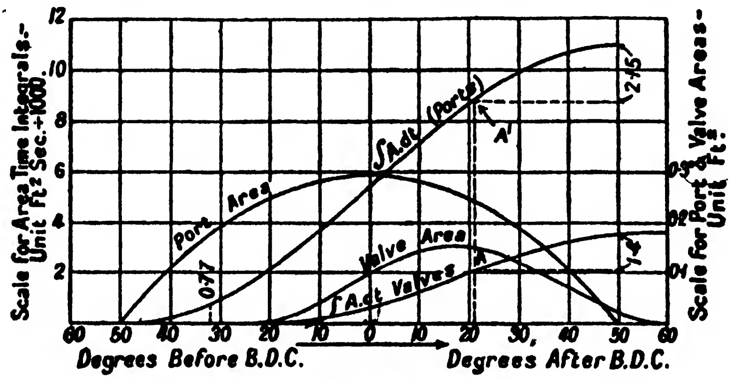


FIG. 32.

the mean area is one-half the maximum, and the final value of $\int A{\cdot}dt$ is given by :—

$$\frac{0{\cdot}153}{2}\times 0{\cdot}000556\times 85^\circ=10^{-3}\times 3{\cdot}61 \text{ ft.}^2 \text{ secs.}.$$

And since the port area curve is nearly parabolic the maximum value of $\int A{\cdot}dt$ for the ports is given by :—

$$0{\cdot}295\times \tfrac{2}{3}\times 0{\cdot}000556\times 100^\circ=10^{-3}\times 10{\cdot}94 \text{ ft.}^2 \text{ secs.}$$

The intermediate values are readily found by planimetering the areas under the port and valve area curves.

Scavenge Air Pressure.—A first approximation to the value of the pressure in the scavenger air pipe required by the conditions of the problem is readily obtained, on the assumption that the scavenge air is delivered against a constant pressure of 15 lb. per sq. in. abs.

Volume discharged . .	$=1{\cdot}02$ ft.^{3}
$\int A{\cdot}dt$	$=3{\cdot}61\times 10^{-3}$

Apparent velocity . . $=\dfrac{1{\cdot}02\times10^{-3}}{3{\cdot}61}=283$ ft./sec.

Dividing by a discharge coefficient of 0·7 the
"calculated" velocity is $283\div0{\cdot}7=405$ ft./sec.

And the corresponding scavenge air pressure from Fig. 29 is :—

16·35 lb. per in.^{2} abs.

This figure cannot, however, be accepted as final, since the effect of discharging the excess of air through the exhaust ports has been ignored.

Assume for trial a scavenge pressure of 17 lb. per in.^{2} abs. The calculated velocity from Fig. 29 is 498 ft./sec. and the apparent velocity :—

$$=498\times0{\cdot}7=349 \text{ ft./sec.}$$

The value of $\int A{\cdot}dt$ which must be attained to fill the stroke volume and clearance space (0·73 ft.^{3}) is therefore :—

$$\frac{0{\cdot}73}{349}=2{\cdot}09\times10^{-3}$$

This corresponds to point A on Fig. 32, and shews that with the assumed scavenge pressure the cylinder would be filled with scavenge air at about 21° after the bottom dead centre. From this point until the point where the exhaust ports are covered three processes are occurring simultaneously, viz. :—

(1) Scavenge air is escaping out of the exhaust ports.
(2) Scavenge air is entering the cylinder and raising the pressure of its contents.
(3) The piston is rising and reducing the volume of the cylinder contents, at the same time tending to raise the pressure.

The pressure which exists in the cylinder at the instant at which the piston covers the ports may conveniently be termed the "initial charge pressure," and will be denoted by P_1. Now if the value of P_1 were equal to the scavenge pressure, a probable value for it, the amount of air introduced into the cylinder in the interval under consideration would be about equal to :—

$$\tfrac{2}{3}\int A{\cdot}dt \text{ (from 21° to 50° after B.D.C.)}\times349$$
(for valves).

From Fig. 32 $\int A \cdot dt$ for this interval is $1{\cdot}4 \times 10^{-3}$, so the volume introduced is :—

$$\tfrac{2}{3} \times 1{\cdot}4 \times 10^{-3} \times 349 = 0{\cdot}326 \text{ ft.}^3$$

Adding to this the volume already introduced, viz. 0·73 ft.³, we obtain 1·056 ft.³, which is a trifle in excess of the required quantity.

The value assumed for the scavenge pressure, viz. 17 lb./in.² abs., therefore, seems to be not far out.

The precise value of P_1 is a matter of considerable importance, as on it depends the value of the charge weight, and, consequently, the power capacity of the cylinder. Its predetermination, however, appears to be a difficult matter, and in practice it is usually adjusted experimentally by advancing or retarding the scavenge cam, according as the value of P_1 found by light spring indicator diagram is too low or too high. In the former case an adjustment is easily effected by putting a baffling diaphragm in the exhaust pipe.

The above calculations, however, are a sufficient check on the design to ensure the provision of suitable valve or port areas.

The volume (@ 15 lb./in.² abs.) of scaveng air lost through the exhaust ports is roughly equal to :—

$$\tfrac{1}{2} \int A \cdot dt \text{ (from } 21^\circ \text{ to } 50^\circ \text{ after B.D.C.)} \times 498 \times 0{\cdot}65$$

(for ports).

From Fig. 32 $\int A \cdot dt$ for this interval is $2{\cdot}15 \times 10^{-3}$ ft.² sec., so the volume required is :—

$$\tfrac{1}{2} \times 2{\cdot}15 \times 10^{-3} \times 498 \times 0{\cdot}65 = 0{\cdot}35 \text{ ft.}^3$$

So that the air charge is equivalent to $(1{\cdot}02 - 0{\cdot}35) = 0{\cdot}67$ ft.³ @ 15 lb. per sq. in. abs.

Its actual volume at the point when the piston covers the ports is $0{\cdot}05 + (0{\cdot}85 \times 0{\cdot}68) = 0{\cdot}63$ ft.³

and its pressure is therefore about

$$\frac{15 \times 0{\cdot}67}{0{\cdot}63} = 16{\cdot}2 \text{ lb./in.}^2$$

This figure being 0·8 lb./in.² below the assumed scavenge pressure of 17 lb./in.² indicates that the latter figure is sufficient for the supply of the requisite quantity of air under the conditions specified. P_1 may be expected to have a probable value of about 16·5 lb./in.²

Exhaust of Two Stroke Engines.—As already mentioned, it is essential that the scavenge receiver should not be put into communication with the cylinder until the contents of the latter have fallen to a pressure almost equal to that of the scavenge air, by the release of the products of combustion, through the exhaust slots. This process usually takes a period of time equivalent to 20 to 30 degrees of revolution of the crank-shaft. The calculation of this period is much facilitated by the fact that in the interval considered the port area is increasing very approximately in proportion to the time (see Fig. 32). It can easily be shewn that if during an interval from $t=o$ to $t=t_1$ an orifice area increases uniformly with respect to time from O to A_1 and the velocity of efflux also varies uniformly with respect to time from V_0 to V_1, then the discharge is given by :—

$$Q=\frac{A_1T_1}{3}(V_1+\tfrac{1}{2}V_0) \quad \ldots\ldots\ldots\ldots(1)$$

If A represents ft.^{2}, t secs., and V ft./secs. of an incompressible fluid, then Q represents cub. ft.

If, as in the case we are about to consider, V represents lb. per sec. per unit of area, then Q represents lb. We proceed to apply equation (1) to the exhaust period of the two stroke engine specified in the previous article.

Exhaust Calculation.—*Data :* Cylinder pressure at the point at which the ports become uncovered, 60 lb./in.^{2} abs. Temperature at the same point is equal to :—

$$\frac{\text{initial charge temperature}\times 60}{\text{initial charge pressure}}=\text{say}\frac{670\times 60}{18}$$
$$=2240^\circ \text{ F. abs.}$$

Pressure in exhaust pipe, 15 lb./in.^{2} abs.

Problem : To find the position of the crank when the cylinder pressure has fallen to 18 lb./in.^{2} abs.

Assume that the charge has the thermal properties of pure air and that the expansion is adiabatic.

At the instant when the ports first become open the volume of the charge is :—

$$0{\cdot}05+0{\cdot}85\times 0{\cdot}68=0{\cdot}63 \text{ ft.}^3$$

and the weight of the charge is :—

$$(60\times 144\times 0{\cdot}63)\div(53{\cdot}2\times 2240)=0{\cdot}0456 \text{ lb.}$$

When the cylinder pressure has fallen to 18 lb./in.^{2} abs. the volume is not certainly known, but will not differ much from $0{\cdot}05+0{\cdot}92\times0{\cdot}68=0{\cdot}675$ ft.^{3}, and its temperature will be :—

$$2240\left(\frac{18}{60}\right)^{0\cdot285}=1590^\circ \text{ F. abs.} \checkmark$$

and the charge weight is reduced to

$$(18\times144\times0{\cdot}675)\div(53{\cdot}2\times1590)=0{\cdot}0207 \text{ lb.}$$

The weight discharged in the interval is therefore :—

$$0{\cdot}0456-0{\cdot}0207=0{\cdot}0249 \text{ lb.}$$

The next step is to calculate the rate of discharge per unit of port area.

At the higher pressure of 60 lb./in.^{2} the throat pressure is $0{\cdot}53\times60=31{\cdot}8$ lb./in.^{2} abs., and the throat temperature is :—

$$2240\left(\frac{31{\cdot}8}{60{\cdot}0}\right)^{0\cdot285}=1870^\circ \text{ F. abs.}$$

The calculated throat velocity is therefore :—

$$109{\cdot}5\sqrt{2240-1870}=2110 \text{ ft./sec.}$$

and the apparent velocity $=2100\times0{\cdot}65=1365$ ft./sec.

Now the specific volume at the throat is :—

$$\frac{53{\cdot}2\times1870}{31{\cdot}8\times144}=21{\cdot}8 \text{ ft.}^3 \text{ per lb.}$$

The rate of discharge is therefore :—

$$\frac{1365}{21{\cdot}8}=62{\cdot}7 \text{ lb. per sec. per ft.}^2 \text{ of port area.}$$

At the lower pressure of 18 lb./in.^{2} the throat pressure is 15 lb./in.^{2} abs., and the throat temperature is :—

$$2240\left(\frac{15}{60}\right)^{0\cdot285}=1510^\circ \text{ F. abs.}$$

The calculated throat velocity is therefore :—

$$109{\cdot}5\sqrt{1590-1510}=978 \text{ ft./sec.}$$

and the apparent velocity $=978\times0{\cdot}65=635$ ft./sec.

Now the specific volume at the throat is :—

$$\frac{53{\cdot}2\times1510}{15\times144}=37{\cdot}2 \text{ ft.}^3 \text{ per lb.}$$

and the rate of discharge is therefore :—

$$\frac{635}{37{\cdot}2}=17{\cdot}1 \text{ lb./sec. per ft.}^2 \text{ of port area.}$$

By ① $A_1t_1=3\times0{\cdot}0249\div(17{\cdot}1+31{\cdot}3)=1{\cdot}54\times10^{-3}$,

$\int A.\,dt.=\frac{1}{2}A_1t_1$ (approx.) $=0{\cdot}77\times10^{-3}$ ft.^{2} sec.

Alternative Method of Calculation.

Let :—

t=time in seconds, counting from the instant when the ports begin to be uncovered by the piston.

P=pressure of cylinder contents in lb./ft.^{2} during the period considered, so that P varies from 60×144 to 18×144.

T=temperature of cylinder contents deg. F. abs. (variable).

w=weight in lb. of cylinder contents (variable).

A=port area in ft.^{2} (variable).

v=volume in ft.^{3} of cylinder contents (actually varies during the period considered, but treated as constant) $=0{\cdot}65$ ft.^{3}

v_s=specific volume in ft.^{3}/lb. at the throat of the stream issuing from the ports.

V=throat velocity in ft./sec.

$f(P)=\dfrac{dP}{dt}\div A$=the rate of pressure drop per unit of port area,

then $\dfrac{dP}{dt}=f(P)\times A$

and $\int A{\cdot}dt=\int\dfrac{dP}{f(P)}$

$f(P)$ has now to be calculated.

$$P.v=53{\cdot}2\ wT \qquad \therefore \quad P=\frac{53{\cdot}2}{v}w.T.$$

$$\text{and } \frac{dP}{dt}=\frac{53{\cdot}2}{v}\left(T.\frac{dw}{dt}+w.\frac{dT}{dt}\right)$$
$$=\frac{53{\cdot}2}{v}\left(T.\frac{dw}{dt}+w.\frac{dT}{dP}.\frac{dP}{dt}\right)$$

$$\text{so that } \frac{dP}{dt}=\frac{\dfrac{53{\cdot}2}{v}\left(T.\dfrac{dw}{dt}\right)}{\left(1-\dfrac{53{\cdot}2}{v}.w.\dfrac{dT}{dP}\right)}=\frac{\dfrac{53{\cdot}2}{v}.T.\dfrac{dw}{dt}}{\left(1-\dfrac{P}{T}.\dfrac{dT}{dP}\right)}$$

$$\text{Now } \frac{dw}{dt}=A.\frac{V}{v_s}$$

and since $T=T_0\left(\dfrac{P}{P_0}\right)^{0{\cdot}285}$, where T_0 and P_0 are the initial values of T and P,

$$\frac{dT}{dP}=\frac{0{\cdot}285\ T_0P^{(0{\cdot}285-1)}}{P_0^{0{\cdot}285}}$$

ıd $\frac{P}{T}\cdot\frac{dT}{dP}=0{\cdot}285$ (i.e. constant),

$$\text{so that } f(P)=\frac{dP}{dt}\div A=\frac{\frac{53{\cdot}2}{v}.T.\frac{V}{v_s}}{0{\cdot}715}$$

Values of f(P) are easily calculated for evenly spaced values of P and the integration of $\int\frac{dP}{f(P)}$ can be effected by Simpson's rule.

The table on page 112 shews the application of this method to the example of the preceding article.

$P_0=60$ lb./in.^{2} abs. $T_0=2240°$ F. abs. $V=0{\cdot}65$ ft.^{3}

For convenience P has been expressed in lb./in.^{2}, instead of lb./ft.^{2}, so that a factor of 144 is required. Since the interval between successive values of P is 10·5 lb./in.^{2}, we have :—

$$\int A{\cdot}dt=\frac{dP}{f(P)}=\frac{10{\cdot}5\times144\times96{\cdot}18\times10^{-6}}{3}$$
$$=0{\cdot}485\times10^{-3}$$

Dividing by the discharge coefficient 0·65 the required value of $\int A{\cdot}dt$ is :—

$$\frac{0{\cdot}485\times10^{-3}}{0{\cdot}65}=0{\cdot}74 \text{ ft.}^2\text{ sec.}\div1000$$

which agrees with the value obtained previously within about 4%.

The value of the discharge coefficient (0·65) has been found by comparing the results of calculations similar to the above with the information afforded by light spring indicator cards. It is also in good agreement with experiments on the flow of gases through sharp-edged orifices. It is by no means easy, however, to obtain light spring indicator diagrams which are at all reliable with an ordinary indicator. The fall of pressure is so rapid that the shape of the card is distorted by very little indicator stickiness, and oscillations are almost inevitable.

With care the latter may be approximately allowed for, but cards shewing appreciable indicator stickiness must be rejected. These troubles may be largely eliminated by using an optical indicator, which is perhaps the only instrument well adapted to this class of investigation.

1	2	3	4	5	6	7	8	9	10	11	12
P lb./in.² abs.	T deg. F. abs.	Throat Press. lb./in.² abs.	Throat Temp. deg. F. abs.	Temp. drop, deg. F.	Throat velocity, V ft./sec.	v_s	$\frac{V}{v_s}$	$\frac{53{\cdot}2\ T{\cdot}V}{v{\cdot}v_s}$	$\frac{1}{f(P)} = \frac{0{\cdot}715}{[9]}$	Simpson's figure.	(10)×(11).
60·0	2240	31·7	1870	370	2110	21·8	96·8	$1{\cdot}78\times10^{7}$	$4{\cdot}02\times10^{-8}$	1	$4{\cdot}02\times10^{-8}$
49·5	2120	26·2	1770	350	2050	25·1	81·7	$1{\cdot}42\times10^{7}$	$5{\cdot}04\times10^{-8}$	4	$20{\cdot}16\times10^{-8}$
39·0	1980	20·6	1655	325	1975	29·8	66·4	$1{\cdot}08\times10^{7}$	$6{\cdot}62\times10^{-8}$	2	$13{\cdot}24\times10^{-8}$
28·5	1810	15·0	1510	300	1900	37·2	51·1	$0{\cdot}76\times10^{7}$	$9{\cdot}44\times10^{-8}$	4	$37{\cdot}76\times10^{-8}$
18	1590	15·0	1510	80	980	37·2	26·3	$0{\cdot}34\times10^{7}$	$21{\cdot}00\times10^{-8}$	1	$21{\cdot}00\times10^{-8}$
											$96{\cdot}18\times10^{-8}$

Fig. 33 shews a light spring diagram taken from a two stroke Diesel Engine with an ordinary indicator in good order.

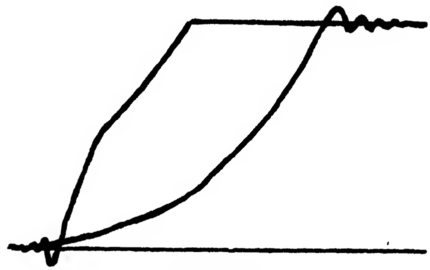


FIG. 33.

Port Scavenging.—If the 10″×15″ engine referred to in the preceding articles is required to run at 600 R.P.M. instead of 300 R.P.M., it can easily be seen from Fig. 32 that the crank angle required for the cylinder pressure to drop to 18 lb./in.^{2} is increased from 18° to about 25°. This is not a serious matter; but with the scavenge valves specified the scavenge pressure would rise to about 12 lb./in.^{2}, thus throwing a heavy load on the scavenge pump and seriously reducing the mechanical efficiency. By fitting four valves instead of two the pressure could be greatly reduced, but it would not be easy to find space for four valves of the size given.

The conditions of the problem are better met by releasing the exhaust through the valves and admitting scavenge air through the ports. In this case the ports would have a height equal to about 10% of the stroke, opening and closing about 40° before and after bottom dead centre. The exhaust valve would begin to open about 70° before bottom dead centre, giving 30° for exhaust evacuation before the scavenge ports begin to be uncovered. The total value of $\int A{\cdot}dt$ for the ports (@600 R.P.M.)=2·4 and the theoretical velocity is therefore,

$$1{\cdot}02\times10^3\div2{\cdot}4=430 \text{ ft./sec.}$$

If the ports are streamlined the coefficient of discharge will be high, say 0·9, giving a throat velocity=480 ft./sec. corresponding to a pressure difference of about 2·0 lb./in.^{2}. If the exhaust valve and the scavenge ports close simultaneously pressure will build up during the scavenge period leaving the cylinder slightly supercharged. A probable scavenge pressure is 5 to 6 lb./in.^{2} above atmospheric but the exact value is difficult to determine accurately by calculation as it is influenced by ejector action of the exhaust in passing through the valve and also possibly by resonance effects in the exhaust pipe.

The two methods so far described are examples of uniflow scavenge. Other examples are the opposed piston arrangement in which one piston controls the exhaust ports and the other the scavenge ports, and a sleeve valve control with scavenge and exhaust ports at opposite ends of the cylinder. Well designed uniflow scavenge engines have the advantage of practically complete evacuation of the exhaust products and are therefore able to develop high mean pressures.

Loop Scavenge.—This term is applied to scavenge systems in which exhaust is released and air admitted through ports arranged at one end of the cylinder (see p. 12). If the ports are uncontrolled by valves of any kind the exhaust ports must be uncovered before the air ports and are consequently covered again on the return stroke after the air ports. There is, therefore, a relatively great loss of effective stroke, and no opportunity for supercharging unless the exhaust is throttled in some way. Furthermore, the proximity of air and exhaust ports and the looped shape of the stream lines connecting them lead to unavoidable mixing of air and exhaust products.

The charging efficiency of a two stroke cylinder has been worked out by the late Prof. Bertram Hopkinson on the assumption of complete mixing at each stage with the following results :

Volume of air introduced ÷volume of cylinder . .	1·2	1·4	1·6	1·8
% of cylinder volume filled with pure air	70	75	80	84

With carefully developed designs the assumption of complete mixing no doubt errs on the side of pessimism and better results than the above are obtainable.

The charging efficiency with loop scavenge can be improved by valve control of the inlet ports or the exhaust ports or both. By this means the compression is enabled to begin earlier and supercharging is possible.

The following table shews approximately the relative capacities of three systems assuming an excess air ratio of 1·4 and a scavenge pressure of 2 lb./in.^{2} in each case.

		Uniflow Scavenge.	Plain Loop Scavenge.	Loop Scavenge with controlled air ports.
1	Compression stroke % of full stroke	90	76	73
2	Combustion space % of stroke volume	8	8	8
3	1+2	98	84	81
4	Scavenge efficiency	0·95	0·85	0·85
5	Pressure at begin. of compression in atmospheres	1·05	1·0	1·14
6	3×4×5 figure of merit	98	72	79

The last line (6) shews the relative capability of producing indicated mean pressure under the assumed conditions. The figure of merit for the uniflow engine assumes that the air and exhaust ports or valves close simultaneously on the upstroke. If the exhaust remains open appreciably after the inlet the figure of merit is reduced by about 5% to 10%. The figures for loop scavenge assume a free exhaust. They can be increased by throttling the exhaust by timed valves. In all cases the pressure at the beginning of compression may be subject to effects of surging in the exhaust pipe (exhaust resonance) the result of which may be favourable or unfavourable.

Exhaust Interference and Resonance.—Since the process of exhaust and scavenge occupies about 120° of crank rotation, the exhaust of one cylinder is liable to interfere with the scavenge of another if more than three cylinders exhaust into the same manifold. The effect of this interference may be minimised by using a manifold of large volume and/or by shielding the exhaust outlet of each cylinder from the effect of the flow caused by the others. By suitable design of exhaust pipes and expansion chambers, the effects of interference may be used to increase charging efficiency. The effects aimed at are as follows:

(a) Minimum pressure (subatmospheric) in the exhaust manifold during the middle period of the scavenge of each cylinder to facilitate entry of air into the cylinder and evacuation of exhaust.

(b) High pressure (above atmospheric) in the exhaust manifold at the point of closing of the air or exhaust ports (whichever closes latest).

Consider for simplicity a single cylinder engine with exhaust opening 70° before B.D.C.; the rush of exhaust gas through the ports initiates a pressure wave which travels along the exhaust pipe with a speed of about 1500 ft./sec. On reaching the expansion chamber the wave is reflected as a pulse of negative pressure with the same speed. The condition for this wave of rarefaction to reach the cylinder 20° before B.D.C. is as follows:—

$$\frac{(70^\circ-20^\circ)\times 60\times 1500}{360\times n}=2\text{ L}.$$

where L=length of pipe.
n=R.P.M.,

from which $L=\dfrac{6250}{n}$ ft. . . . .(1)

After a further period of 50° of crank-shaft rotation the wave returns once more as a pulse of positive pressure beginning 30° after B.D.C. and reaching a maximum value 30°+25=55° after B.D.C., coinciding fairly well with the beginning of compression. Naturally the precise constant in equation (1) above is to be found by experiment. If there are three cylinders exhausting into the same pipe the return of the positive pressure wave may be made to coincide with the beginning of the pressure wave due to the release of exhaust from a neighbouring cylinder. Similar results are obtainable with four and five cylinders with a shorter duration of the negative pressure period. Larger numbers of cylinders can be dealt with on the same lines by using two or more manifolds. Arrangements of this kind have been patented by Burmeister & Wain and extensively used on the S.A. and D.A. B. & W. type two stroke engines.

Similar principles appear to be involved in the system described by Mr. Petter, whereby a two stroke cylinder is charged without the use of a scavenge pump of any kind.

Optimum Port Height.—If the scavenge ports have insufficient height an undue loss is incurred in the scavenge pump. If too high, an excessive fraction of the effective stroke is lost. Let h=height of scavenge ports expressed as a fraction of the stroke. The loss of potential indicated mean pressure due to shortening of the compression stroke is about 100 h lb./in.^{2}. If p is the drop of pressure through the scavenge ports the lost mean pressure due to this is about 2 p. Now the $\int A\cdot dt$ for the scavenge ports varies approximately as h^2 and the scavenge pressure varies inversely as the square of $\int A\cdot dt$, hence :—

$$p=\frac{c}{h^4}$$

where c varies as the square of the revolutions.

Regarding h as variable the brake mean pressure will be a maximum when,

$$\frac{d}{dh}\left(2\,p\right)=\frac{d}{dh}\left(100h\right)$$

$$\text{i.e. } \frac{d}{dh}\left(\frac{2C}{h^4}\right)=100$$

$$\frac{8C}{h^5}=100$$

$$8p=100\,h$$

$$p=12{\cdot}5\,h$$

For example, if h=0·1 and p=3 lb./in.^{2} the formula indicates that a better result would be obtained by heightening the ports. The total loss of potential mean pressure is :—

$$(0{\cdot}1\times 100)+(2\times 3)=16.$$

Increasing h to 0·12, p drops to $3\div 1{\cdot}2^4=1{\cdot}44$ lb./in.^{2}, which nearly verifies the formula. The total loss is now

$$(0{\cdot}12\times 100)+(2\times 1{\cdot}44)=15$$

shewing a gain of about 1% in brake mean pressure and about 2% in mechanical efficiency.

The pressure drop p through the scavenge ports is not to be confused with the total scavenge pressure which may be 1½ to 3 times as much depending on the areas of exhaust ports or valves.

Air Resistance in Pipework.—Induction and exhaust systems should be checked for resistance since excessive resistance is prejudicial to performance of both four and two stroke engines. In a four stroke engine the resistances of induction and exhaust systems should preferably not exceed 6″ and 15″ of water respectively. With exhaust turbo blowers the resistance of the exhaust systems beyond the blower should be even lower, say 8″ or less.

A simple way of dealing with exhaust resistances is to work out the velocities and resistances as for air @ 60° F. and then increase the resistances so found in the ratio,

$$\frac{\text{Absolute temperature of gas (F. scale).}}{521}$$

The velocity head varies as the $(\text{velocity})^2$ and for 100 ft./sec.$=2{\cdot}28$ in. of water for air @ 60° F.

Resistance roughly equal to the velocity head is encountered at every sharp right angled turn, or sudden enlargement of path, and in each 30 diameters of straight circular pipe. In estimating velocities at points of constriction suitable contraction coefficients must be used, e.g. 0·6 for a sharp-edged orifice, 0·8 for a passage with a sharp obstruction one side and a wall parallel to the flow on the other, 0·98 for a good bell mouth orifice, and so forth.

Literature.—Funck, G., " Two Stroke Engines with special reference to the Design and Calculation of Ports."—*Automobile Engineering*, May, 1918, *et seq.*

Petter, H., " The Escape of Exhaust Gas in Two Stroke Engines."—*Engineering*, January 4th, 1918.

Thompson Clarke Research, " Air Flow through Poppet Valves."—Report to U.S. Nat. Advisory Committee for Aeronautics. See *Engineering*, January 10th, 1919.

Petter, P. W., " Induced Air Scavenge for Two Stroke Engines."—*Engineer*, August 17th, 1934.

Caunter, C. F., " The Two Cycle Engine."—Pitman.

" The Scavenging of Two Cycle Engines."—*Sulzer Technical Review*, No. 4, 1933.

Curtis, " Improvements in Scavenging and Supercharging Two Cycle Diesel Engines."—*Diesel Power*, September, 1933.

Hopkinson, B., Trans. N.E. Coast Inst. Engs., Vol. XXX, July, 1914.

CHAPTER VII

SUPERCHARGING

Supercharging of Four Stroke Diesel Engines.—The mean indicated pressure which it is prudent to use in Diesel Engines under service conditions is limited by considerations of:

(1) The limiting mean pressure at which practically perfect combustion is obtainable. Poor combustion means not only poor economy but also results in stuck piston rings, defective lubrication of liners, and other evils. Consequently a substantial margin should be allowed between the limiting mean pressure referred to above and the actual mean pressure used in service.

(2) Temperature stresses.

(3) Overheating of liner surfaces, valves, etc. By supplying air to the cylinder at a pressure above atmospheric, the weight of air retained in the cylinder at each cycle may be augmented, and consequently the working mean pressure may be increased without encroaching on the margin of safety referred to under (1).

Consider a normal four stroke Diesel Engine working at a M.I.P. of 100 lb./in.^{2} for example. If the inlet valves be connected to an air receiver maintained by a blower at a pressure of 1·2 atmospheres absolute, and if the exhaust valves be connected to a receiver with an outlet throttled so as to maintain a back pressure of 1·2 atmospheres absolute, then the charge weight and compression pressure will both be increased 20% as compared with the normal engine. Also, if 20% more fuel be injected per cycle, the mean pressure will be 20% above normal, i.e. 120 lb./in.^{2}, and all the pressures of the cycle will be 20% above normal. Since, however, the proportion of fuel to air is normal, all temperatures throughout the cycle will be the same as in a normal engine.

It does not follow that the wall temperatures and temperature stresses will be normal. On the contrary, they will in general be higher unless a surplus of air is passed through for cooling purposes.) Heat flow on account of gas radiation increases with density according to the power 0·5 approximately. Heat flow due to convection increases with density according to a power 0·5 to 0·8 according to circumstances. The two effects together may be taken to vary with density according to the power 0·6 about. The effect of supercharging 20% may therefore be expected to increase heat flow to jackets about 12% and temperature stresses by the same amount. Since the percentage increase of heat flow (as compared with a normal engine) is less than the percentage increase of power, the efficiency will be higher if the power required to drive the blower be disregarded. The latter consideration is not irrelevant as the exhaust gases in our hypothetical example being 20% above atmospheric in pressure, may be made to drive an exhaust turbine and the power obtainable in this way is theoretically much in excess of that required to drive the blower. A simple calculation shews that the turbine can drive the blower if the combined efficiency of turbine and blower exceeds about 45%. It is sufficient therefore if each unit has an efficiency of 67% approximately.

The system outlined above was first applied by Rateau to Aircraft Petrol Engines, and a similar scheme has been applied to Diesel Engines, notably by Büchi. The following calculations illustrate the principles.

Four Stroke Diesel Engines.

20% Supercharging by Turbo blower.

Normal engine. Heat equivalent of.				Supercharged engine.
B.H.P.	33%	33+20%=39·½	÷ 118 =	33½%
Jacket heat	25%	25+12%=28·0		24%
Exhaust, etc. Radiation	42%	42+20%=50·½		42½%
Total	100%	118·0		100%

If the engine is allowed to exhaust to atmosphere direct, instead of through a turbine, there will be a gain of mean indicated pressure of about $0{\cdot}2 \times 14{\cdot}7 = 3$ lb./in.², and the brake thermal efficiency becomes $33\frac{1}{2} \times 103 \div 100 = 34\frac{1}{2}\%$, as compared with 33% for the normal engine, i.e. a gain of $4\frac{1}{2}\%$, taking 33% as unity. Against this must be set the power required to drive the fan or blower electrically or otherwise, and this just about balances the account again. It seems, therefore, that no improvement in overall efficiency is to be expected with supercharging by separately driven fans. If the mean pressure of an engine is limited by considerations of combustion or exhaust temperature, but yet has a margin on the score of temperature stresses, then supercharging is a convenient means of safely augmenting the mean pressure. If temperature stresses are the limiting consideration, then it appears from the above reasoning that 20% supercharging warrants an increase of mean pressure not exceeding about 8% above normal, arrived at as follows :—

Normal Engine.		Supercharged Engine.	
Heat equivalent of			(Same jacket heat as normal engine.)
B.H.P.	. . . .	33 add 20% = 39·5	. 35·5
Jacket heat	. . .	25 add 12% = 28·0	. 25·0

$$(35{\cdot}5 - 33) \div 33 = 7{\cdot}6\%$$

A series of experiments carried out by the M.A.N. support this conclusion. A normal engine developing 80 B.H.P. developed with supercharge of 35% a B.H.P. of 89 for the same flow of jacket heat as the normal engine, i.e. a power increase of 12% for 35% supercharge for equal heat flow. If a large excess of air is passed through the engine the jacket heat can naturally be reduced materially but at the expense of fan power.

In the Büchi system of exhaust turbo-charging the blower supplies a considerable excess of air and the inlet and exhaust valve openings are arranged with a large overlap at the top dead centre. Furthermore the piping between the exhaust valves and the exhaust turbine is arranged in such a way that pulsations of pressure are set up which result in the exhaust back pressure being appreciably lower than the scavenge pressure during the period of overlap. This pressure difference causes a flow of air from the inlet valve into the cylinder and out of the exhaust valve, thus clearing out residual exhaust gases and providing internal cooling just where it is most needed, viz. at the centre of the cylinder cover between the inlet and exhaust valves and at the centre of the piston.

Exhaust Turbine Blower Combination.—Reverting to the 20% supercharged engine with exhaust turbine driven blower, let the exhaust temperature be 750° F., i.e. 750+461=1211° F. abs., then the temperature after adiabatic expansion to atmospheric pressure

$$=1211 \div 1{\cdot}2^{\left(1-\frac{1}{1{\cdot}4}\right)}=1151° \text{ F. abs.}$$

The available heat drop is then

$$(1211-1151)\times 0{\cdot}24=14{\cdot}4 \text{ B.T.U./lb.}$$

If the scavenge air at atmospheric pressure has a temperature of 60° F., i.e. 521° F. abs., then the temperature after adiabatic compression to 1·2 atmospheres :—

$$=521\times 1{\cdot}2^{\left(1-\frac{1}{1{\cdot}4}\right)}=548° \text{ F. abs.}$$

The thermal equivalent of the work required to compress and deliver is then :—

$$(548-521)\times 0{\cdot}24=6{\cdot}5 \text{ B.T.U./lb.}$$

In order that the turbine may be capable of driving the blower, the combined efficiency must, on the above assumptions, be not less than

$$100\left(\frac{6{\cdot}5}{14{\cdot}4}\right)=45\%$$

Actually the exhaust flow exceeds the inlet flow on account of added fuel and blast air (if any), and the specific heat of the exhaust gas is higher than that of the scavenge air. On the other hand, the ratio of specific heats is somewhat less than 1·4 for exhaust gas. The above calculations are easily amended for different values of the constants and for different conditions of pressure and temperature. For example, consider a case of 100% supercharge, and take :—

Exhaust temperature 750° F.=1211° F. abs.
γ=ratio of specific heats=1·3 for exhaust.
Specific heat at constant pressure=0·28 for exhaust.
Exhaust discharge wt.=1·12 inlet wt. on account of fuel and air blast.
Air temperature=60° F.=521° F. abs. γ=1·4.

Then exhaust temperature after adiabatic expansion,

$$=1211\div 2{\cdot}0^{\left(1-\frac{1}{1\cdot 3}\right)}=1034^\circ \text{ F. abs.}$$

The available heat drop,

$$=1{\cdot}12\ (1211-1034)\times 0{\cdot}28=55{\cdot}5 \text{ B.T.U. per lb. of inlet air.}$$

Air temperature after adiabatic compression,

$$=521\times 2{\cdot}0^{\left(1-\frac{1}{1\cdot 4}\right)}=635^\circ \text{ F. abs.}$$

Heat equivalent of work to compress and deliver,

$$=(635-521)\times 0{\cdot}24=27{\cdot}4 \text{ B.T.U./lb.}$$

Minimum efficiency required of turbine blower unit,

$$\frac{27{\cdot}4}{55{\cdot}5}=49\tfrac{1}{2}\%$$

This is satisfied by turbine and blower each having an efficiency of 70%. This is readily obtainable with careful design. Arrangements are possible in which the exhaust pressure is lower than the scavenge pressure. With a given engine running at a fixed speed against a steady load and with fixed adjustments of exhaust turbine and blower, the exhaust temperature and pressure, also the scavenge pressure, adjust themselves to stable values. The designer's problem is so to proportion the turbine and blower that these stable values correspond to conditions of maximum efficiency of the turbine blower unit and give a minimum exhaust temperature.

Four stroke engines supercharged on the Büchi system of exhaust turbo-charging commonly develop a mean indicated pressure of about 125 lb./in.^{2} with a mean scavenge pressure of 5 lb./in.^{2} at full load. Overloads up to 150 lb./in.^{2} can be obtained with very little increase of exhaust temperature. The system works as well (if not better) with airless injection as with air injection.

The most excellent results on this system have been obtained with 6 or 12 cylinder engines. Multiple exhaust manifolds are used, one manifold for every 3 cylinders. The cylinders connected to any one manifold are so chosen that the first rush of exhaust from any one cylinder does not interfere with the scavenge process of any other cylinder connected to the same manifold. Similar arrangements are made for other numbers of cylinders.

Fig. 34 shews results obtained from a large 6 cylinder engine supercharged on the Büchi system. Fig. 35 gives on a time base the fluctuations of pressure in the scavenge and exhaust manifolds.

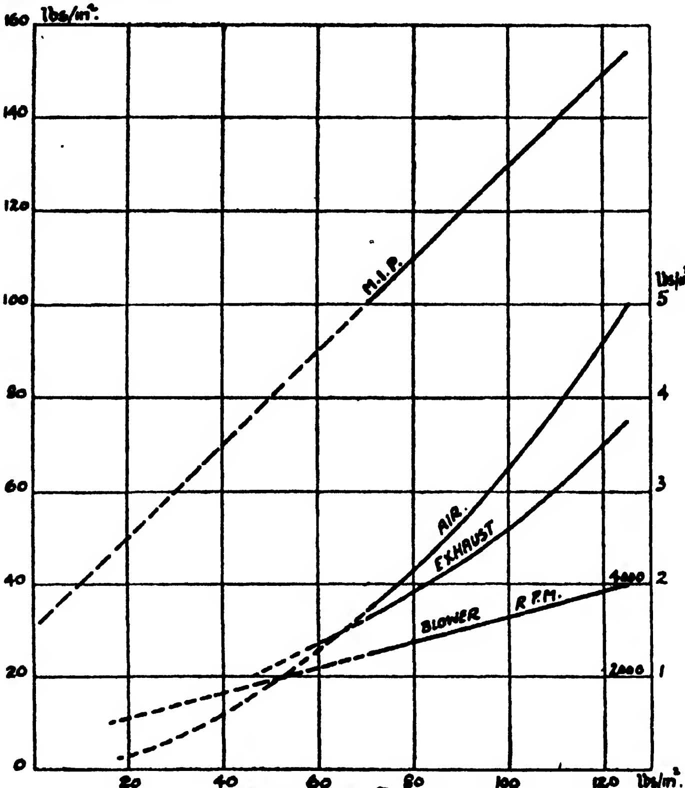


Fig. 34.

Steam Driven Blower.—It has also been proposed to drive the blower by a turbine fed with steam raised in an exhaust heated boiler. Consider an engine exhausting to atmosphere at 750° F. through a boiler at 100 lb./in.² (steam temperature 338° F.) in which the gases fall to 400° F. The heat available for evaporation,

$$=1{\cdot}12\times 0{\cdot}28\ (750-400)=110 \text{ B.T.U. per lb. of inlet air.}$$

Assuming a boiler efficiency of 80% and feed temperature of 100° F., this corresponds to an evaporation of about 0·079 lb. per lb. of inlet air. The adiabatic heat drop per lb. of steam

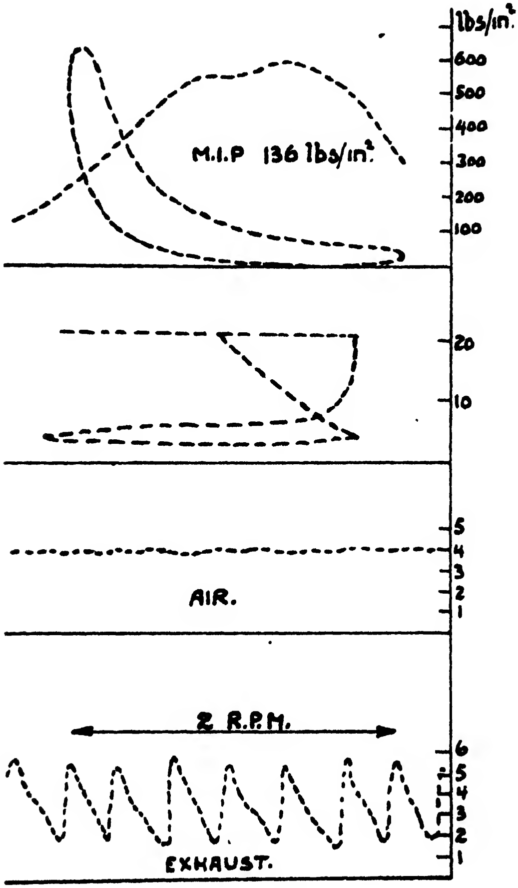


Fig. 35.

from 100 lb. saturated to a vacuum of 26″=270 B.T.U. So the available heat drop per lb. of inlet air is

$$270 \times 0{\cdot}079 = 21{\cdot}4 \text{ B.T.U.}$$

which is sufficient to give a supercharge of about 20% with an overall turbine-with-blower efficiency of 30%.

An advantage of this arrangement is that the exhaust fired boiler could be arranged to give practically no back pressure, so that a substantial gain in efficiency, amounting in this case to about 2%, should accrue on this account.

The preceding calculations assume that the exhaust temperature has approximately "normal" value. If an excess of air is passed through the cylinder for cooling purposes the exhaust temperature is thereby reduced and relatively less heat is recoverable.

A further variation would be the use of a waste heat back pressure exhaust fired boiler discharging through an exhaust turbine.

Engine Driven Blowers.—Several arrangements of engine driven blowers have been successfully used for supercharging four stroke engines, viz. :—

(a) Reciprocating blowers with automatic suction and discharge valves; these blowers are quite similar to the scavenge blowers fitted to many two stroke engines.

(b) Centrifugal blowers connected to the crank-shaft through a chain of gears; this arrangement follows previous practice in connection with petrol aircraft engines. The disadvantages of this system are concerned chiefly with the gearing. It seems to be necessary to provide slipping clutches to avoid high inertia stresses at starting or stopping, and the gearing is liable to be expensive and noisy.

(c) Blowers of the Rotary displacement type driven through gearing at two or three times engine speed; there are numerous types of such blowers, many of them based on the general Roots type long familiar in connection with foundry plant. The blowers themselves require little or no lubrication as the working elements have a fine but definite mechanical clearance. The gearing problem is much less acute than with system (b) on account of the lower gearing ratio (see Fig. 37).

(d) Using the underside of the pistons (crosshead engines) as blower cylinders; this system was first used extensively on Werkspoor engines. The underside of the cylinder is boxed in and provided with automatic inlet and discharge valves.

Since there is a pumping stroke every revolution and an engine induction stroke every second revolution only, the volumetric efficiency of the underside of the piston in its capacity as a pump only needs to be 75% in order to provide a 50% excess of air for supercharging and scavenging. Apart from the indicated power of the pumping action, there are no mechanical losses associated with the system. If the mean scavenge pressure is 5 lb./in.^{2} and the average m.i.p. of the scavenge pumping diagrams is $5\frac{1}{4}$ lb./in.^{2} the nett increase of lost mean pressure due to the pumping and scavenging action is about

$$2 \times 5\tfrac{1}{4} - 5 = 5\cdot5 \text{ lb./in.}^2.$$

It is important to fit large valves and passages to reduce the m.i.p. of the pumping action to a minimum in comparison with the mean scavenge pressure. This remark applies to all engine driven scavenge blowers.

The "Topping-up" System.—Some of the Burmeister & Wain four stroke engines have been fitted with special inlet valves provided with multiport piston valves connected to the valve stems and controlling two sources of air supply, viz. a suction pipe to atmosphere and a connection to a pressure manifold charged by an engine driven blower of rotary type. Towards the end of the exhaust stroke and whilst the exhaust valve is still open, the inlet valve lifts and admits air from the pressure manifold. This air clears out the residual exhaust gases and cools the centre of the piston and cover. Some degrees after top dead centre the exhaust valve closes, and the further opening of the inlet valve cuts off communication with the scavenge manifold and establishes communication with the atmosphere. During most of the induction stroke air is drawn into the cylinder in the usual way. In the neighbourhood of bottom dead centre, as the inlet valve approaches its seat once more, the process is reversed and the cylinder is put into communication with the scavenge manifold, and when the inlet valve finally seats, the cylinder starts compression with a charge at approximately the pressure of the scavenge air. With this arrangement the engine driven blower is only called upon to deliver about one-half of the total quantity of air which passes into the cylinder. On the other hand, the negative loop of the cylinder indicator diagram is not appreciably reduced by the scavenge pressure. If the blower efficiency is 70% and the scavenge pressure 5 lb./in.^{2} the lost mean pressure due to the pumping action is about $0\cdot75 \times 4\cdot4 \div 0\cdot7 = 4\cdot7$ lb./in.^{2}.

Comparison of Systems.—Assuming in all cases a scavenge pressure of 5 lb./in.^{2}, an air flow of 1·5 times the swept cylinder volume, and a blower efficiency of 70% for direct driven blowers, an indicated mean pressure of 120 lb./in.^{2} and a lost mean pressure of 16 lb./in.^{2} (airless injection engines) covering losses not associated with supercharging, the relative mechanical efficiencies work out as follows :—

System.	Loss due to blower lb./in.	Gain lb./in.^{2} due to excess of scavenge pressure over exhaust.	Nett loss. — or gain +	Total lost mean pressure lb./in.	Mechanical efficiency.
Direct driven blower delivering full quantity of air, including as special case the use of under sides of pistons as scavenge blowers	9·4	5·0	—4·4	20·4	·83
"Topping up" system blower delivery half full quantity of air . .	4·7	—	—4·7	20·7	·83
Büchi exhaust turbo charging system . .	—	1·0	+1·0	15·0	·875

Efficiency of Reciprocating Blowers.—The efficiencies of a blower are required to be known in order to determine the size for a given duty and the power absorbed.

The mechanical friction losses are small and difficult to determine accurately ; they are probably covered by a lost mean pressure referred to the main engine cylinder of 1 lb./in.^{2} or less.

The remainder of the article will be concerned with the volumetric efficiency and the indicated efficiency as defined under.

Let,

X = Excess air coefficient, that is, the ratio of the volume of free air at atmospheric temperature actually delivered per cycle to the volume swept by the main engine pistons per cycle (two stroke engines) or half this volume (four stroke engines).

p_s = Scavenge pressure in receiver.

p_t = Theoretical mean pressure of blower diagram.

p_a = Actual mean pressure of blower diagram.

p_L = " Lost mean pressure " due to blower, referred to the main engine pistons.

η_v = Volumetric efficiency of blower, i.e. the ratio of volume of free air at atmospheric temperature actually delivered to the swept volume of the blower.

η_i = Indicated efficiency of blower, i.e. the ratio of the theoretical indicated work to the actual indicated work required to deliver a given volume of free air.

Then,

$$p_L = Xp_t \div \eta_i = Xp_a \div \eta_v \quad \ldots\ldots\ldots\ldots \quad (1)$$

$$\frac{p_t}{\eta_i} = \frac{p_a}{\eta_v} \quad \ldots\ldots\ldots\ldots \quad (2)$$

$$\frac{\eta_i}{\eta_v} = \frac{p_t}{p_a} = \frac{p_t}{p_s} \cdot \frac{p_s}{p_a} \quad \ldots\ldots\ldots\ldots \quad (3)$$

Values of p_s, p_t and p_s/p_t are tabulated under :—

p_s Scavenge pressure lb./in.²	2	3	4	5	6	7	8	9	10
p_t Theoretical mean pressure lb./in²	1·90	2·77	3·60	4·38	5·10	5·76	6·42	7·10	7·55
$p_s \div p_t$	1·05	1·08	1·11	1·14	1·18	1·22	1·25	1·28	1·32

The amount by which the volumetric efficiency η_v differs from unity depends primarily on the delivery pressure and the percentage clearance volume ; also upon the piston speed and the ratio of valve area to piston area. It is also affected by the heating effect of the cylinder walls (which depends on the delivery pressure), and leakage past the piston.

The ratio p_t/p_s depends chiefly on the resistance to flow, or windage, through the valves. The difference $p_s - p_t$ varies approximately as square of the air speed through the valves plus a constant due to the weight and stickiness of the valves. For any given compressor or any series of geometrically similar compressors the difference $p_s - p_t$ varies approximately as the piston speed (V) squared. Accordingly p_t/p_s and therefore by (3) η_i/η_v are functions mainly of $V/\sqrt{p_t}$, which we therefore adopt as the most convenient criterion.

We may illustrate this point with reference to data obtained from the blower of a two stroke engine as follows :—

$$\left.\begin{array}{lr} \text{Diameter} \quad . \quad . & 62'' \\ \text{Stroke} \quad . \quad . & 41'' \\ \text{Max. R.P.M. about} & 100 \end{array}\right\} \text{of blower.}$$

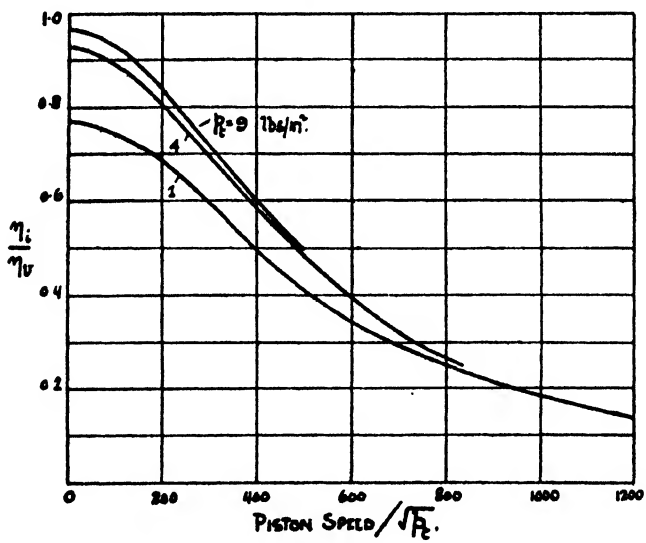


FIG. 36.

At 100 R.P.M. the mean indicated pressure p_s was 4·3 lb./in.² when pumping against a delivery pressure p_s of 2·25 lb./in.² The corresponding value of p_t=2·12 lb./in., so that $p_t/p_s = \eta_i/\eta_v$=0·49. The volumetric efficiency η_v as deduced in various ways was high, probably about 0·95; adopting this value η_i becomes 0·46.

The values of p_a corresponding to various values of p_s and the revolutions are approximately consistent with,

$$p_a = p_t + 0{\cdot}3 + 1{\cdot}9\ (\text{R.P.M.}/100)^2$$
$$= p_t + 0{\cdot}3 + 1{\cdot}9\ (\text{piston speed}/685)^2$$

$$\text{From which } \frac{p_t}{p_a} = \cfrac{1}{1 + \cfrac{0{\cdot}3 + 1{\cdot}9\ (\text{piston speed}/685)^2}{p_t}} = \frac{\eta_i}{\eta_v}$$

The adjoining diagram, Fig. 36, shows η_i/η_v plotted against piston speed (ft./min.) $\div \sqrt{p_t}$, for three values of p_t. Apart from the constant term 0·3 the three curves would coincide thus justifying the choice of the criterion piston speed $\div \sqrt{p_t}$. For any given value of this criterion η_i/η_v increases with p_t, but since η_v decreases with p_t, η_i does not vary so much as η_i/η_v and remains nearly constant.

When a four stroke or two stroke engine is charged by a direct driven blower the air pressure varies nearly as the (R.P.M.)^2 so that piston speed $\div \sqrt{p_t}$ remains nearly constant and the blower efficiency does not change greatly with R.P.M.

The slope of the curves shown in Fig. 36 would be reduced if the ratio of valve area to piston area were increased.

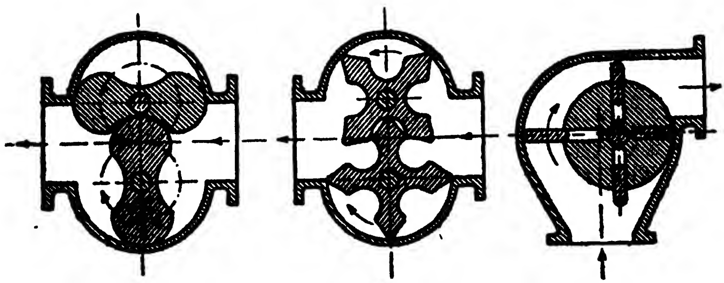


FIG. 37.

If the blower referred to above were used to supercharge a four stroke engine running at 100 R.P.M. with a scavenge pressure of 5 lb./in.^{2} and an excess air coefficient $X = 1{\cdot}5$, the value of p_t would be 4·38 lb./in.^{2} and the piston speed $\div \sqrt{p_t} = 685 \div \sqrt{4{\cdot}38} = 328$. From Fig. 36, $\eta_i/\eta_v = 0{\cdot}64$. Taking $\eta_v =$

0·9, η_l becomes 0·58 and the lost mean pressure from (1) becomes

$$4{\cdot}38\times1{\cdot}5\div0{\cdot}58=11{\cdot}3 \text{ lb./in.}^2$$

(Compare with table, p. 128.)

Efficiency of Positive Rotary Blowers.—In most types of positive rotary blowers including the well-known Roots type, see Fig. 37, a certain volume C_1 is carried round in each revolution from suction to discharge and a smaller volume C_2 from discharge to suction. The quantity (C_1-C_2) may be regarded as the swept volume of the blower. Neglecting all leakage or other losses, the theoretical volumetric efficiency η_v is then,

$$\eta_{vt}=\frac{C_1-EC_2}{C_1-C_2}$$

where EC_2 is the volume of C_2 after adiabatic expansion from the discharge pressure p_s to atmospheric pressure. On this theoretical basis the annexed table gives values of η_{vt} corresponding to various delivery pressures and values of C_2/C_1.

Delivery Pressure p_s lb./in.^{2}	2	4	6	8	10
E	1·102	1·203	1·300	1·398	1·489
η_{vt} for $\frac{C_2}{C_1}=$ ·04	·996	·992	·988	·984	·980
·08	·992	·984	·976	·968	·960
·12	·988	·976	·964	·952	·940

The actual volumetric efficiency η_v is less than this on account of leakage which varies approximately as the square root of the discharge pressure and inversely as the revolutions. To make the results apply to all geometrically similar blowers (with clearances proportional to linear dimensions) we may use the peripheral speed V in place of the R.P.M. We then have,

$$\eta_v=\eta_{vt}-\frac{k\sqrt{p_s}}{V}\quad\text{.................(1)}$$

For any given value of p_s there will be a value of V which makes the discharge and therefore $\eta_v=0$. Let p_o and V_o be such values. Then from (1),

$$k=\eta_{vt}\frac{V_o}{\sqrt{p_o}}$$

and

$$\eta_v=\eta_{vt}\left\{1-\frac{V_o/\sqrt{p_o}}{V/\sqrt{p_s}}\right\}\quad\text{...............(2)}$$

Neglecting friction and windage the work done per revolution is $p_s (C_1-C_2)$ as against the theoretical work $\eta_v p_t (C_1-C_2)$; the "indicated" efficiency is therefore,

$$\eta_i = \eta_v \frac{p_t}{p_s} \quad \dots\dots\dots\dots\dots\dots\dots (3)$$

To obtain the overall blower efficiency η_b we may introduce a term p_w to represent the mean pressure referred to (C_1-C_2) due to friction and windage; then,

$$\eta_b = \eta_v \left(\frac{p_t}{p_s + p_w}\right) \quad \dots\dots\dots\dots\dots\dots (4)$$

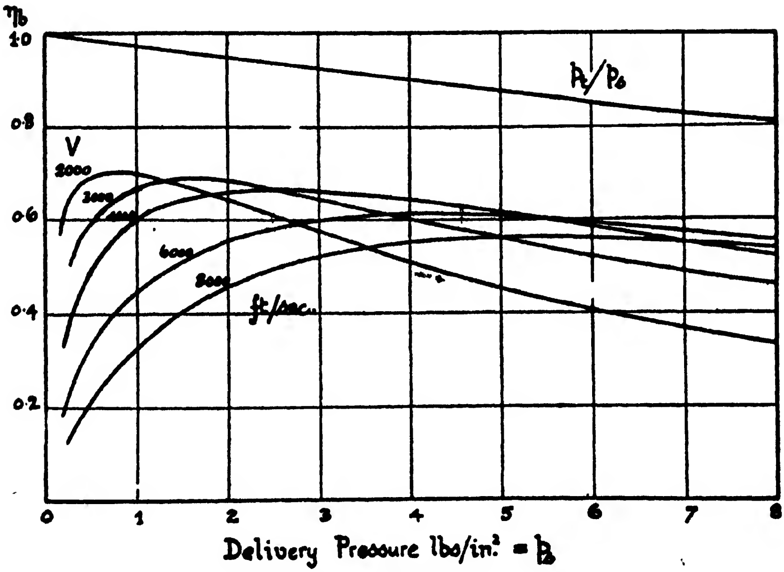


Fig. 38.

Fig. 38 shews the variation of η_b with p_s and V on the following assumptions, viz. :—

$$\frac{C_2}{C_1} = 0{\cdot}12,\ V_o/\sqrt{p_o} = 400,\ p_w = \left(\frac{V}{6000}\right)^2$$

the complete formula for η_b being,

$$\eta_b = \eta_{vt}\left\{1 - \frac{400}{V/\sqrt{p_s}}\right\}\left\{\frac{p_t}{p_s + \left(\frac{V}{6000}\right)^2}\right\} \quad \dots\dots\dots\dots (5)$$

Tests of actual blowers give results conforming to the general characteristics of the above diagram, but the actual efficiencies depend on the friction and leakage characteristics. Efficiencies as high as 80% have been recorded from quite small blowers when delivering against a pressure difference of 5 lb./in.²

Centrifugal Blowers.—Fig. 39 shews an exhaust turbine

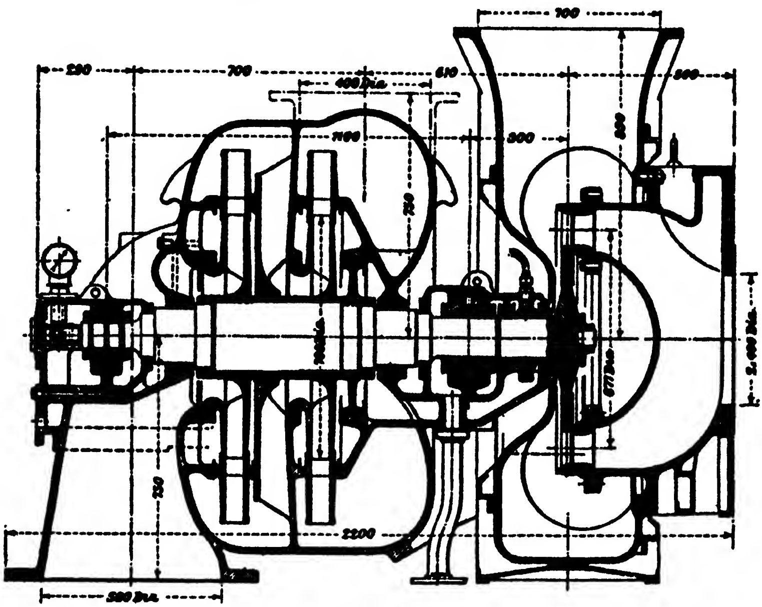


FIG. 39.

driven centrifugal turbo-blower constructed by Brown Boveri for supercharging a four stroke engine on the Büchi system.*

Typical results at full load are as follows :—

Temperature of exhaust before turbine .	867° F.
,, ,, after turbine .	803° F.
Pressure (abs.) of exhaust before turbine .	18·2 lb./in.²
,, , ,, after turbine .	14·7 ,,
,, ,, air before blower . .	14·7 ,,
,, ,, ,, after blower . .	19·4 ,,
Temperature of air before blower . .	57° F. ,,
,, ,, after blower . .	121° F. ,,

* Later Buchi Turbochargers have been fitted with single stage blowers.

It will be noticed that the rise of air pressure, viz. 4·7 lb./in.^{2} exceeds the drop of pressure, viz. 3·5 lb./in.^{2} of the exhaust.

The peripheral speed of the two stage blower wheels required to develop the above air pressure at a stable point of the blower characteristic is about 480 ft./sec.

The velocity of exhaust on leaving the nozzle plate is easily calculated from the adiabatic drop of temperature (see p. 122) and is found to be about 900 ft./sec. assuming a velocity coefficient of 94%. The optimum speed of the turbine wheel at the middle of the blade annulus is about half this, i.e. 450 ft./sec. The nozzles outlets are inclined at a tangential angle of about 15°, and the best angles of the rotating blades are easily determined by the usual vector diagrams for compounding the velocities concerned. The turbine efficiency is about 75%.

The overall efficiency of the combined turbine and blower unit is about 55%.

In a range of similar blowers linear dimensions will vary as the $\sqrt{\text{B.H.P.}}$ of the engine at full load, and revolutions inversely as the $\sqrt{\text{B.H.P.}}$ This gives equal velocities and velocity heads and equal margins against critical speeds of whirling.

Literature.—Blache, H. H., "The Present Position of the Diesel Engine for Marine Purposes."—I.N.A., 1933, deals with supercharging on the "topping up" principle.

Lugt, G. J., "Supercharging Diesel Engines, with special reference to the Werkspoor Type."—I.E. & S. in Scotland, Feb., 1931.

Unsigned, "A New Supercharging Blower" (Power-Plus rotary displacement type).—*Motorship*, Jan., 1930.

Büchi, A., "Turbo-charging of internal combustion engines, especially Diesel Engines."—Inst. Marine Engs., 1928.

Rateau, A., "Use of the Turbo-Compressor for attaining the greatest possible speeds in Aviation."—Inst. Mech. E., 1922.

CHAPTER VIII

CRANK-SHAFTS

Material.—For Marine Diesel Engine crank-shafts the usual material is open hearth steel having a tenacity of 28 to 32 tons per sq. in. and minimum elongation of 25 to 29% in 2 in., the lower minimum elongation being associated with the higher tenacity.

For stationary engines it is not unusual to employ steel of a tenacity of 34 tons and upwards, specifying a minimum elongation of 25% in 2 in.

For high speed engines of say 600 R.P.M. and upwards alloys steels are frequently used for the sake of surface hardness to resist wear; for example 3% nickel steel or nickel chrome steel, heat treated after rough machining.

General Construction.—Small Diesel Engine crank-shafts are usually solid or semi-built. When the ratio of stroke to bore exceeds about 1·8 a built-up shaft is sometimes used, and typical proportions are given in Fig. 40, the unit being the bore of the cylinder, for a stroke bore ratio$=2{\cdot}0$.

Fig. 41 shews a solid forged crank-shaft for a 4 cylinder stationary engine.

Similar shafts are made in one piece for 1, 2, 3, 4, 5, 6, 7 and 8 cylinders with shaft diameters up to about 10″. The larger sizes of crank-shaft are made in two or more sections. This practice is usual for large marine engines of 6 cylinders and upwards. Very large crank-shafts for 9 and 12 cylinders may even be divided into 3 or 4 sections.

The angular spacing of cranks is determined by considerations of balance, turning effort and torsional resonance, see Chapters X and XIII.

A selection of orders of firing is given in Fig. 42 for four stroke S.A., two stroke S.A., and two stroke D.A. engines. Sometimes evenness of turning effort is sacrificed in the interest of good balance by a symmetrical arrangement of

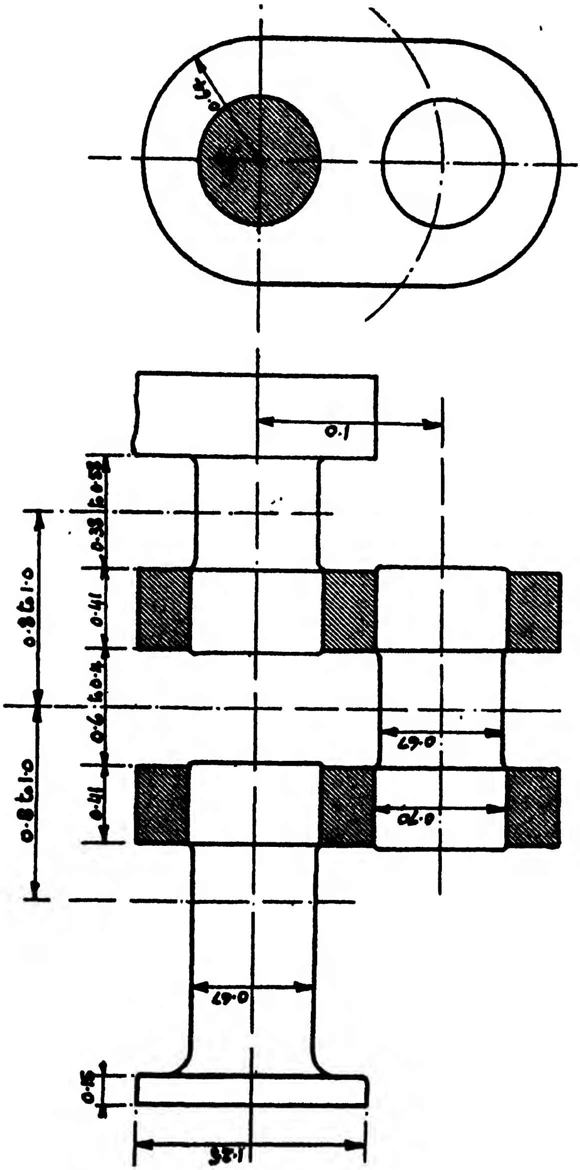


Fig. 40.

cranks involving simultaneous firing of pairs of cylinders. For the larger numbers of cylinders several alternative arrangements of cranks are consistent with good balance. The best order of firing from the point of view of torsional vibration depends on the "vibration form," e.g. whether the node is at the centre or near one end of the crank-shaft.

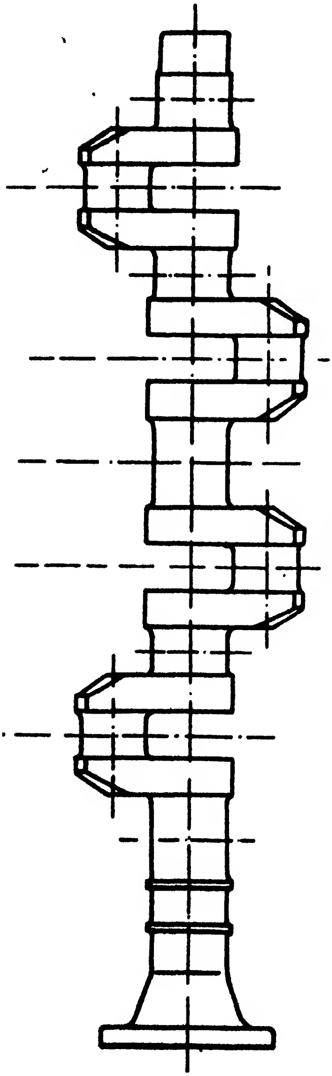


FIG. 41.

Lubrication.—In modern designs of Diesel Engine forced lubrication for the principal bearings is now the rule rather than the exception. When crossheads and totally enclosed crankcases isolated from the cylinders are used it is obvious that forced lubrication can be employed with all the convenience and economy which characterises this system as employed in the high speed steam engine.

With trunk piston engines certain precautions have to be observed to secure economy.

In the first place, oil must be prevented from reaching the underside of the hot piston crown. This is simply effected by a light diaphragm across the piston a few inches from the crown, in fact, just clear of the connecting rod top end bearing. The diaphragm usually takes the form of a circular cover bolted to an internal flange. In some cases this forms a jacket for water or oil cooling of the piston crown. With uncooled pistons a sheet steel diaphragm is sometimes used with good effect. In any case such a diaphragm, by protecting the small end from radiant heat, practically eliminates trouble with this bearing.

In the second place, oil must be prevented from splashing in undue quantity on to the surface of the liner by means of

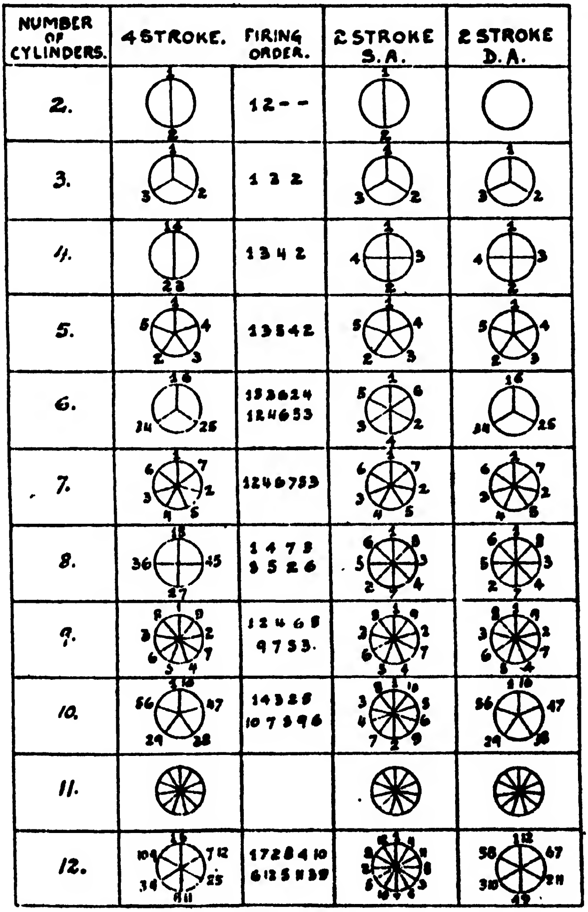


FIG. 42.

guards over the cranks, or attached to the piston itself or the cylinder mouth, and arrangements must be made to scrape the surplus oil off the liner.

Finally, the crank-case must be practically oil-tight to prevent loss of oil and the ingress of dirt.

So far as economy of lubricating oil is concerned, ordinary ring lubrication for the main bearings and the centrifugal banjo arrangement for the big ends leave little to be desired. With suitable arrangements for filtering the oil which is drained from the crank-pit, and using over and over again, the net lubricating oil consumption is readily kept below 0·002 lb. per B.H.P. hour (trunk piston engines). Similar economies are obtainable with forced lubricated trunk engines when the above precautions are observed. Efficient use of non-forced lubrication necessitates certain special features in connection with the crank-shaft. Rings are turned on the latter at each

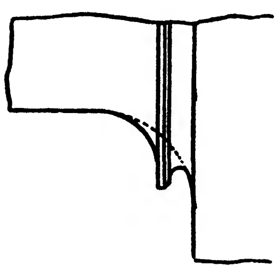


FIG. 43.

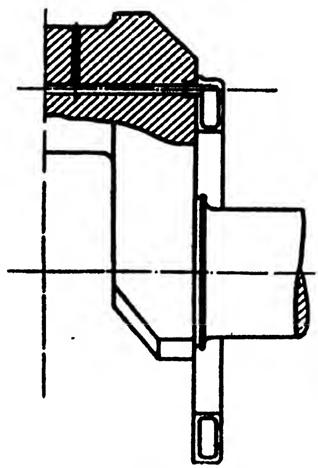


FIG. 44.

end of each journal to throw the squeezed out oil into suitable catcher grooves in the bearing brasses, whereby it is returned to the oil well instead of being thrown off by the crank webs. As these oil throwers have been shewn in some cases to weaken the shaft at its already weakest point, they should be designed so as not to interfere with a good radius between the journal and the web. Fig. 43 shews a section through such an oil thrower.

Fig. 44 shews a crank fitted with centrifugal banjo lubricator. The oil hole leading to the surface of the crank-pin is preferably drilled at an angle of about 30° in advance of the dead centre, so that the upper connecting rod brass receives a supply of oil just before the ignition stroke.

With forced lubrication oil throwers and catchers are not usually fitted, but the shaft requires to be drilled to conduct oil from the journals to the crank-pins. Two systems of drilling are shewn in Figs. 45 and 46.

DETAILS.—(1) **Webs.**—Various types of solid forged webs are

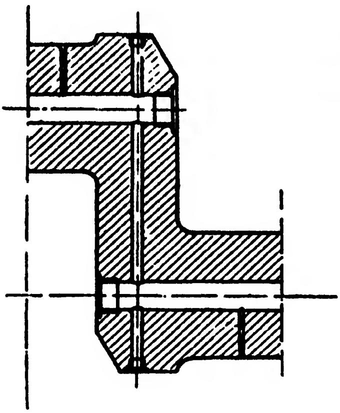


FIG. 45.

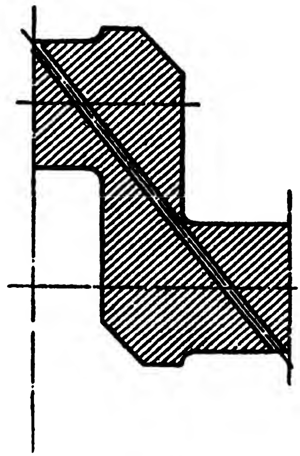


FIG. 46.

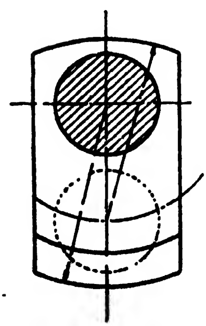


FIG. 47.

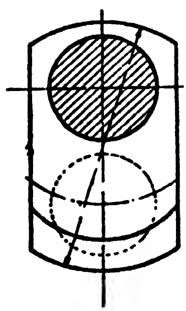


FIG. 48.

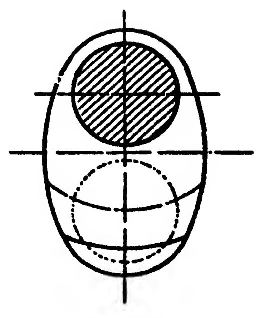


FIG. 49.

shewn in Figs. 47, 48 and 49. The two ends of the straight-sided webs are sometimes turned from the journal and crank-pin centres respectively. Weight can be reduced slightly by turning the two ends at one setting from a centre midway between these two points. Triangular-shaped segments are usually turned off the projecting corners of the webs, and this reduces weight, facilitates feeling the big ends of the connecting rods and gives more clearance for the indicating gear. Balance weights are frequently fitted to one and two cylinder engines to balance the revolving weight of the crank-pins, the big ends, and the otherwise unbalanced portion of the crank webs. The chief difficulty in designing a balance weight is usually to get a sufficiently heavy weight in the space available. The magnitude of the balance weight required is equal to the weight to be balanced, multiplied by the radius of the crank and divided by the radius measured from the centre of the shaft to the centre of gravity of the balance weight. The problem thus resolves itself into a matter of trial and error. Various modes of securing balance weights are illustrated in Figs. 50, 51 and 52. The bolts, or other form of attachment, should be sufficiently strong to carry the centrifugal force of the weight with a low stress.*

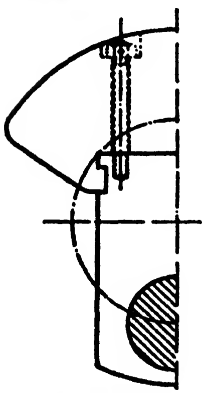


FIG. 50.

(2) **Couplings.**—The couplings connecting the sections of a shaft are made with spigot and faucet joints, the spigots being

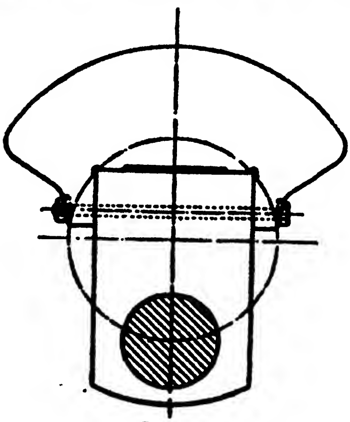


FIG. 51.

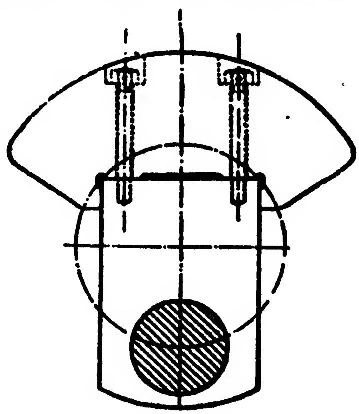


FIG. 52.

turned off after the bolts have been fitted. The bolts belonging to the coupling to which a gear-wheel is fitted are usually made of additional length, and used to secure the wheel. If separate means of securing the wheel are used the interchangeability

* For balancing of multicylinder engines see Chap. XIII.

of the sections of shaft is prejudiced. This is of small importance where land engines are concerned. With marine engines, if it is desired to carry a spare section of shaft, the latter should be provided with any keyways, extra bolt-holes, etc., requisite to enable it to replace any section of the shaft in the event of failure, with a minimum of fitting.

When heavy fly-wheels are fitted, as for instance with dynamo drives, an outer bearing is sometimes placed between the fly-wheel and the driven shaft. The coupling used to connect the projecting end of the crank-shaft to the drive may conveniently be of the common cast iron flanged type, provided with a shrouding to cover the nuts and bolt-heads. See Fig. 53.

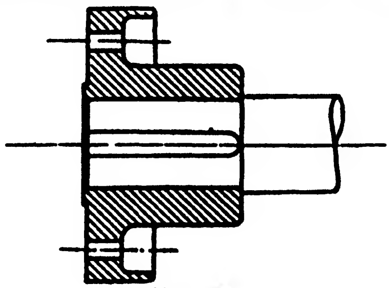


FIG. 53.

Proportions of Crank-shafts.—The strength of Diesel Engine crank-shafts has received careful study by Lloyd's Register over a long period of years and the rules of the Society have been widely adopted. The minimum diameter d of the crank-shaft is expressed as a function of the bore, stroke, mean pressure and maximum pressure, as well as the distance between the inner edges of adjacent bearings. The width and thickness of webs are expressed as fractions of d as follows :—

Thickness of shrunk web	$h = 0{\cdot}625\ d$
,, ,, solid ,,	$h = 0{\cdot}56\ d$
Width of solid web	$w = 1{\cdot}33\ d$
Thickness round eyeholes of shrunk web	$t = \sqrt{\dfrac{0{\cdot}12\ d^3}{h}}$

In order to keep severe critical speeds of torsional vibration outside the working range, the tendency is to reduce the cylinder centres in relation to cylinder bore and to make the journals considerably larger in diameter than ordinary strength considerations would require. This applies more particularly to engines with fly-wheels. Customary cylinder centres are about as follows :—

Small engines for road or rail traction or aircraft	1·2 B to 1·4 B
Single acting engines of heavier build.	1·6 B to 2·0 B
Double ,, ,, ,, ,,	1·8 B to 2·2 B

The webs should join the crank-pins and journals with well-formed radii ; practice varies from about r=0·05 to 0·10 d for solid forged shafts.

Calculation of Stresses in the Crank-shaft.—It is as well to state at the outset that the problem of determining the stresses in a multi-crank-shaft is rather laborious, if done conscientiously, and actual designs are usually based on rational formulæ such as those given in Lloyd's Rules. Supposing a suitable analysis to have been made for a correctly aligned shaft, the whole calculation would require to be revised before the results could be applied with accuracy to the case of a shaft of which the bearings were at different heights owing to the unequal wear of the white metal or flexure of the bedplate. This latter consideration is of itself valuable in emphasising the need of massive foundations where land engines are in question and the desirability of providing an extremely rigid seating for marine designs. On this account the engine seatings and the construction of the ship's floor in way of the engines should be strong and stiff. The framework of the engine may also be designed in such a way as to secure a considerable degree of rigidity against sagging or hogging tendencies. The component parts of the crank-shaft, viz., journals, crank-pins and webs, are subject to bending and twisting actions which vary periodically as the shaft revolves. In the past it has been customary to compute equivalent bending or twisting moments corresponding to the calculated co-existing bending and twisting moments and to proportion the shaft accordingly. Experiments by Guest and others indicate that steel under the influence of combined bending and twisting begins to fail when the shearing stresses, as calculated from the formula quoted below, attain a definite value (about 12,000 lb. per sq. in. for very mild steel under alternating stress) which is independent of the relative amounts of bending and twisting.

$$\text{Maximum shear stress} = \tfrac{1}{2}\sqrt{4 f_s^2 + f_n^2}$$

Where f_n=Normal stress due to bending.
f_s=Shear stress due to twisting.

The equivalent twisting moment which would give the same shear stress as the maximum shear stress due to the combined action of the actual bending and twisting moments is given by

$$T_e = \sqrt{T^2 + B^2}$$

Where T_e = Equivalent twisting moment.
T = Actual twisting moment.
B = Bending moment.

A good approximation to the twisting moment at any point of the shaft at any degree of revolution is obtained by combining in correct sequence the twisting moment curves corresponding to all cylinders "for'd" of the section under consideration. In future the terms "forward" and "aft" will be used to denote the free end and fly-wheel end of the engine respectively regardless of whether the engine under consideration is of marine or land type. The negative twisting moment due to mechanical friction of the moving parts is almost always neglected. That due to a compressor is sometimes allowed for. When dealing with the stress in a crank-pin it should be borne in mind that the twisting moment due to any cylinder is not transmitted through its own crank-pin. For example, if the cranks are numbered as usual from the forward end, the twisting moment in No. 3 crank-pin is that due to cylinders Nos. 1 and 2. The calculation of bending moments is by no means straightforward, and the methods adopted form the distinguishing features of the systems of crank-shaft calculation described below.

Fixed Journal Method of Crank-shaft Calculation.—The assumption underlying this method is that each journal is rigidly fixed at its centre and that the section of shaft between two journals may therefore be treated as a beam encastré at its ends. The assumption of fixed journals would be true for a row of cylinders all firing at the same time. For ordinary conditions, however, the assumption would only hold good if the bearings had no running clearance and were capable of exerting a bending effect on the shaft by virtue of their rigidity. Apart from the fact that the construction of bearings and bearing caps does not suit them for this heavy duty, examination of the bearing surface of well-worn bearings reveals no trace of such cornering action and justifies the view that the bearings merely fulfil their proper functions of carrying thrust in one direction at a time.

Free Journal Method.—With this system each crank is supposed to be loaded at the centre of the crank-pin and supported freely at the centre of the journals, so that the maximum bending moment occurs at the centre of the crank-pin. This assumption would be approximately true for a single cylinder engine if the weight of the fly-wheel and the influence of outboard bearings are neglected. With this method the twisting moment is of very secondary importance, in many cases almost negligible. Comparing this system with that described above, it will be seen that both involve the construction of twisting moment diagrams, though the accuracy of the result is of less importance in the case of the free journal method.

In the following articles a four throw crank-shaft will be investigated on somewhat different lines, with a view to eliminating as many unjustifiable assumptions as possible.

Stress Calculation for a Four Throw Crank-shaft

Data:

Type of engine	four stroke
Number of cylinders	four
Bore of cylinders	10 in.
Stroke	15 in.
Revolutions per minute	300
Connecting rod	5 cranks long
Maximum pressure at firing dead centre,	500 lb. per sq. in.
Diameter of journals and crank-pins	5·25 in.
Length of journals	8 in.
Length of crank-pins	5·5 in.
Thickness of webs	3·25 in.
Width of webs	9 in.
Centres of cylinders	$8+5{\cdot}5+6{\cdot}5=20$ in.
Weight of piston	170 lb.
Weight of connecting rod	190 lb.
Weight of crank-pin	33 lb.
Weight of unbalanced parts of two crank-webs	110 lb.

The method employed in the following investigation is to calculate the values of the forces acting on the shaft when one crank is at its firing top dead centre. The reactions on the bearings will be calculated on the assumption that the centres of the journals remain level and all loads will be treated as concentrated. The effect of unequal level of bearings will also be investigated. In computing the forces acting on the crank-shaft the dead weight of the latter, and also that of the running gear, fly-wheel, etc., will be neglected and the effect of any air compressor will not be considered, nor will the small exhaust pressure remaining in the cylinder which completes its exhaust stroke at the same instant that the cylinder under consideration begins its firing stroke, so that the forces to be dealt with are :—

(1) Those due to cylinder pressure.
(2) Centrifugal force of revolving parts.
(3) Inertia force of reciprocating parts.

These will now be calculated :—

Weight of revolving part of connecting rod, $0 \cdot 65 \times 190$	124 lb.
Weight of unbalanced part of crank webs .	110 lb.
Weight of crank-pin	33 lb.
Total weight of unbalanced revolving parts	267 lb.
Weight of reciprocating part of connecting rod, $0 \cdot 35 \times 190$	66 lb.
Weight of piston	170 lb.
Total weight of reciprocating parts .	236 lb.

Centrifugal acceleration in the crank circle is :—

$$\omega^2 r = \left(\frac{2\pi . 300}{60}\right)^2 \times \frac{7 \cdot 5}{12} = 615 \text{ ft. per sec.}^2$$

Therefore centrifugal effect of revolving parts :—

$$\frac{267 \times 615}{g} = 5100 \text{ lb.}$$

Inertia effect of reciprocating parts at top dead centre :—

$$\frac{236 \times 615}{g} \times 1\tfrac{1}{5} = 5400 \text{ lb.}$$

Inertia effect of reciprocating parts at bottom dead centre :—

$$\frac{236 \times 615}{g} \times (1 - \tfrac{1}{5}) = 3600 \text{ lb.}$$

Combined centrifugal and inertia effect at top dead centre :—

$= 5100 + 5400 = 10{,}500$ lb. upwards.

Combined centrifugal and inertia effect at bottom dead centre :—

$= 5100 + 3600 = 8700$ lb. downwards.

Maximum load due to cylinder pressure :—

$$=0{\cdot}785\times10^2\times500=39{,}000 \text{ lb.}$$

Resultant downward effect of pressure, centrifugal force and inertia force at firing dead centre :—

$$=39{,}000-10{,}500=28{,}500 \text{ lb.}$$

Method of Calculating the Reactions at the Bearings.—Let A I (Fig. 54) represent the axis of the crank-shaft, A C E G I being the centres of the journals and B D F H those of the crank-pins.

R_2 R_4 R_6 R_8 are the applied forces due to cylinders 1, 2, 3, 4 respectively.

R_1 R_3 R_5 R_7 R_9 are the reactions at the bearings unknown in magnitude and direction. Under the influence of these forces the centre line of the shaft assumes some deflected shape,

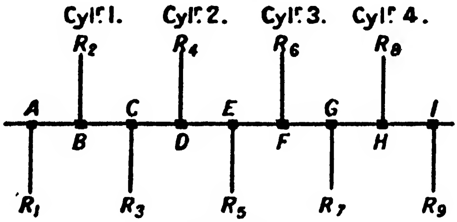


FIG. 54.

and the deflection at any point above or below the straight line joining A I is equal to the sum of the deflections at the same point which would be produced by each of the forces R_2 R_3 R_4 R_5 R_6 R_7 R_8 acting alone, supposing the shaft supported freely at A and I. At C E and G the sum of these deflections must be zero if the bearings are level (ignoring the effect of running clearance). Let c_2 e_2 and g_2 be the deflections at C E and G due to unit load applied at B (the position of R_2), assuming the shaft supported freely at A and I.

$$\left.\begin{array}{c} c_3\ e_3 \text{ and } g_3 \\ c_4\ e_4 \text{ and } g_4 \\ c_5\ e_5 \text{ and } g_5 \\ \text{etc., etc.} \\ c_8\ e_8 \text{ and } g_8 \end{array}\right\} \text{are the corresponding deflections at C E and G due to unit load applied at C D E, etc.}$$

The values of c_2 e_2 and g_2, etc., are readily found by the usual formulæ for the deflection of beams.

Now since the total deflection at C E and G is zero, the following equations hold good :—

$$R_2\ c_2+R_3\ c_3+R_4\ c_4+R_5\ c_5+R_6\ c_6+R_7\ c_7+R_8\ c_8=0 \ldots(1)$$
$$R_2\ e_2+R_3\ e_3+R_4\ e_4+R_5\ e_5+R_6\ e_6+R_7\ e_7+R_8\ e_8=0 \ldots(2)$$
$$R_2\ g_2+R_3\ g_3+R_4\ g_4+R_5\ g_5+R_6\ g_6+R_7\ g_7+R_8\ g_8=0 \ldots(3)$$

In these three equations the only unknown quantities are $R_3\ R_5\ R_7$, which can therefore be determined. The remaining unknown reactions R_1 and R_9 are found by equating moments about I and A respectively. In solving the above equations downward forces and deflections will be considered positive and upward forces and deflections negative.

Determination of c_2 e_2 g_2, etc.—In determining these constants it will be assumed that the shaft deflects under load as though it were a cylindrical beam of the same diameter as the crank-pins and journals. Considering that the webs of a Diesel Engine crank are short, the presumption is probably not inaccurate, but it would be interesting to see this point investigated, as it could readily be, by means of models. So long as bearings at constant level are assumed, the actual value of the deflection is of no importance, as the method of calculation depends only on the relative deflection at the different points considered. The problem therefore resolves itself into finding the deflected form of a beam freely supported at each end under the influence of a concentrated load placed anywhere between the supports. This may be done by treating each end of the beam as a cantilever. The deflection of the end of a cantilever carrying a load at the end is given by :—

$$\text{Deflection at the end of cantilever in in.}=\frac{W.l^3}{3\,EI}$$

Where W = Load in lb.

E = 30,000,000 lb. per sq. in. (for steel).

I = Moment of inertia (transverse) of the section of beam in.

For the shaft under consideration :—

$$I=\frac{\pi}{64}\times 5{\cdot}25^4=37{\cdot}3\text{ in.}^4$$

Fig. 55 shews the values of the deflection at various fractional points in the length of the cantilever, the deflection at the end being unity. These results are applied to the case of a beam as follows :—

Let AB be a beam (Fig. 56) supported at A and B and carrying a load W_1 at any point C. The reaction at A is equal to $W_1\frac{CB}{AB}$. Let this be denoted by W_2. At A erect a perpendicular AD equal to some convenient scale to the deflection of the cantilever AC, due to the load W_2 at its end. Draw the deflected shape of this cantilever (DC) by means of the proportions given

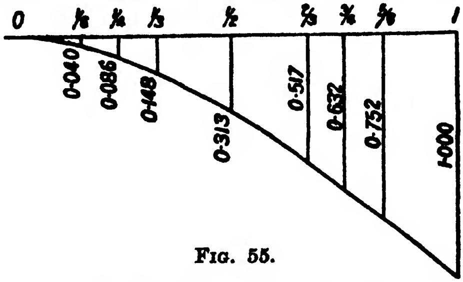


Fig. 55.

in Fig. 55. Proceed similarly with cantilever C B, obtaining the deflected shape C F. Join D and F, then the deflection of the beam at any point X is the vertical intercept " x " shewn in the figure.

Application to the Case in Hand.—The constants c_2 e_2 g_2 being the deflections at various points on a beam due to unit load applied at other various points, are independent of the system of loads and will therefore be dealt with before special

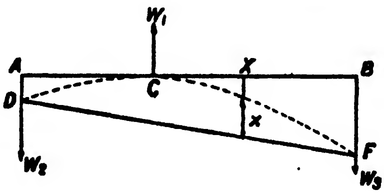


Fig. 56.

cases of loading are considered. In order to obtain manageable figures, the deflection will be reckoned in thousandths of an inch and the unit load will be taken as 10,000 lb. Deflection of cantilevered portion of shaft in thousandths of an inch is given by :—

$$\frac{W.(\frac{l}{10})^3}{3\times 30\times 37\cdot 3}=\frac{W.(\frac{l}{10})^3}{3360}$$

Where W=Load in lb.
l=Length in inches.

The diagram Fig. 57 shews the process of determining $c_2 e_2 g_2$, etc., in particular :—

Unit load at B=10,000 lb., AB=10″, BI=70″

$$\text{Reaction at A} = \frac{10{,}000\times 70}{80} = 8750 \text{ lb.}$$

$$\text{,, \quad ,, \quad I} = \frac{10{,}000\times 10}{80} = 1250 \text{ lb.}$$

$$\text{Deflection of cantilever AB} = \frac{8750\times 1^3}{3360} = 2{\cdot}6\tfrac{\text{in.}}{1000}$$

$$\text{,, \qquad ,, \qquad IB} = \frac{1250\times 7^3}{3360} = 127\tfrac{\text{in.}}{1000}$$

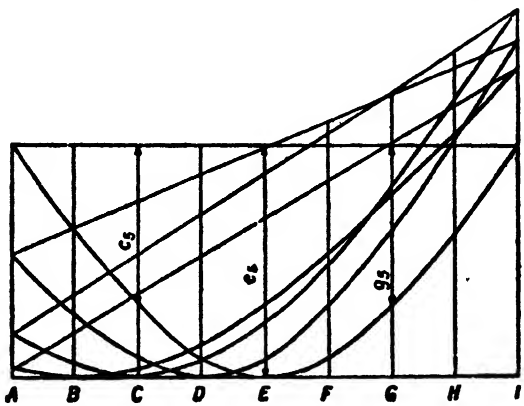


FIG. 57.

Deflected shapes of cantilevers drawn by plotting ordinates from the proportion given in Fig. 55.

The constants are found to be :—

$c_2=30{\cdot}2 \quad e_2=35{\cdot}0 \quad g_2=22{\cdot}0$

Since the deflection at G, due to a load at H, is the same as that at C, due to the same load at B, and so on, therefore :—

$g_8=30{\cdot}2 \quad e_8=35{\cdot}0 \quad g_8=22{\cdot}0$

By the same methods (see Fig. 57) :—

$c_3=53{\cdot}6 \quad e_3=65{\cdot}5 \quad g_3=41{\cdot}7$
$g_7=53{\cdot}6 \quad e_7=65{\cdot}5 \quad c_7=41{\cdot}7$
$c_4=65{\cdot}2 \quad e_4=87{\cdot}1 \quad g_4=57{\cdot}0$
$g_6=65{\cdot}2 \quad e_6=87{\cdot}1 \quad c_6=57{\cdot}0$
$c_5=65{\cdot}5 \quad e_5=95{\cdot}3 \quad g_5=65{\cdot}5$

These values have been computed with rather less trouble and with greater accuracy by the formulæ for the deflection at any point in a beam due to a load at any other point (see Morley's "Strength of Materials"). The graphical method of determining the constants should not be relied upon except as a check, as small errors in the constants give rise to large errors in the calculated reactions.

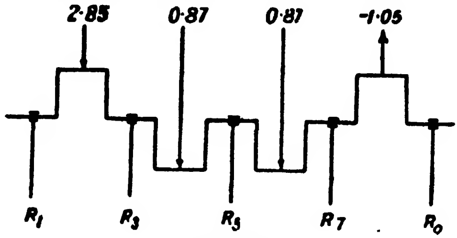


FIG. 58.

The conditions of loading will now be considered.

Case I. Crank 1 on Firing Dead Centre.—The magnitudes and directions of the applied forces are shewn in Fig. 58. Hence :—

$$R_2c_2=2{\cdot}85\times30{\cdot}2=86{\cdot}1 \qquad R_4c_4=0{\cdot}87\times65{\cdot}2=56{\cdot}7$$
$$R_2e_2=2{\cdot}85\times35{\cdot}0=99{\cdot}7 \qquad R_4e_4=0{\cdot}87\times87{\cdot}1=75{\cdot}8$$
$$R_2g_2=2{\cdot}85\times22{\cdot}0=62{\cdot}7 \qquad R_4g_4=0{\cdot}87\times57{\cdot}0=49{\cdot}6$$

$$R_6c_6=0{\cdot}87\times57{\cdot}0=49{\cdot}6 \qquad R_8c_8=-1{\cdot}05\times22{\cdot}0=-23{\cdot}1$$
$$R_6e_6=0{\cdot}87\times87{\cdot}1=75{\cdot}8 \qquad R_8e_8=-1{\cdot}05\times35{\cdot}0=-36{\cdot}8$$
$$R_6g_6=0{\cdot}87\times65{\cdot}2=56{\cdot}7 \qquad R_8g_8=-1{\cdot}05\times30{\cdot}2=-31{\cdot}7$$

From which :—

$$R_2c_2+R_4c_4+R_6c_6+R_8c_8=169{\cdot}3$$
$$R_2e_2+R_4e_4+R_6e_6+R_8e_8=214{\cdot}5$$
$$R_2g_2+R_4g_4+R_6g_6+R_8g_8=137{\cdot}3$$

Substituting these values in equations (1), (2), and (3), we obtain :—

$$53{\cdot}6\ R_3+65{\cdot}5\ R_5+41{\cdot}7\ R_7=-169{\cdot}3 \quad\ldots\ldots\ldots(4)$$
$$65{\cdot}5\ R_3+95{\cdot}3\ R_5+65{\cdot}5\ R_7=-214{\cdot}5 \quad\ldots\ldots\ldots(5)$$
$$41{\cdot}7\ R_3+65{\cdot}5\ R_5+53{\cdot}6\ R_7=-137{\cdot}3 \quad\ldots\ldots\ldots(6)$$

From which :—

$$R_3=-2{\cdot}434 \qquad R_5=-0{\cdot}735 \qquad R_7=+0{\cdot}230$$

Equating moments about A :—

$$8\ R_9+7\ R_8+6\ R_7+5\ R_6+4\ R_5+3\ R_4+2\ R_3+R_2=0$$

$$\therefore\ 8\,R_9 = 7\times1{\cdot}05 - 6\times0{\cdot}230 - 5\times0{\cdot}87 + 4\times0{\cdot}735 - 3\times0{\cdot}87 + 2\times2{\cdot}434 - 2{\cdot}850$$

Whence $R_9 = 0{\cdot}496$.

Equating moments about I :—

$$8\,R_1 + 7\,R_2 + 6\,R_3 + 5\,R_4 + 4\,R_5 + 3\,R_6 + 2\,R_7 + R_8 = 0$$

$$\therefore\ 8\,R_1 = -7\times2{\cdot}85 + 6\times2{\cdot}434 - 5\times0{\cdot}87 + 4\times0{\cdot}735 - 3\times0{\cdot}87 - 2\times0{\cdot}230 + 1{\cdot}05$$

Whence $R_1 = -1{\cdot}097$.

Knowing all the forces, the bending moment at A B C, etc., can now be tabulated thus :—

Point.	Moments.	B.M. in. in. lb.
A	. . . .	0
B	$10{,}970\times10$	+109,700
C	$10{,}970\times20-28{,}500\times10$	−65,600
D	$10{,}970\times30-28{,}500\times20+4{,}340\times10$ . .	+2,500
E	$10{,}970\times40-28{,}500\times30+4{,}340\times20--8{,}700\times10$	−16,400
F	$-4960\times30+10{,}500\times20-2300\times10$ . .	+38,200
G	$-4960\times20+10{,}500\times10$	−5,800
H	-4960×10	−49,600
I	. . . .	0

Since the bending modulus of the shaft is

$$\frac{\pi}{32}\times5{\cdot}25^3 = 14{\cdot}2 \text{ in.}^3$$

Therefore,

Maximum stress due to bending (occurring in No. 1 crank-pin at top firing centre) is equal to :—

$$\frac{109{,}700}{14{\cdot}2} = 7{,}720 \text{ lb. per in.}^2$$

Bending stress, according to fixed journal method :—

$$\frac{W.L}{8\,Z} = \frac{28{,}500\times20}{8\times14{\cdot}2} = 5{,}020 \text{ lb. sq. in.}$$

According to free journal method :—

$$\text{Stress} = \frac{28{,}500\times20}{4\times14{\cdot}2} = 10{,}040 \text{ lb. sq. in.}$$

Case II. No. 2 Crank on Firing Dead Centre.—The magnitudes and directions of the applied forces are shewn in Fig. 59,

$$R_2=0{\cdot}870 \quad R_4=2{\cdot}850 \quad R_6=-1{\cdot}05 \quad R_8=0{\cdot}870$$

$$R_2c_2=0{\cdot}87\times30{\cdot}2=26{\cdot}3 \qquad R_4c_4=2{\cdot}85\times65{\cdot}2=185{\cdot}6$$
$$R_2e_2=0{\cdot}87\times35{\cdot}0=30{\cdot}5 \qquad R_4e_4=2{\cdot}85\times87{\cdot}1=248{\cdot}0$$
$$R_2g_2=0{\cdot}87\times22{\cdot}0=19{\cdot}2 \qquad R_4g_4=2{\cdot}85\times57{\cdot}0=162{\cdot}3$$

$$R_6c_6=-1{\cdot}05\times57{\cdot}0=-59{\cdot}8 \qquad R_8c_8=0{\cdot}87\times22{\cdot}0=19{\cdot}2$$
$$R_6e_6=-1{\cdot}05\times87{\cdot}1=-91{\cdot}4 \qquad R_8e_8=0{\cdot}87\times35{\cdot}0=30{\cdot}5$$
$$R_6g_6=-1{\cdot}05\times65{\cdot}2=-68{\cdot}5 \qquad R_8g_8=0{\cdot}87\times30{\cdot}2=26{\cdot}3$$

From which

$$R_2c_2+R_4c_4+R_6c_6+R_8c_8=171{\cdot}3$$
$$R_2e_2+R_4e_4+R_6e_6+R_8e_8=217{\cdot}6$$
$$R_2g_2+R_4g_4+R_6g_6+R_8g_8=139{\cdot}3$$

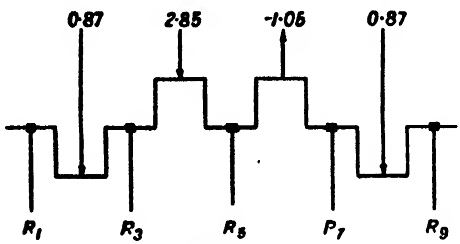


Fig. 59.

The three equations for determining $R_7R_5R_3$ are therefore :—

$$53{\cdot}6\ R_7+65{\cdot}5\ R_5+41{\cdot}7\ R_3=-171{\cdot}3$$
$$65{\cdot}5\ R_7+95{\cdot}3\ R_5+65{\cdot}5\ R_3=-217{\cdot}6$$
$$41{\cdot}7\ R_7+65{\cdot}5\ R_5+53{\cdot}6\ R_3=-139{\cdot}3$$

from which

$$R_7=-2{\cdot}514 \quad R_5=-0{\cdot}713 \quad R_3=0{\cdot}226$$

Taking moments about A and I,

$$R_1=-0{\cdot}072 \quad R_9=-0{\cdot}467$$

from which the following figures for the bending moments are obtained :—

Point.	Moments.	B.M. in in. lb.
A	. . . .	0
B	720×10.	$+7{,}200$
C	$720 \times 20 - 8700 \times 10$.	$-72{,}600$
D	$720 \times 30 - 8700 \times 20 + 25{,}140 \times 10$ . .	$+99{,}000$
E	$720 \times 40 - 8700 \times 30 + 25{,}140 \times 20 - 28{,}500 \times 10$	$-14{,}400$
F	$4670 \times 30 - 8700 \times 20 - 2260 \times 10$ . .	$-56{,}500$
G	$4670 \times 20 - 8700 \times 10$	$+6{,}400$
H	4670×10	$+46{,}700$
I	. . . .	0

$$\text{Maximum bending stress } \frac{99{,}000}{14\cdot 2} = 6{,}970 \text{ lb. sq. in.}$$

Considerably less than in Case I.

Case III. Loading as in Case I, but bearings C and G supposed worn down 20 thousandths and bearing E 25 thousandths of an inch below the level of the line joining A I.

The difference in height of the bearings is very simply allowed for by putting 20, 25, and 20 respectively on the right-hand side of the equations (1), (2), and (3) instead of zero.

The equations for determining R_3, R_5, and R_7 then become :—

$$53\cdot 6\ R_3 + 65\cdot 5\ R_5 + 41\cdot 7\ R_7 = 20 - 169\cdot 3 = -149\cdot 3$$
$$65\cdot 5\ R_3 + 95\cdot 3\ R_5 + 65\cdot 5\ R_7 = 25 - 214\cdot 5 = -189\cdot 5$$
$$41\cdot 7\ R_3 + 65\cdot 5\ R_5 + 53\cdot 6\ R_7 = 20 - 137\cdot 3 = -117\cdot 3$$

The following values are obtained for the reactions at the bearings :—

$$R_1 = -1\cdot 398 \quad R_3 = -1\cdot 880 \quad R_5 = -1\cdot 243 \quad R_7 = +0\cdot 790$$
$$R_9 = +0\cdot 191$$

The bending moments are as follows :—

Point.	Moments.	B.M. in in. lb.
A	. . . .	0
B	13,980 × 10	+139,800
C	13,980 × 20 − 28,500 × 10 . . .	−5,400
D	13,980 × 30 − 28,500 × 20 + 18,800 × 10 .	+37,400
E	13,980 × 40 − 28,500 × 30 + 18,800 × 20 − 8,700 × 10	−6,800
F	1910 × 30 + 10,500 × 20 − 7,900 × 10 .	+73,700
G	1910 × 20 + 10,500 × 10 . . .	+66,800
H	1910 × 10.	+19,100
I	. . . .	0

Maximum bending stress $\frac{139,800}{14 \cdot 2} = 9,800$ lb.sq. in.

Comparing the above figures with those obtained in Case I it will be seen that the maximum stresses have been increased to the extent of about 28% by the difference of level of the bearings. The levels of the bearings may be checked by various means, for example, by " clock gauge " measurements between the crank-web. The errors should be corrected before reaching the values assumed above.

Conclusions.—1. The value of the bending moments at a crank-pin on the top firing centre is greater for a crank-pin situate at one end of the shaft than that at one nearer the centre of the shaft.

2. The bending moments at certain crank-pins and journals may be as great or greater than the bending moment at the crank-pin which is receiving the greatest applied load.

3. No general rule for the bending moment at a Diesel Engine crank-pin or journal can represent the true state of affairs, but every different arrangement of cranks and number of cylinders requires to be investigated individually.

4. Difference of level of the bearings, due to wear or otherwise, gives rise to greatly increased bending moments, which can be calculated approximately in the manner described.

The methods of calculation which have been illustrated in this chapter can be applied to cases involving any number of cranks, and the effects of fly-wheels, the rotors of electric generators, outboard bearings, etc., can be included. When the number of cylinders is three or less the weight of the fly-wheel is considerable, and cannot therefore be ignored. In these cases allowance should be also made for the practice of packing the outward bearing above the level of the engine main bearings. For engines of four cylinders and over, the weight of the fly-wheel and the presence of outboard bearings can probably be neglected with safety.

The labour involved in solving simultaneous linear equations increases as the square of the number of unknowns. For a description of a machine devised to do this work mechanically, see the "Treatise on Natural Philosophy," vol. i, Kelvin and Tait.

Graphical Determination of the Twisting Moments.—In the processes described below the following approximations have been made:—

1. The negative twisting moments due to any air compressor at the forward end of the engine have been neglected. These moments are small in comparison with the moments due to the working cylinders, and being opposite in direction to the maximum moments tend to reduce the latter by a small amount.

2. Moments due to the dead weight of the revolving and reciprocating parts have also been neglected. In a very large engine it would be advisable to take these into consideration, as other things being equal the dead weight of the running gear per square inch of piston area increases as the scale of the engine.

3. The twisting moments due to mechanical friction have been neglected, as (so far as the present writer is aware) the distribution and variation of the friction forces are not known with any exactitude, and in any case one is a little on the safe side in neglecting them. These friction moments of course accumulate as one passes from the forward to the aft end of the engine, where they amount in aggregate to about 15% of the mean indicated twisting moment, so their effect on the forward end of the shaft is quite negligible.

4. The moment of inertia of the fly-wheel has been assumed to be large in comparison with the fly-wheel effect of the revolving and reciprocating parts of the running gear. In cases where the fly-wheel is very small, or omitted altogether, as in some two stroke marine engines, the irregularities of turning effort are mainly absorbed by the angular acceleration of the crank masses. Allowance is readily made for this effect in the following manner. The combined twisting moment curve for all the cylinders is first found without allowance for fly-wheel effect, and a new zero line is taken at the height corresponding to the mean twisting moment. Ordinates measured to this new zero line represent fluctuations of the twisting moment from its mean value. These ordinates are now divided into segments proportional to the fly-wheel effects of the fly-wheel and crank masses. For example, if there are four cylinders and the moment of inertia of the fly-wheel is three times that of one set of crank masses, then the ordinates will be divided into seven parts, one part being applied in opposite sense to corresponding points on the twisting moment curve of each cylinder and the remaining three parts of each ordinate form the ordinates of a curve of the twisting moments absorbed by the angular inertia of the fly-wheel.

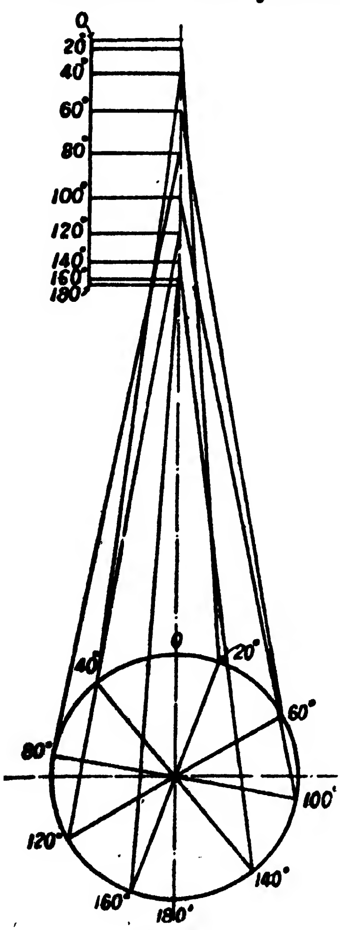


Fig. 60.

In the example worked out below it will be assumed that the fly-wheel is large compared with the crank masses, so that the process described briefly above is not necessary.

Fig. 60 is a skeleton diagram of the connecting rod positions for every 20 degrees of revolution of the crank-shaft for the determination of piston displacements.

Fig. 61 is a typical full load indicator card calibrated for pressures vertically and degrees of revolution horizontally. Points corresponding to each 20 degrees of revolution are

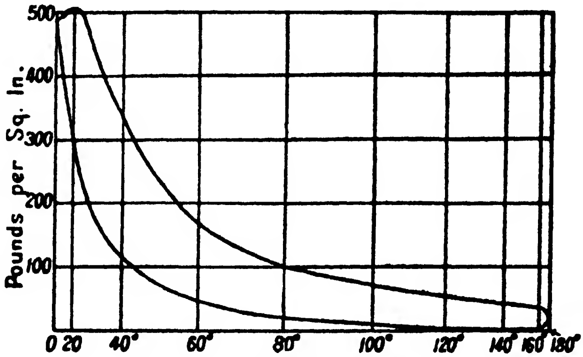


Fig. 61.

marked on the diagram by scaling the piston displacements off Fig. 60.

On Fig. 62 cylinder pressures are plotted on a crank angle base from 0 to 720 degrees (four stroke engine). The pressure during the suction and exhaust strokes is assumed atmospheric. The inertia effect of the reciprocating parts per square inch of piston area is plotted from the following figures :—

Inertia effect at top dead centre $\dfrac{236\times615}{g}(1+\frac{1}{5})=5400$ lb.

" " bottom " $\dfrac{236\times615}{g}(1-\frac{1}{5})=3600$ "

" " $90°=\dfrac{236\times615}{g}\times\dfrac{1}{5}=900$ lb.

" " $45°=\dfrac{236\times615}{g}\times\dfrac{1}{\sqrt{2}}=3200$ lb.

Corresponding figures per sq. in. of piston area (78·5 sq. in.) are 69, 46, 11·5, and 41 lb. per sq. in. respectively.

The centrifugal forces of the revolving masses being radial produce no twisting effect on the shaft. The dotted line (Fig. 62) is the resultant of the pressure and inertia curves. The twisting moments are now computed as in the following table :—

Degrees from top dead centre.	Leverage in ins.	Resultant force in lb./in.² of piston area.	Twisting moments in. lb.	Degrees from top dead centre.	Leverage in ins.	Resultant force in lb./in.² of piston area.	Twisting moments in. lb.
0	0	418	0	360	0	—69	0
20	3·00	444	104,000	340	—3·00	—60	14,100
40	5·63	284	125,000	320	—5·63	—46	20,400
60	7·20	138	78,000	300	—7·20	—25	14,100
80	7·65	100	60,000	280	—7·65	0	0
100	7·13	90	50,000	260	—7·13	20	—11,200
120	5·85	90	41,400	240	—5·85	39	—17,900
140	4·08	88	28,205	220	—4·08	47	—15,100
160	2·10	85	14,000	200	—2·10	59	—9,750
180	0	73	0	180	0	73	0
360	0	—69	0	720	0	418	0
380	3·00	—60	—14,100	700	—3·00	232	—54,500
400	5·63	—46	—20,400	680	—5·63	69	—30,500
420	7·20	—25	—14,100	660	—7·20	22	—12,400
440	7·65	0	0	640	—7·65	22	—13,200
460	7·13	20	11,200	620	—7·13	32	—17,900
480	5·85	37	17,000	600	—5·85	42	—19,300
500	4·08	44	14,100	580	—4·08	44	—14·100
520	2·10	45	7,400	560	—2·10	45	—7,400
540	0	46	0	540	0	46	0

The leverage tabulated in the second column is found by the well-known graphical construction in Fig. 60, where the line of the connecting rod is produced (if necessary) to meet the horizontal line through the centre of the shaft, the intercept being the leverage required to the same scale as the rest of the diagram. The twisting moments are found by multiplying the leverage in inches by the resultant forces per sq. in. of

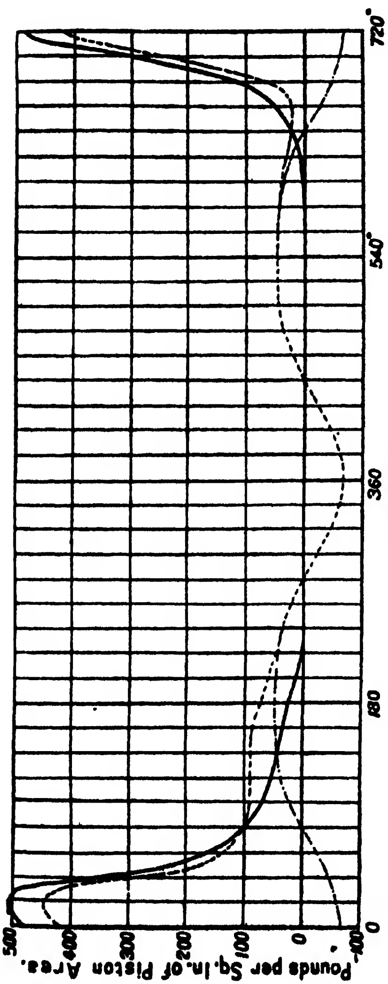


Fig. 62.

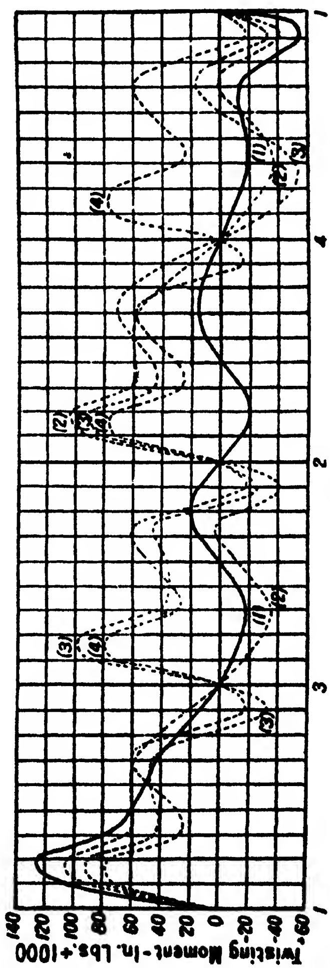


Fig. 63.

piston area in lb. per sq. in. and by the piston area in sq. in. (in this case 78·5 sq. in.).

Forces acting towards the crank are considered positive, whether they are expansion forces or otherwise, and those acting away from the crank negative. Assuming rotation clockwise, leverages to the right hand of the centre line are positive and those to the left negative. The signs of the moments then look after themselves according to the signs of their factors. It is not unusual to see the leverage in a case of this sort treated as though it were always positive. The disadvantage of this proceeding is that in order to get the signs of the moments correct those of the resultant forces have to be reversed at every dead centre, which besides being incorrect from a mathematical standpoint is inconvenient for the draughtsman and confusing to others. The tabulated values of the twisting moment are plotted in Fig. 63 (full line curve). Identical curves for cylinders 3, 4, and 2 could be plotted in their respective places at 180 degrees apart, in the order named, but are omitted for the sake of clearness. Dotted curve numbered 2 is the resultant of the curves belonging to cylinders 1 and 2. Dotted curve 3 is the resultant of the curves belonging to cylinders 1, 2, and 3, and so on. The simplest way of obtaining these resultants is to trace the primary curve on a piece of transparent paper and move it sideways into its required position for the next cylinder, and then for every required ordinate move the paper vertically (guided by the vertical degree lines) until the zero line coincides with the top of the ordinate of the curve to which it is required to add the effect of another cylinder. In this position prick through the top of the ordinate of the curve on the tracing-paper to the diagram underneath. These resultant curves enable the twisting moment at any crank-pin or journal at any angular position to be read off the diagram.

Combined Effect of Bending and Twisting.—It will be seen that the peaks of the twisting moment curves occur about 30 degrees after the dead centres, and that the results previously obtained for the bending moments with the cranks on dead centre apply very closely to this position also, so that the tabulated values of the bending moments at the various journals and crank-pins combined with the twisting moments existing at these points 30 degrees after the corresponding firing dead centres have been passed, represent the maximum conditions of stress at the points in question. The conditions of bending when cranks 3 and 4 are on firing centre are of course the same as those obtaining when cranks 2 and 1 respectively are in that position, the order in which the bending moments occur being reversed.

For example, the bending moment at No. 2 crank-pin when No. 4 cylinder is firing is the same as the bending moment at No. 3 crank-pin when No. 1 cylinder is firing, and so on. A comparison of the following table with the twisting moment curves and the bending moments tabulated in the previous articles will make the matter clear.

The equivalent twisting moment equals $\sqrt{T^2+B^2}$, and is that twisting moment which would give the same shear stress as the maximum shear stress due to the combined action of twisting and bending moments actually obtaining.

Position.	Which crank 30° past firing dead centre.	Bending moment in. lb. (see previous tables).	Number of twisting moment curve.	Twisting moment in. lb. from curves.	Equivalent twisting moment in. lb.	Maximum shear stress lb. per sq. in.
A. Journal	—	—	—	—	—	—
B. Crank-pin No. 1	No. 1	109,700	—	—	109,700	3,870
	No. 2	7,200				
	No. 3	46,700				
	No. 4	49,600				
C. Journal	No. 1	65,600	(1)	127,000	143,000	5,040
	No. 2	72,600		11,000		
	No. 3	6,400		13,000		
	No. 4	5,800		17,000		
D. Crank-pin No. 2	No. 1	2,500	(1)	127,000	127,000	4,480
	No. 2	99,000		11,000		
	No. 3	56,500		13,000		
	No. 4	38,200		17,000		
E. Journal	No. 1	16,400	(2)	112,000	116,000	4,090
	No. 2	14,400		115,000		
	No. 3	14,400		31,000		
	No. 4	16,400		31,000		

Position.	Which crank 30° past firing dead centre.	Bending moment in. lb. (see previous tables).	Number of twisting moment curve.	Twisting moment in. lb. from curves.	Equivalent twisting moment in. lb.	Maximum shear stress lb. per sq. in.
F. Crank-pin No. 3	No. 1 No. 2 No. 3 No. 4	38,200 56,500 99,000 2,500	(2)	112,000 115,000 31,000 31,000	128,000	4,510
G. Journal	No. 1 No. 2 No. 3 No. 4	5,800 6,400 72,600 65,600	(3)	100,000 93,000 93,000 45,000	100,000	3,530
H. Crank-pin No. 4	No. 1 No. 2 No. 3 No. 4	49,600 46,700 7,200 109,700	(3)	100,000 93,000 93,000 45,000	112,000	3,950
I. Journal	No. 1 No. 2 No. 3 No. 4	—	(4)	82,000 82,000 82,000 82,000	—	2,890

Conclusions.—Maximum Shear Stress, 5,040 lb. sq. in.—Taking the fatigue stress in shear for mild steel, subject to combined bending and twisting at 15,000 lb. per sq. in., the factor of safety for a shaft newly lined up is about 3, and diminishes very considerably as the bearings become worn out of level.

The high values of the stresses at the centre of the shaft point to the advisability of making all couplings between sections of the crank-shaft of the full torsional strength of the shaft, i.e. the aggregate shearing area of the coupling bolts multiplied by the radius of their pitch circle should be equal to the twisting modulus of the shaft. Thickness of coupling flanges $^1/_4$ diameter of the shaft.

The subject of Torsional Vibration of Crank-shafts is considered in Chapter X.

Twisting Moments in Crank-shafts.—As indicated on page 158, the method usually adopted of computing accumulated twisting moments is not strictly correct. The method consists in regarding the twisting moment at any journal as the algebraic sum of the moments due to all cylinders forward of the section under consideration. This would be correct for an engine at rest with the fly-wheel (aft) rigidly fixed. It is also approximately correct when the moment of inertia of the fly-wheel is very large in comparison with that of the crank-masses. In many instances this is not the case as the fly-wheel may be replaced with a light turning wheel, and fully built cranks with or without balance weights may be fitted having considerable fly-wheel effect. In such cases the variations of turning effort are mainly absorbed by the angular inertia of the crank-masses. A method of calculating the twisting moments under these conditions is described below.

The mean twisting moment simply adds up from forward to aft so that if Tm is mean twisting moment due to one cylinder, the mean twisting moment aft of the second cylinder is 2Tm, and so on. It is reasonable in calculating these mean twisting moments to make a deduction of about 15% from the "indicated" twisting moments to allow for mechanical friction. If there is a compressor forward the mean twisting moment required to drive it should be deducted. This being arranged, it is only necessary in what follows to consider variations of twisting moment above and below the mean.

Imagine that the twisting moment curve for one cylinder is laid down on a crank angle base and repeated in correct sequence for as many cylinders as there may be. Call the curves 1, 2, 3.

Let (1) = Curve 1.
(2) = Sum of curves 1 and 2.
(3) = ,, ,, ,, 1, 2 and 3, i.e. (2) + 3.
(N) = ,, ,, ,, 1, 2, 3 ....N = (R) say.

Let ${}_1T_2$ = curve of twisting moment between cylinders 1 and 2.

${}_2T_3$ = ,, ,, ,, 2 and 3.
${}_{N-1}T_N$ = ,, ,, ,, N − 1 and N.
${}_NT_f$ = ,, ,, ,, N and fly-wheel.

Then if the fly-wheel is very heavy in comparison with the crank-masses :—

$$\left.\begin{aligned} {}_1T_2&=(1) \\ {}_2T_3&=(2) \\ \hline {}_NT_F&=(R) \end{aligned}\right\} \text{Approximately ; this is the usual assumption} \quad \ldots\ldots \quad \text{I}$$

A more accurate formula will now be given applicable to all cases, including those in which the fly-wheel is light or absent.

For simplicity we suppose there is no compressor forward.

Let I_1=moment of inertia of crank-masses of No. 1 cylinder.

I_2 ,, ,, ,, ,, ,, ,, ,, 2 ,,

I_N ,, ,, ,, ,, ,, ,, N ,,

I_F ,, ,, ,, ,, ,, ,, fly-wheel.

$\Sigma I = I_1 + I_2 + I_3 + \ldots\ldots\ldots\ldots I_N + I_F$.

Then,

$$\left.\begin{aligned} {}_1T_2&=(1)-\frac{I_1}{\Sigma I}\cdot(R) \\ {}_2T_3&=(2)-\frac{I_1+I_2}{\Sigma I}(R) \\ {}_NT_F&=(N)-\frac{I_1+I_2 \ldots\ldots I_N}{\Sigma I}(R) = \frac{I_F}{\Sigma I}(R) \end{aligned}\right\} \ldots\ldots \quad \text{II}$$

The rule is, therefore, as follows :—

From the twisting moment curves (1), (2), (3), etc., as found in the usual manner, deduct the ordinates of a curve, the ordinates of which are a fraction of the resultant twisting moment variation curve above and below the mean: the fraction being equal to the ratio of the aggregate moment of inertia forward of the section under consideration to the total moment of inertia. The necessary modifications to the above formulæ to take into consideration an air compressor, dynamo, closely coupled propeller, etc., should be obvious.

Strictly the above formulæ only apply to those harmonic components of twisting moment the critical speeds of which are much above the running speed, but these generally comprise the major parts of the torque variations. In the case of a marine engine coupled to a propeller the latter should be included in the calculation. For this purpose the moment of inertia of the propeller should be multiplied by the factor $1 \div \left\{1-\left({}^{n}/_{n_c}\right)^2\right\}$ where n=running speed and n_c=critical speed of the shafting system with regard to the main component of resultant torque variation, since this factor is that by which the torque in the shafting is magnified by resonance.

The derivation of the formulæ given above is easily illustrated by an example. Consider ${}_2T_3$ for instance: it is made up of part due to cylinders 1 and 2 and part due to cylinders 3 to N. Of all the torque variations a fraction $=I_1+I_2/\Sigma I$ is absorbed in accelerating and decelerating masses I_1 and I_2; the remaining fraction $\left\{\Sigma I-\left(I_1+I_2\right)\right\}/\Sigma I$ is absorbed in accelerating and decelerating masses I_3, I_4..I_N, I_P. Also applied torque variations of the same sign (+ or —), with regard to a section aft of all the masses are of opposite sign with regard to any section between cranks according as the section is forward or aft of the applied torque.

Of torque variation (2) the fraction $\left\{\Sigma I-\left(I_1+I_2\right)\right\}/\Sigma I$ is transmitted through section 2 to 3. Of torque variation (R)—(2) the fraction $(I_1+I_2)/\ \Sigma I$ is transmitted through the same section.

Hence,

$$
\begin{aligned}
{}_2T_3 &= (2)\frac{\Sigma I-(I_1+I_2)}{\Sigma I} - \left\{(R)-(2)\right\}\frac{I_1+I_2}{\Sigma I} \\
&= (2)-(R)\frac{I_1+I_2}{\Sigma I}
\end{aligned}
$$

and similarly for other sections.

On examining actual examples of multicylinder engines it will be found that the distribution of mass moment of inertia along the shaft has a beneficial effect in reducing stress as compared with concentrating it in a heavy fly-wheel.

Examples of Crank-shafts.—By way of contrast Fig. 64 shews two complete 6 throw crank-shafts for a large marine engine and a road vehicle engine respectively. The relative scales are about as 6 : 1. The marine shaft is made of 28 to 32 ton steel of Lloyd's quality, fully built, i.e. with the webs shrunk into the crank-pins and journal pieces. After the pieces are shrunk together, the journals are finish turned in the lathe and the crank-pins are finished with a Moll machine.

The lorry shaft is made of nickel chrome steel of about 55 tons tensile strength, solid forged, heat treated after rough machining. The crank-pins and journals are finished in a grinding machine.

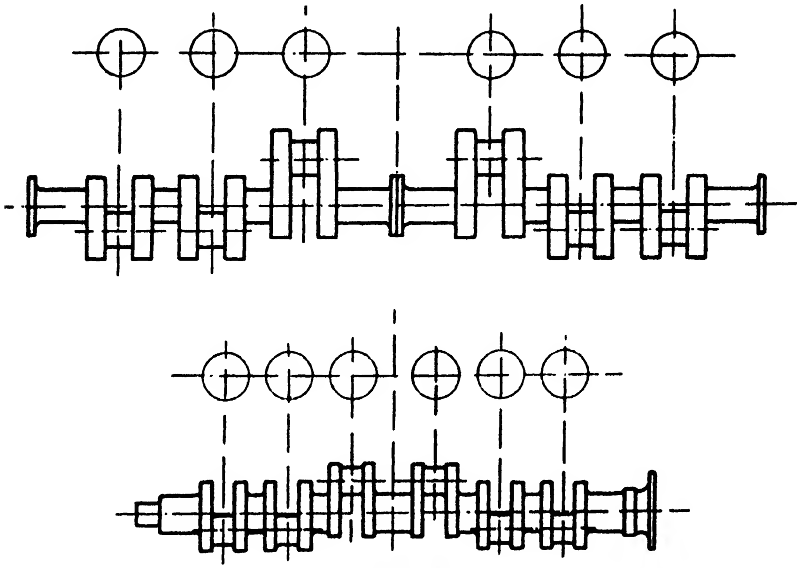


FIG. 64.

Literature.—Dorey, S. F., "Some Factors influencing the Sizes of Crank-shafts for Double Acting Diesel Engines."—N.E. Coast Inst. of Engineers and Shipbuilders.

CHAPTER IX

FLY-WHEELS

The Functions of a Fly-wheel are:—

1. To keep the degree of uniformity within specified limits.
2. Where alternators running in parallel are in question to limit the angular advance or retardation of rotation to a specified fraction of a degree ahead of or behind an imaginary engine rotating with perfectly uniform angular speed.
3. To limit the momentary rise or fall in speed when full load is suddenly thrown off or on.
4. To facilitate starting under compressed air.

In addition to the above the fly-wheel usually serves as a barring or turning wheel and a valve setting disc; also the inertia of the fly-wheel has great influence in determining the critical speed at which torsional oscillations of the crank-shaft are set up.

Fly-wheel Effect.—The fly-wheel effect of a rotating body is its polar moment of inertia (mass × radius of gyration squared) about its axis of rotation. For a fly-wheel or pulley it is found approximately by multiplying the weight of the rim in pounds by the square of the distance in inches from the axis to the centre of gravity of the section of the rim, the result being in in.^{2} lb. units. This underestimates the moment of inertia slightly, and a more accurate method will be described later. The fly-wheel effect of the running gear of one cylinder is found with sufficient accuracy for most purposes by adding the weight of the revolving parts (crank-pin + unbalanced part of two crank webs + 0·65 to 0·5 of the connecting rod) to half the weight of the reciprocating parts (0·35 to 0·5 of the connecting rod + cross-head + piston-rod + piston, etc.), and multiplying the sum by the square of the crank radius.

For a screw propeller the radius of gyration may be taken as 0·36 and 0·42 of the radius for built and solid screws respectively (see p. 203).

Degree of Uniformity.—

$$\text{Degree of uniformity} = \frac{\text{Max. speed} - \text{Min. speed}}{\text{Mean speed}}$$

Let d = Degree of uniformity.
w_1 = Max. angular speed in radians per second.
w_2 = Min. " " " " "

$$\text{Then } d = 2\frac{(w_1 - w_2)}{(w_1 + w_2)} \quad \ldots\ldots \quad (1)$$

For a specified value of "d" the necessary fly-wheel effect is calculated by means of the resultant twisting moment curve of the engine. Let Fig. 65A represent the twisting moment curve and the line A C the mean twisting moment. Let A B C be the loop of largest area (with multi-cylinder engines there are in general as many positive and negative loops in a complete cycle as there are cylinders, and the area of each positive loop is the same as that of each negative loop). If the loop A B C is above the line A C, then the speed of the engine is a minimum at A and a maximum at C, and the increase of rotational energy of the fly-wheel, etc., between A and C is equal to the work represented by the area of the loop A B C.

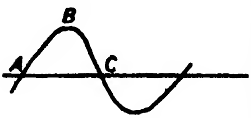


FIG. 65A.

Let A = Area of loop A B C in sq. in. on the diagram.
E = Work represented by A B C in in. lb.
a = Scale to which turning moments are plotted in in. lb. to the inch.
b = Scale to which crank-shaft degrees are plotted in degrees to the inch.

Then $E = \frac{A \times a \times b}{57 \cdot 3}$, 57·3 being the number of degrees in a radian.

Let WK^2 = Fly-wheel effect (moment of inertia) in in.^{2} lb.

Then kinetic energy of wheel $= \frac{WK^2 . w^2}{2g}$ ($g = 386$ in./sec.^{2}).

Change of kinetic energy from A to C

$$= \frac{WK^2}{2g}(w_1^2 - w_2^2) = \frac{WK^2}{2g}(w_1 - w_2)\ (w_1 + w_2) = \frac{WK^2 . d}{4g}(w_1 + w_2)^2$$
$$= WK^2 . d . w^2(\text{MEAN}) \div g$$

if the difference between w_1 and w_2 is small.

But the change of kinetic energy is equal to E.

$$\therefore \quad E = \frac{WK^2}{386}.d.w^2$$

$$\text{and } d = \frac{E \times 386}{WK^2.w^2} \text{ or } WK^2 = \frac{E \times 386}{w^2.d} \ldots\ldots\ldots\ldots\ldots\ldots\ldots (2)$$

Example: Single cylinder engine 10″ bore × 15″ stroke Revs. 300. Turning moment diagram as in Fig. 63, full line. E = 151,000 in. lb. Radius of gyration of wheel 30″. Required to find the weight of the wheel to give a degree of uniformity of 1/80.

$$w = \frac{300 \times 2\pi}{60} = 31{\cdot}4 \text{ radians per sec.}$$

$$W.K^2 = \frac{E \times 386}{w^2.d} = \frac{151{,}000 \times 80 \times 386}{31{\cdot}4^2} = 4{,}730{,}000$$

Fly-wheel effect of running gear $\left(267 + \frac{236}{2}\right) \times 7{\cdot}5^2 = 21{,}600$ in.^{2} lb.

$$WK^2 \text{ for fly-wheel} = 4{,}730{,}000 - 21{,}600 = 4{,}708{,}400 \text{ in.}^2 \text{ lb.}$$

$$\text{but } K = 30''$$

$$\therefore \quad W = \frac{4{,}708{,}400}{30^2} = 5230 \text{ lb.} = 2{\cdot}34 \text{ tons.}$$

Twisting Moment Diagrams for two and four stroke engines having from one to eight cylinders are shewn in Figs. 65 to 67. These have been drawn for an engine 10″ bore by 15″ stroke. As the twisting moments of two engines of different sizes are proportional to the bore^2 × stroke, these curves may be used for engines of any size by multiplying the moments by the bore^2 (in inches^2) × the stroke (in inches) and dividing by 1500. The excess energy represented by the largest loop in each diagram is given in the schedule below for each case.

	Four Stroke Engines.		Two Stroke Engines.	
Number of Cylinders.	E in in. lb. for 10″ × 15″ Cylinder.	E in in. lb. for 1″ × 1″ Cylinder.	E in in. lb. for 10″ × 15″ Cylinder.	E in in. lb. for 1″ × 1″ Cylinder.
1	151,000	101·0	125,500	83·7
2	127,500	84·8	58,500	39·0
3	110,000	73·4	48,700	32·4
4	38,500	25·7	39,000	26·0
6	39,100	26·1	11,150	7·4
8	31,700	21·1	2,200	1·5

Substituting those values of E for a cylinder 1 in. $\times$ 1 in. in equation (2) the following formula is obtained :—

$$WK^2 = \frac{C.B^2.S}{d\left(\frac{n}{100}\right)^2} \qquad \ldots\ldots\ldots\ldots\ldots\ldots \quad (3)$$

Where B = Bore of cylinder in inches.
S = Stroke in inches.
n = Revolutions per minute.
Values of C are given in the following schedule :—

No. of Cylinders.	C for 4 Stroke Engine.	C for 2 Stroke Engine.
1	355	243
2	298	137
3	257	114
4	90	91
6	91	26
8	74	5

Values for "d" used in Diesel Engine Practice.—For certain purposes, as for instance spinning mills, a fine degree of uniformity is desirable, and d = about $\frac{1}{100}$.

For direct coupled continuous current dynamos d = $\frac{1}{80}$ is sufficiently fine to prevent flickering of lights and may be used unless considerations of momentary governing demand a heavier wheel than the use of this figure would give rise to.

For marine engines and land drives, where regularity of turning is not of importance, "d" may be about $\frac{1}{20}$.

The above values for the degree of uniformity must be used with caution, as in a large number of cases (particularly four stroke engines of six cylinders and upwards and two stroke engines of three cylinders and upwards) the considerations discussed in the next article outweigh those of regularity in turning (see also B.S.I. 649/1935).

Momentary Governing.—Under the head of governing it is usually specified that the rise in speed when the load is thrown off suddenly or the fall in speed when the load is suddenly thrown on shall not exceed a certain percentage (usually between 5 and 12) of the mean speed. With air injection the governor has little control over this rise or fall of speed, as at the instant when the load is thrown off sufficient fuel has already been deposited in the pulverisers to carry the engine against full load for a period which may be anything up to

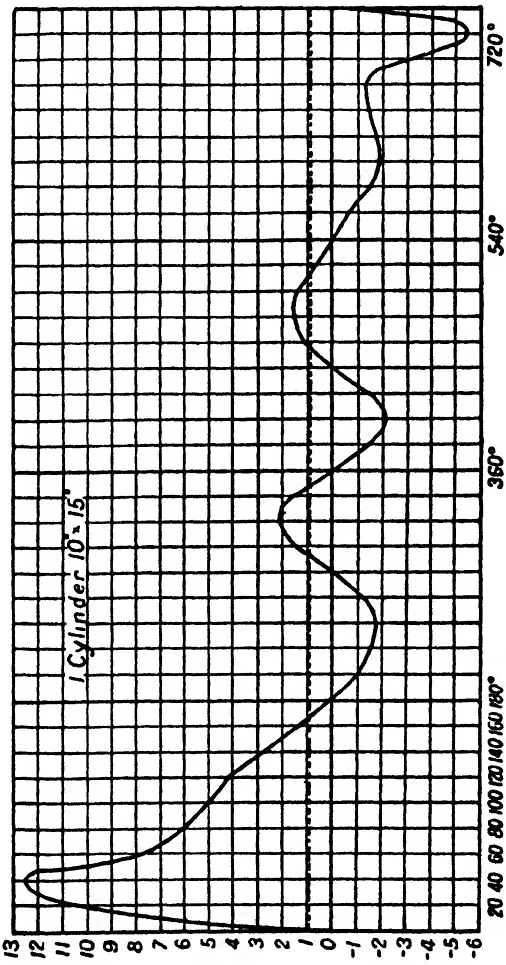


Fig. 65.
FOUR STROKE ENGINE

two revolutions in the case of a four stroke engine. The brake energy developed during this period is entirely devoted to accelerating the fly-wheel and other rotating masses. Owing to the fact that the governor does not act immediately the load is thrown off, the wheel should be capable of absorbing the

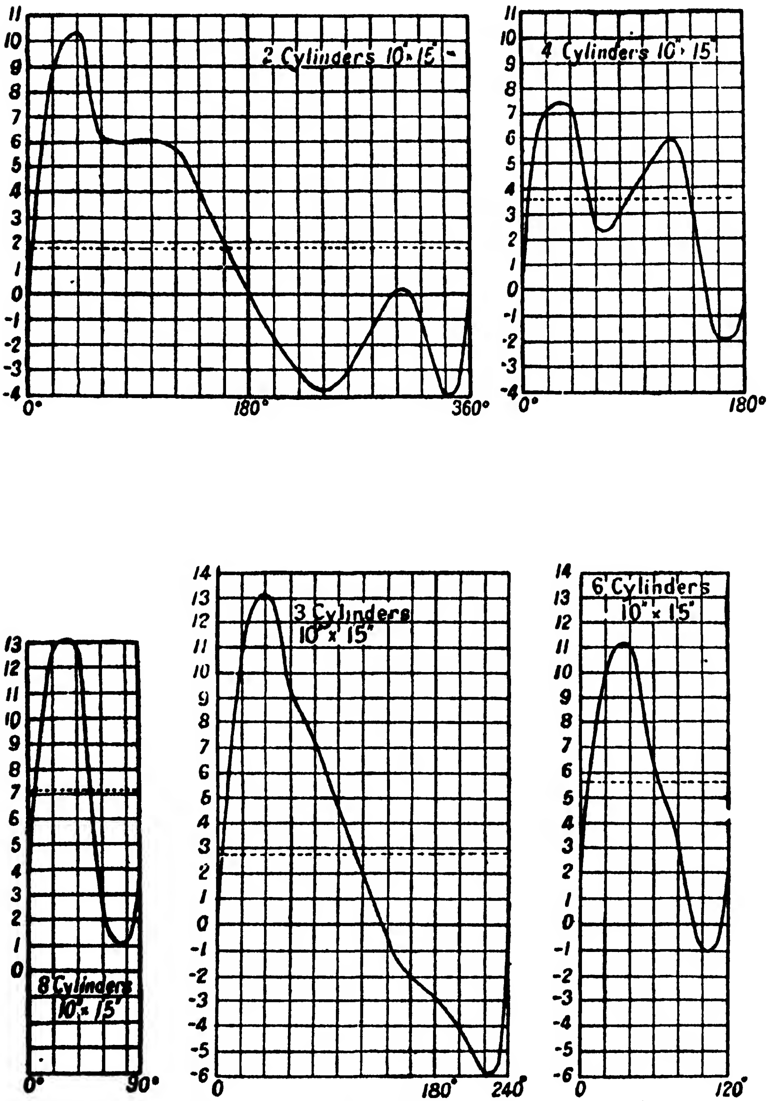


FIG. 66.

FOUR STROKE ENGINES. S.A.

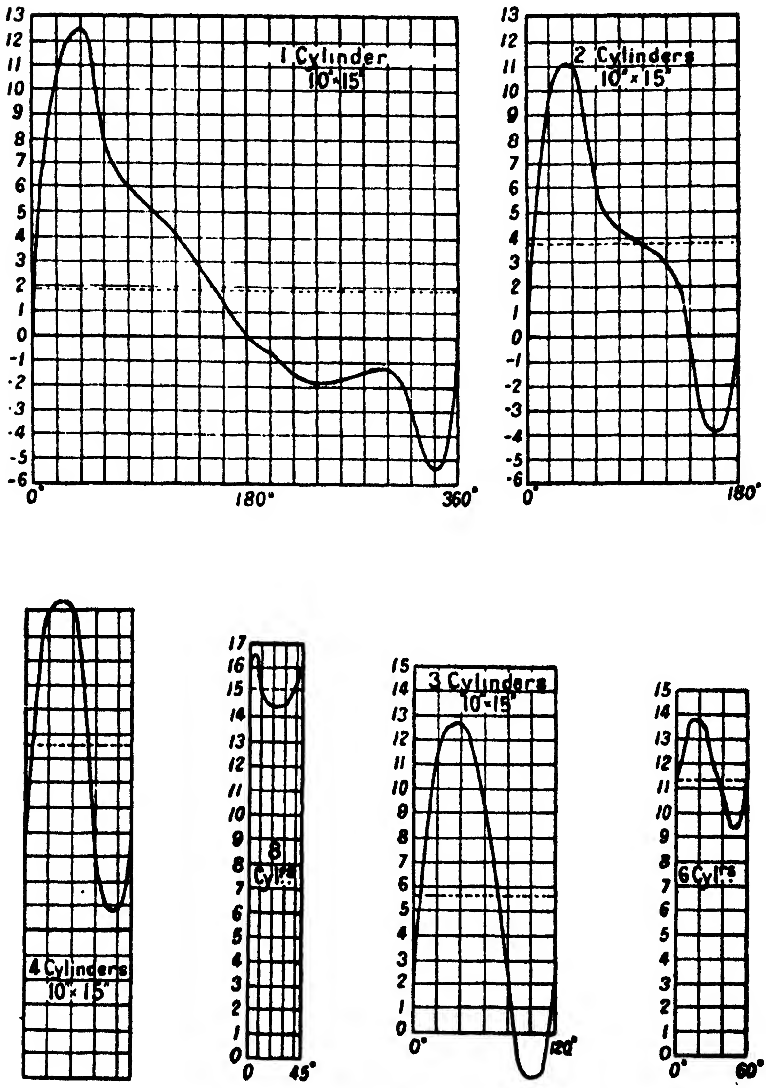


Fig. 67.

TWO STROKE ENGINES. S.A.

whole power of the engine for about three revolutions for a four stroke engine, and 1·5 for a two stroke engine; for airless injection about two revolutions and one respectively.

Example :	B.H.P. of engine (four stroke)	180
	Revolutions per minute .	375
	Momentary rise in speed when full load is suddenly thrown off	12%
	Radius of gyration of wheel .	18 in.

It is required to find the weight of the fly-wheel, neglecting the fly-wheel effect of the running gear.

Work done per revolution at full load $\frac{180 \times 33{,}000 \times 12}{375}$ in. lb.

Energy corresponding to three revolutions

$$=\frac{180 \times 33{,}000 \times 36}{375}=570{,}000 \text{ in. lb.}$$

Angular speed at full load $\frac{2\pi \times 375}{60}=39{\cdot}3$ radians per second.

Momentary angular speed when load is suddenly thrown off

$$39{\cdot}3 \times 1{\cdot}12=44{\cdot}0 \text{ radians per second.}$$

If W=weight of wheel, then :--

$$\frac{W \times 18^2}{2 \times 386}(44^2-39{\cdot}3^2)=570{,}000 \text{ in.}^2 \text{ lb.}$$

and W=3470 lb.

Alternators in Parallel.—When two or more alternator sets are being run in parallel it is a necessary condition for working that they keep almost exactly in phase. Due to inequalities of twisting moment, slight differences of phase inevitably occur, and these give rise to synchronising currents between the various machines, the tendency of these currents being to accelerate the lagging machines and retard the leading ones. This effect keeps the whole system in a state of stability, but cannot be relied on to correct any large fluctuations, and on this account it is usual to specify that the maximum deviation from uniform rotation shall not exceed three electrical degrees on either side of the mean. If the alternator under consideration has a field of two poles only, then the electrical degrees correspond to crank-shaft degrees. In general, if the number of pole pairs is "p," then one crank-shaft degree corresponds to "p" electrical degrees. So far as the engine designer is concerned, then, the problem consists in ascertaining the fly-wheel effect required to keep the cyclic fluctuations on the engine fly-wheel within a certain number of degrees, or more commonly within a certain fraction of a degree of revolution, on either side of the mean.

The method of calculation may be described briefly thus :—

(1) Assume any convenient figure for the fly-wheel effect, e.g. 100,000 in.^{2} lb.
(2) Plot twisting moment curve for complete period, taking the zero of ordinates at the mean twisting moment.
(3) Reduce crank angles to time in seconds, assuming uniform rotation.
(4) Reduce twisting moments to angular acceleration in degrees per second^2 by dividing by the assumed fly-wheel effect and by the acceleration due to gravity (386 in. per sec.^{2}) and multiplying by the number of degrees in a radian (57·3).
(5) Plot angular acceleration to time or crank angle base.
(6) Integrate by planimeter, or otherwise, obtaining angular speed curve.
(7) Integrate again, obtaining angular displacement curve.
(8) Measure maximum deviation from the mean position in degrees.
(9) Increase or decrease the assumed fly-wheel effect in proportion as the angular deviation so found is more or less than the deviation specified. This gives the fly-wheel effect required.

Example : Three cylinder, four stroke engine :—

Bore	20 in.
Stroke	32 ,,
Revolutions per minute . . .	150
Number of pole pairs	20
Angular deviation.	$\pm$ 3 electrical deg.

Twisting moment curve as in Fig. 68. The mean twisting moment being taken as the basis.

The whole calculation is contained in the table below, in conjunction with Figs. 68, 69 and 70.

Since the engine makes 150 revolutions per minute, therefore

$$20^\circ = \frac{20 \times 60}{150 \times 360} = 0{\cdot}0222 \text{ sec.}$$

Assume fly-wheel effect of 10^6 in.^{2} lb. for purposes of calculation. Then :—

$$\text{Acceleration in radians per sec.}^2 = \frac{\text{Twisting moment} \times 386}{10^6}$$

$$\text{in degrees per sec.}^2 = \frac{\text{T.M.} \times 57{\cdot}3 \times 386}{10^6} = \frac{\text{T.M.}}{45{\cdot}3}$$

Referring to the table below :—

Values given in column 2 are scaled off Fig. 68.

" " 3 are obtained by dividing those in column 2 by 45·3.

" " 4 are obtained by multiplying the values in column 3 by 0·0222 sec.

Column 5 is obtained by successive addition of speed increments.

Column 6 contains corrections necessitated by the fact that the resultant of column 5 is not zero, owing to errors.

Column 7 gives corrected speeds which are plotted in Fig. 69.

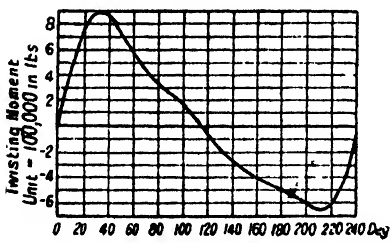


FIG. 68.

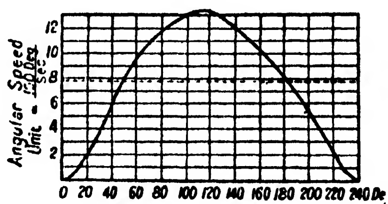


FIG. 69.

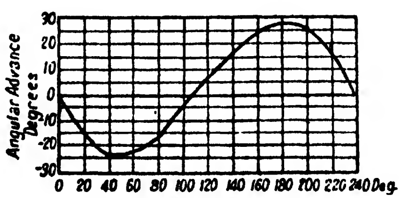


FIG. 70.

Columns 8, 9 and 10 are obtained similarly to columns 2, 4 and 5.

Total swing in phase (see Fig. 70) $24+28=52°$ or $26°$ each side of the mean.

Degrees.	Average Twisting Moment over 20°. In lb. ÷ 1000	Average Acceleration over 20°. Deg./sec.²	Increase or Decrease in Speed. Deg./sec.	Summation of Speed Increments. Deg./sec.	Correction. Deg./sec.	Corrected Speed. Deg./sec.	Average Speed above or below Mean Speed. Deg./sec.	Increase or Decrease of Angle. Deg.	Summation of Angular Advances or Retardations. Deg.
0				0	0	0			0
0–20	440	9710	216				—700	—15·5	
20				—216	9	207			—15·5
20–40	870	19,200	426				—340	—7·5	
40				642	18	624			—23·0
40–60	730	16,200	360				40	0·9	
60				1002	27	975			—22·1
60–80	440	9710	216				300	6·7	
80				1218	35	1183			—15·4
80–100	270	5960	132				480	10·7	
100				1350	44	1306			—4·7
100–120	30	662	15				530	11·8	
120				1365	53	1310			7·1
120–140	—180	—3980	—88				470	10·4	
140				1277	62	1215			17·5
140–160	—340	—7500	—167				330	7·3	
160				1110	71	1039			24·8
160–180	—430	—9480	—211				150	3·3	
180				899	80	819			28·1
180–200	—540	—11,900	—264				—90	—2·0	
200				635	88	547			26·1
200–220	—640	—14,100	—313				—390	—8·7	
220				—322	97	225			17·4
220–240	—440	—9710	—216				—700	—15·5	
240									1·9

Allowable swing, 3 electrical deg. $=\frac{3}{20}=$ crank-shaft deg.

Fly-wheel effect assumed for calculation, 1,000,000 in.^{2} lb.
Therefore fly-wheel effect required

$$=\frac{1{,}000{,}000\times 20\times 26}{3}=173\times 10^6 \text{ in.}^2 \text{ lb.}$$

If radius of gyration of wheel is 65 in.

$$\text{Weight of wheel}=\frac{173\times 10^6}{65^2\times 2240}=18{\cdot}3 \text{ tons.}$$

Allowance for the fly-wheel effect of the alternator rotor would reduce this figure a little.

Precautionary Notes—It seems advisable to add a few words of warning regarding the data given and methods described above. The turning moments and values of excess energy are definitely related to the indicator diagram Fig. 61 and other data for the engine in question. If a different diagram had been used, for example, a supercharged engine diagram, the results would naturally be different.

The shape of the resultant turning moment diagram may be greatly influenced by inertia of the reciprocating parts. In the case of a 4 or 6 cylinder four stroke engine this effect is so pronounced that with a constant indicator diagram there is one speed which gives a minimum degree of irregularity. Reciprocating inertia should be taken into consideration in finding the resultant turning moment of 1, 2 and 3 cylinder two stroke engines and 2, 4, and 6 cylinder four stroke engines.

In any case very fine degrees of irregularity due to small calculated values of excess energy are liable to be exceeded in practice due to unequal firing. A method of allowing for this will be given later.

Calculation by Harmonics.—If the twisting moment diagram for one cylinder is analysed into harmonic components (Fourier Series) the results can be used for calculating degrees of irregularity, etc., for multicylinder engines as described below. If the ignitions follow at equal angles and without simultaneous firing, the resultant twisting moment curve of an N cylinder engine consists of harmonics of order N, 2N, 3N, etc., multiplied by N and superimposed on the mean twisting moment. For an approximate result the harmonics 2N, 3N, etc., may be neglected, leaving N times the Nth harmonic.

Let N = number of cylinders.

h_N = Nth harmonic of turning moment expressed in lb. per in.^{2} of piston area tangential to the crank circle.

The height of one arch of the resultant twisting moment curve is then,

$$Nh_N \cdot \frac{\pi B^2}{4} \cdot \frac{S}{2} = \frac{\pi}{8} Nh_N B^2 S.$$

The mean height of the arch is $2/\pi$th. of this,

$$\text{i.e.} \quad \frac{Nh_N B^2 S}{4}$$

The angle in radians occupied by the arch is $2\pi/2N$, for a two stroke engine and $4\pi/2N$ for a four stroke engine, so the excess energy,

$$\left.\begin{array}{ll} E = \frac{\pi}{4} h_N B^2 S & \ldots\ldots\ldots 2 \text{ cycle} \\ E = \frac{\pi}{2} h_N B^2 S & \ldots\ldots\ldots 4 \quad ,, \end{array}\right\} \ldots\ldots\ldots\ldots\ldots\ldots\ldots (4)$$

(N.B. h_N is not the same for 2 cycle and 4 cycle.)

On the adjoining table values of E and C (equation 3) have been tabulated for four stroke and two stroke single-acting engines and double-acting two stroke engines, having 4 to 12 cylinders. The values may be compared with those given on page 171.

The angular fluctuations can be dealt with similarly.

The angular acceleration for a 2 cycle engine is given by,

$$\ddot{\theta} = \frac{\pi N h_N B^2 S}{8} \cdot \frac{386}{Wk^2} \cdot \sin\left(\frac{N 2\pi n}{60}\right) t$$

where n = revolutions per minute and t = time in seconds.

The amplitude of θ in degrees is given by,

$$\theta^\circ = \pm 57 \cdot 3 \frac{\pi N h_N B^2 S}{8} \cdot \frac{386}{Wk^2} \cdot \left(\frac{60}{N 2\pi n}\right)^2$$

$$\left.\begin{array}{ll} = 79 \dfrac{h_N B^2 S}{N(\frac{n}{100})^2 Wk^2} & \ldots\ldots\ldots 2 \text{ cycle} \\ = 306 \dfrac{h_N B^2 S}{N(\frac{n}{100})^2 Wk^2} & \ldots\ldots 4 \text{ cycle} \\ = \dfrac{D . B^2 S}{(\frac{n}{100})^2 Wk^2} & \ldots\ldots\ldots\ldots\ldots\ldots \end{array}\right\} \ldots\ldots\ldots\ldots\ldots\ldots (5)$$

Values of D are also given in the adjoining table.

	4 Stroke S.A.						2 Stroke S.A.						2 Stroke D.A.					
N cylds.	h_w lb./in.^{2}	E. in. lb.	C	D	F	G	h_w lb./in.^{2}	E. in. lb.	C	D	F	G	h_w lb./in.^{2}	E. in. lb.	C	D	F	G
4	34	53·5	188	2620	1160	3780	37	28·0	98	730	580	1310	58	45·5	16·0	1150	580	1730
5	28	44·0	154	1720	1460	3180	23	18·0	63	360	730	1090	14·5	11·4	4·0	230	730	960
6	25	39·4	138	1280	1750	3030	12	9·4	33	160	880	1040	19	14·9	5·2	250	880	1130
7	20	31·4	110	880	2040	2920	7·1	5·5	19·4	80	1020	1100	8·1	6·3	2·2	91	1020	1110
8	15·5	24·4	85	600	2320	2920	4·7	3·7	13·0	46	1180	1230	9·2	7·2	2·5	91	1180	1270
9	11·0	17·3	61	380	2620	3000	3·1	2·4	8·4	27	1310	1340	3·5	2·8	8·8	31	1310	1340
10	8·5	13·4	47	260	2900	3160	2·4	1·9	6·7	19	1460	1480	4·4	3·4	12·0	35	1460	1500
11	6·3	9·9	35	180	3200	3380	2·1	1·6	5·6	15	1600	1620	4·0	3·1	10·9	29	1600	1630
12	4·6	7·2	25	120	3500	3620	1·8	1·4	4·9	12	1750	1760	2·1	1·6	5·6	14	1750	1770

The additional angular fluctuations due to unequal firing may be calculated on the unfavourable assumption that half the cylinders fire consecutively with mean tangential pressures say 3 lb./in.^{2} for a 2 cycle and $1\frac{1}{2}$ lb./in.^{2} for a 4 cycle engine, above average, and the remaining cylinders the same amounts below average. We consider then a supplementary turning moment diagram of stepped rectangular form; the velocity diagram consists of triangles and the displacement diagram is formed by parabolic arcs. The time occupied by an excursion from the mean position to a positive or negative extreme is one quarter of a cycle, i.e. 15/n seconds (2 cycle) and 30/n (4 cycle). The angular acceleration,

$$=\frac{3N}{2}\left(\frac{\pi}{8}B^2S\right)\left(\frac{386}{Wk^2}\right) \ldots\ldots 2 \text{ cycle}$$

$$=\frac{1{\cdot}5N}{2}\left(\frac{\pi}{8}B^2S\right)\left(\frac{386}{Wk^2}\right) \ldots\ldots 4 \text{ cycle}$$

The angular displacement in degrees $(\pm)=57{\cdot}3\times$ half the angular acceleration $\times$ the square of the time interval,

$$\theta^\circ=\frac{57{\cdot}3}{2}\cdot\frac{3\pi\times 386}{16}\left\{\frac{NB^2S}{Wk^2}\right\}\left(\frac{15}{n}\right)^2 \ldots\ldots\ldots\ldots\ldots\ldots 2 \text{ cycle}$$

$$\theta^\circ=\frac{57{\cdot}3}{2}\cdot\frac{1{\cdot}5\pi\times 386}{16}\left\{\frac{NB^2S}{Wk^2}\right\}\left(\frac{30}{n}\right)^2 \ldots\ldots\ldots\ldots\ldots\ldots 4 \text{ cycle}$$

reducing to,

$$\left.\begin{aligned}
\theta^\circ&=\frac{146NB^2S}{\left(\frac{n}{100}\right)^2 Wk^2}\ldots\ldots\ldots\ldots\ldots\ldots 2 \text{ cycle}\\
\theta^\circ&=\frac{292NB^2S}{\left(\frac{n}{100}\right)^2 Wk^2}\ldots\ldots\ldots\ldots\ldots\ldots 4 \text{ cycle}\\
\theta^\circ&=\frac{F.B^2S}{\left(\frac{n}{100}\right)^2 Wk^2}\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots
\end{aligned}\right\} \ldots\ldots\ldots\ldots (6)$$

where F is a function of N and the cycle (see table).

The table also shews values of G=D+F for the combined effect of cylinder impulses and unequal strength of firing.

To find the Moment of Inertia of a Fly-wheel.—In the first instance, suppose the wheel in question is a disc wheel, i.e. a solid of revolution. Referring to Fig. 71, the thick, full line represents the section of the wheel. Z Z is the axis and S S is a line through the extreme radius of the wheel parallel to the axis at a distance R from the latter. Rule any line A B parallel to the axis, cutting the outline of the section in A and B. Project A and B on to S S at C and D. Join C and D to any convenient point O on the axis, cutting A B in A_1 and B_1. Proceed similarly with different positions of the line A B and join up the various positions of A_1 and B_1, thus obtaining a new figure—the First Derived Figure. Treat this figure as though it were the original figure, and obtain the Second Derived Figure. Similarly with this figure obtaining the Third Derived Figure.

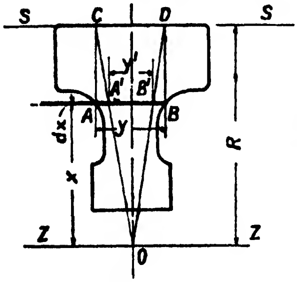


FIG. 71.

Let A = Area of original section in sq. in.
A_1 = Area of First Derived Figure in sq. in.
A_2 = Area of Second Derived Figure in sq. in.
A_3 = Area of Third Derived Figure in sq. in.
w = Weight in lb. of one cubic inch of the material.

Then $A=\int_0^R y.dx$, $A_1=\frac{1}{R}\int_0^R x.y.dx$, $A_2=\frac{1}{R^2}\int_0^R x^2.y.dx$,

$$A_3=\frac{1}{R_3}\int_0^R x^3.y.dx.$$

$$\text{Weight of wheel}=2\pi.w\int_0^R x.y.dx=2\pi.w.R.A_1$$

$$\text{Moment of inertia of wheel}=2\pi w\int_0^R x^3 y.dx=2\pi w\, R^3 A_3$$

$$\text{Radius of gyration}^2=R^2.\frac{A_3}{A_1}$$

The above hold good for any position of the line S S, which may therefore be taken where most convenient. In cases where the section tapers towards the extreme radius (a screw propeller, for instance) the line S S is best located at a distance of about one-half or one-third of the extreme radius from the axis.

In the case of a screw propeller or a fly-wheel with arms, the rotating body must first be reduced to an equivalent disc wheel. This is readily done as follows: Describe a radius R which cuts through the arms or blades, as the case may be. Divide the total area of section at this radius by $2\pi.R$ and the result is the thickness of the equivalent disc at this radius. Repeat for a number of different radii covering the whole range.

Types of Fly-wheels.—Fig. 72 shews a disc wheel cast in one piece and provided with a number of drilled holes in the rim for turning the engine by means of a bar. Degree marks are cut on the edge of the rim to facilitate valve setting. Fig. 73 shews a disc wheel with a separate centre, an arrangement which

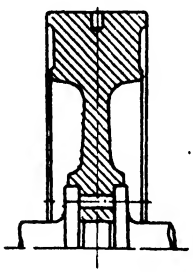


FIG. 72.

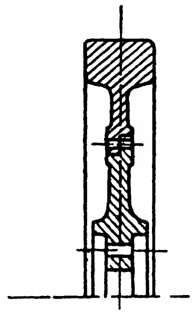


FIG. 73.

makes it easier to obtain a sound casting. Large wheels are usually cast in two pieces, and Fig. 74 shews a design which is suitable for weights up to at least 20 tons. It should be noted that no keys are provided for securing the wheel to the shaft. If the boss of the wheel is bored one-thousandth per inch of

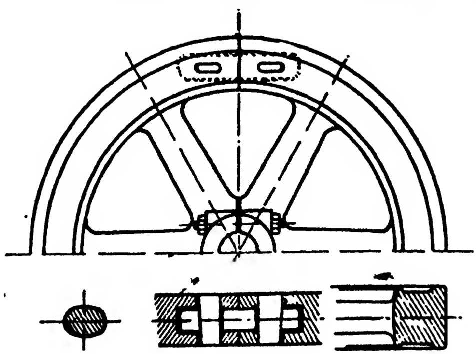


FIG. 74.

diameter less than the shaft and the bolts are drawn up at about the temperature of boiling water the frictional grip is quite sufficient for the largest wheels and the danger of splitting the boss of the wheel involved in the use of keys is avoided. The same applies to pulleys for belt or rope drives. A rather more elaborate wheel, in which greater precautions have been taken, is shewn in Fig. 75. In this case it is advisable to make the bore of the wheel the same as the shaft diameter and to give a shrinking allowance of about one-thousandth per inch of diameter to the bore of the shrunk ring. For large stationary engines some form of barring gear is necessary, and where electric power is available a motor-driven gear is a great convenience. For marine engines a worm, or other self-locking gear, is essential, and where the auxiliaries are electrically driven an electric turning gear should be fitted, as the use of the latter greatly expedites adjustments to the valve gear.

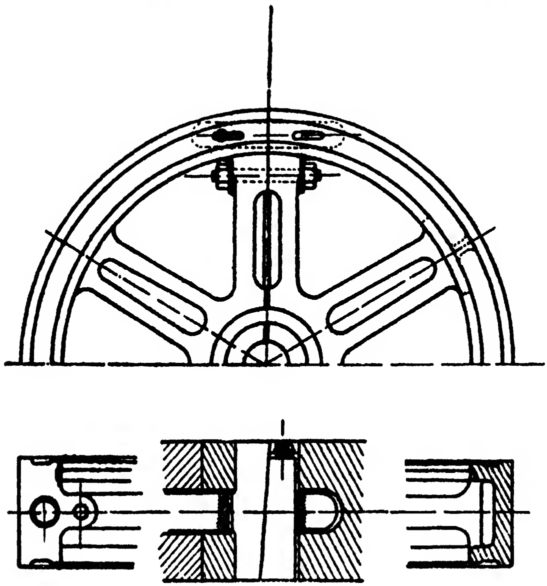


FIG. 75.

Strength of Fly-wheels.—Continental practice favours a peripheral speed of about 100 feet per second for cast iron fly-wheels, and this corresponds to a stress of about 1000 lb. per sq. in. An investigation by Mr. P. H. Smith into the case of a split fly-wheel which burst at Maidenhead in 1912 shewed that the engine (the fly-wheel of which had a normal working peripheral speed of about 100 feet per second) was running about double its normal speed, and as the stress varies as the square of the speed, it follows that the factor of safety under normal conditions was about 4. Destruction tests of wheels and models of wheels shew that the bursting speed is about 200 feet per second for split wheels and 400 feet per second for solid wheels. The discrepancy seems very large and difficult to account for. Average British practice is in favour of a slightly lower peripheral speed (about 90 feet per second). With marine engines considerations of space generally necessitate a still lower figure. The strength calculations for a fly-wheel will be illustrated by an example.

Example: Required to find the approximate dimensions of a fly-wheel suitable for 180 revolutions per minute given that

$$W.K^2 = 40{,}000{,}000 \text{ in.}^2 \text{ lb.}$$

Peripheral speed, 100 ft. per sec.

Maximum twisting moment due to engine, 450,000 in. lb.

Outside radius of wheel $= \dfrac{100 \times 12 \times 60}{2\pi \times 180} = 63{\cdot}7$ in., say 64 in.

Take inside radius of rim $= 52$ in.

Then radius of C.G. of rim section $= 58$ in.

Let $B =$ width of rim. Then :—

$$\text{Weight of rim} = 12 \times B \times 2\pi \times 58 \times 0{\cdot}26$$

And approximate moment of inertia

$$= 12 \times B \times 1{\cdot}64 \times 58^3 = 40{,}000{,}000 \text{ in.}^2/\text{lb.}$$

From which $B = 10{\cdot}5$ in.

Since the stress due to a peripheral speed of 100 ft. per sec. is about 1000 lb./in.^{2}, the total tension at each joint of the rim is equal to $12 \times 10{\cdot}5 \times 1000 = 126{,}000$ lb., for which pull the dowel and cotter section must be designed.

Allowable stress in dowel, say 6000 lb. per sq. in.

Effective area of dowel section $\dfrac{126{,}000}{6000} = 21$ sq. in.

Since about one-third of the section of the dowel is cut away by the cotter hole (see Fig. 76), the gross sectional area of the dowel must be $\dfrac{21 \times 3}{2} = 31{\cdot}5$ sq. in., say 4 in. $\times$ 8 in.

Thickness of cotter $= \dfrac{8}{3} =$ about $2\frac{3}{4}$ in.

Bearing pressure of cotter on dowel

$$\frac{126{,}000}{2{\cdot}75 \times 4} = 11{,}450 \text{ lb. per sq. in.}$$

which is allowable.

If the hole for the dowel is made $\frac{1}{8}$ in. wider than the dowel itself, the bearing length for the cotter on the rim will be $10\cdot5-4\cdot5=6$ in., and bearing pressure of cotter on rim

$$\frac{126{,}000}{2\cdot75\times6}=7650 \text{ lb. per sq. in., also allowable.}$$

Allowable shear stress for cotter (which is in double shear), say 5000 lb. per sq. in.

Depth of cotter $\dfrac{126{,}000}{5000\times2\cdot75}=9\cdot2$ in., say $9\frac{1}{2}$ in. over the rounded ends.

The distance "1" between the inside edge of the cotter hole

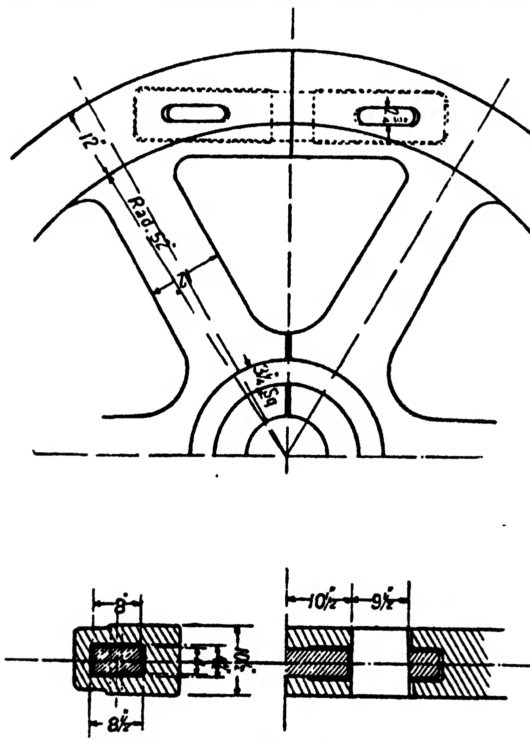


FIG. 76.

and the rim joint must be sufficient to obviate risk of the intervening metal being torn out in double shear. (This point is sometimes overlooked in otherwise well-proportioned rim joints.)

Allowing a shear stress of 1000 lb. per sq. in. :—

$$l=\frac{126,000}{1000\times2\times6}=10\cdot5 \text{ in.}$$

Owing to the difficulty of analysing the straining actions on the arms it is well to give the latter ample proportions.

An approximate method of calculation is given below.

Assuming that the maximum twisting moment due to the engine (450,000 in. lb.) is transmitted to the rim by means of a constant shear force across the arms, and that the bending moment is a maximum at each end of an arm and zero at the centre.

The length of each arm from boss to rims is about 40 in., and the distance of its centre from the centre of the wheel about 32 in.

Then shear force in each arm $=\frac{450,000}{6\times32}=2340$ lb. (assuming six arms).

And maximum bending moment at end of each arm $=2340\times20=46,800$ in. lb.

Taking a low stress of 500 per sq. in., to allow for direct tension in the arms, bending modulus of arm section

$$=\frac{46,800}{500}=93 \text{ in.}^3$$

This is satisfied by a rectangle section 6 in. $\times$ 10 in., which could be replaced by an oval section about 7 in $\times$ 12 in. to reduce wind resistance. The bolts at the hub of the wheel are sometimes made as strong as the rim joint, in which case the core area of two bolts will be the same as the net effective area of one dowel, viz. 21 sq. in.

This gives a bolt of about 4 in. diameter. If shrunk rings are employed these will have a square section about $10\cdot5=(3\frac{1}{4}\text{ in.})^2$.

The above calculations must be regarded as preliminary only, and give the draughtsman a basis on which to start designing. The next step will be to check over the weight and radius of gyration of the complete wheel in the manner already described. The dimensions and stress calculations will then be amended accordingly.

Literature.—For information on the strength of fly-wheels, see :—Unwin, W. C., and Mellanby, A. L., "The Elements of Machine Design," Part II.

CHAPTER X

TORSIONAL VIBRATION

Torsional Vibrations of Crank-shafts.—Volumes have been written on this subject (see list of references, page 215), but a simple method devised by the author can be explained briefly with reference to an example. If there are "n" rotating masses in the system there are (n–1) different modes of free vibration (each with a characteristic frequency) involving 1, 2, 3........(n–1) nodes respectively. Frequently the one noded vibration only is important and this, the simplest case, only will be dealt with here, but the method is fairly general and can be extended to more complicated cases. When there are two masses, for instance a fly-wheel and dynamo rotor each heavy in comparison with the crank masses, then the one noded and the two noded vibrations may be equally important. If the fly-wheel is coupled to a screw propeller by long flexible shafting, then a single node in the crank-shaft system is accompanied by a second node near the propeller, but the amplitude of the crank-shaft vibrations are scarcely affected by the propeller as the amplitude of vibration of the propeller is too small to introduce much damping. Also the frequency is scarcely affected by the shafting and propeller. If the shafting is short the propeller should be included in the calculation along with the crank masses, fly-wheel, etc. The method will be applied to the following data:—

Six cylinder four stroke Diesel Engine. Bore 20″, stroke 40″, crank-shaft dia. 12″, thickness of webs $7\frac{1}{2}$″, thickness of couplings 3″, centres of cylinders (normal) 40″, centres 3 to 4 80″, centre of No. 6 to inside of aft coupling 30″.

Moment of inertia of fly-wheel . . $33{\cdot}0\times10^6$ in.^{2} lb.

„ „ „ each set of crank masses $2{\cdot}2\times10^6$ „ „

(Including $\frac{1}{2}$ recip. mass × crank rad.^{2})

To reduce length to equivalent plain shaft 12″ dia. deduct 30% of web thicknesses and ignore flanges giving :—

Equivalent normal centres, $40-0{\cdot}3\times15$. $=35{\cdot}5''$
,, centres No. 3 to No. 4, $80-6-0{\cdot}3\times15=69{\cdot}5''$
,, No. 6 to fly-wheel $30-0{\cdot}3\times7{\cdot}5$. $=27{\cdot}75''$

To determine the relative amplitudes at the various masses during free vibration only relative masses and relative distances

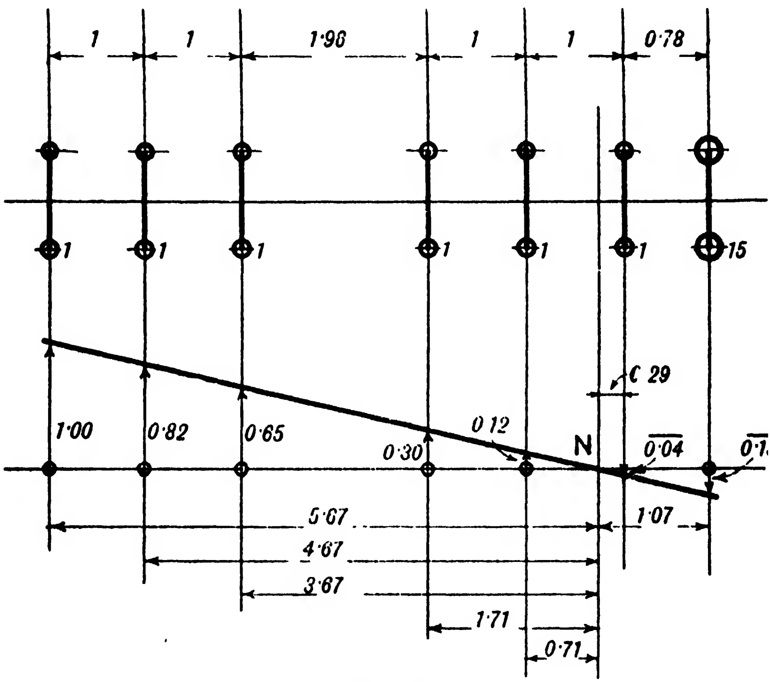


FIG. 77.

are necessary, and these are given in Fig. 77 in terms of a crank mass and normal equivalent centres respectively. Other writers have given graphic and algebraic methods: that adopted here is an arithmetic convergence process given in tabular form on page 193. The first line is a first approximation taking amplitude at cylinder No. 1 equal to unity; node located at C.G. of equivalent masses and vibration form assumed linear. Line 2 gives (amplitudes × relative masses).

Line 3 is obtained from line 2 by chain addition left to right and the figures are proportionate to the relative torques between the masses corresponding to the assumed deflections. Line 4 is obtained by multiplying the figures of line 3 by the relative distances between masses. The figures so obtained are proportional to relative twists between masses. Line 5 is a chain addition of 4 from right to left giving deflection amplitudes relative to the right-hand mass. Adding and dividing by the total relative mass (21) gives deflection of right-hand mass relative to a second approximation to the node (Line 6). Line 7=5−6, amplitudes relative to node. Line 8 the same reduced to unit deflection at cylinder No. 1. This is the second approximation. Lines 9 to 15 give a third and final approximation plotted in Fig. 78.

The frequency of the system is now easily calculated; for a single mass (moment of inertia Wk^2=in.^{2} lbs.) at one end of a steel shaft (C=12,000,000 lb./in.^{2}) of dia. "d", and length "l" (inches) fixed at the other end, the natural frequency is :—

$$F=\frac{204\,.\,d^2}{\sqrt{Wk^2\cdot l\div 10^6}}\sim/\text{min.} \qquad \ldots\ldots (1)$$

For "n" masses the corresponding formula is :—

$$F=\frac{204\,.\,d^2}{\sqrt{\frac{l_1}{a_1}\left(W_1k_1{}^2a_1+W_2k_2{}^2a_2\ldots W_nk_n{}^2a_n\right)\div 10^6}} \qquad \ldots\ldots (2)$$

See Fig. 78.

Applying this to the part of the system lying to the right of the node :—

$$F=\frac{204\times 12^2}{\sqrt{\frac{0\cdot 95\times 35\cdot 5\times 2\cdot 20}{0\cdot 219}\left[(15\times 0\cdot 219)+(1\times\cdot 04)\right]}}=877\,\frac{\sim}{\text{min.}}$$

Similarly to the left of the node :—

$$F=\frac{204\times 12^2}{\sqrt{\frac{0\cdot 83\times 35\cdot 5+2\cdot 20}{0\cdot 192}\left[0\cdot 192+0\cdot 415+0\cdot 79+0\cdot 98+1\cdot 00\right]}}$$

$$=872\,\frac{\sim}{\text{min.}}$$

checking within 1%, mean, say, F=875.

The calculation of vibration stresses due to synchronising

Line.	1		2		3		4		5		6		Fly-wheel.	
					Cranks.									
1	1·00	—	0·82	—	0·65	—	0·30	—	0·12	—	0·04	—	0·19	1st approx.
2	1·00	—	0·82	—	0·65	—	0·30	—	0·12	—	0·04	—	2·85	
3	—	1·00	—	1·82	—	2·47	—	2·77	—	2·89	—	2·85	—	
4	—	1·00	—	1·82	—	4·85	—	2·77	—	2·89	—	2·22	—	
5	15·55	—	14·55	—	12·73	—	7·88	—	5·11	—	2·22	—	0	$\Sigma=58{\cdot}04$
6	2·76	—	2·76	—	2·76	—	2·76	—	2·76	—	2·76	—	—	$=58\div21$
7	12·79	—	11·79	—	9·97	—	5·12	—	2·35	—	0·54	—	2·76	$(\div12{\cdot}79)$
8	1·00	—	0·92	—	0·78	—	0·40	—	0·18	—	0·04	—	0·216	2nd approx.
9	1·00	—	0·92	—	0·78	—	0·40	—	0·18	—	0·04	—	3·24	
10	—	1·00	—	1·92	—	2·70	—	3·10	—	3·28	—	3·24	—	
11	—	1·00	—	1·92	—	5·30	—	3·10	—	3·28	—	2·52	—	
12	17·12	—	16·12	—	14·20	—	8·90	—	5·80	—	2·52	—	0	$\Sigma=64{\cdot}66$
13	3·08	—	3·08	—	3·08	—	3·08	—	3·08	—	3·08	—	3·08	$=64{\cdot}7\div21$
14	14·04	—	13·04	—	11·12	—	5·82	—	2·72	—	0·56	—	3·08	
15	1·00	—	0·93	—	0·79	—	0·415	—	0·192	—	0·04	—	0·219	3rd approx.

of harmonics with this natural frequency is given in the annexed table :—

1.	2.	3.	4.	5.	6.
Order of Harmonic. i	Critical speed R.P.M.	Value of Harmonic. Hi lb. sq. in.	$\Sigma[a]_n$	Equilibrium stress f_e lb./in.^{2}	Vibration stress f_v
6	292	25·0	3·29	1,690	25,000
7	350	20·0	0·55	226	13,500
8	219	15·5	0·22	70	8,000
9	195	11·0	2·15	490	17,300
10	175	8·5	0·22	39	4,500
11	159	6·3	0·55	71	8,200
12	146	4·6	3·29	311	15,100
13	135	3·5	0·55	40	4,600
14	125	2·8	0·22	13	1,600
15	117	2·2	2·15	98	10,200
16	110	1·7	0·22	7·7	800
17	103	1·4	0·55	16	1,900
18	97	1·1	3·29	74	8,500

Column 1 gives orders of harmonics (i) from 6 to 18 in terms of a cycle (two revolutions in the case of a four stroke engine) as the fundamental period. Column 2 gives the revolutions per minute at which these harmonics become critical $=2F/i$ $=1750/i$. Column 3 gives the amplitudes of the harmonics of applied torque per cylinder due to gas pressure at M.I.P. of 100 lb./sq. in. and are expressed in lb. per sq. in. of piston area acting tangentially at the crank circle. The values have been read off curves given by Lewis (see p. 215).

Column 4 gives the resultant $\Sigma[a]_n$ of a vector sum of deflections taken from the vibration form (Fig. 78), the deflections being set out in correct phase relationship in accordance with the scheme set out in Fig. 79. For Harmonics of orders, 1, 7, 13, 19, etc., the phase relationship is that of the cams round the cam-shaft. For Harmonics of orders, 2, 8, 14, etc., the angles are doubled and so on. Column 5 gives "equilibrium stresses" calculated from the formula :—

$$f_e = 20{\cdot}6\ \Sigma[a]_n \cdot H_i \quad \ldots\ldots \quad (3)$$

Finally Column 6 gives the vibration stress which is related to the equilibrium stress by a relation which can be expressed graphically by plotting the values given (see Fig. 80).

The derivation of equation 3 has now to be explained. Fig. 78 under shows a vibration form. Suppose the shaft is rigidly held at the node and that torques T_1 T_2 T_3....are slowly applied to the points indicated. Also suppose that the deflections a_1 a_2 a_3, etc., are so regulated (by gearing, for example)

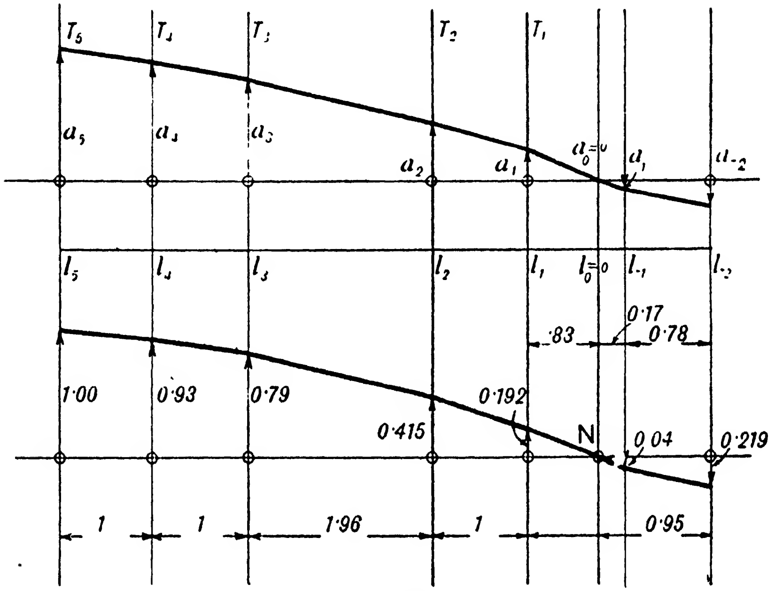


FIG. 78.

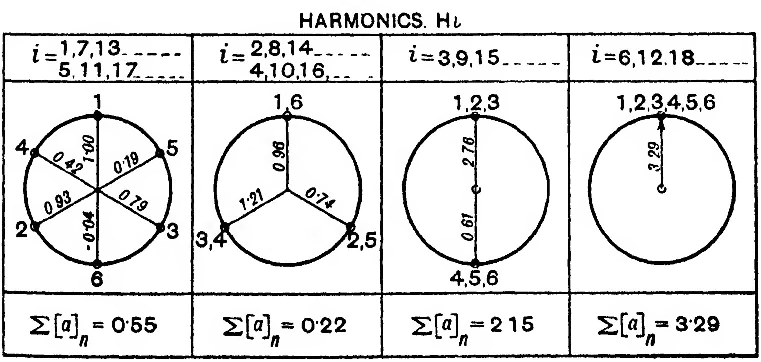


FIG. 79.

that the deflections retain their relative magnitudes whatever may be the relative magnitudes of T_1, T_2, T_3, etc. Let the twisting modulus of section of the shaft $=Z$. Then it can be shewn by the principle of work that the stress at the node in these circumstances is given by :—

$$f_e=\frac{\Sigma T_n \cdot a_n}{Z}. \times \frac{I}{\Sigma \dfrac{(a_n-a_{n-1})^2}{l_n-l_{n-1}}} \times \frac{a_1-a_0}{l_1-l_0}$$

This is the "equilibrium stress," and the relevance of this conception depends upon the fact that at resonance (i.e. at a critical speed) the vibration form is practically the same as that of a free vibration whatever the magnitudes of the applied forces.

But,

$$\Sigma T_n \cdot a_n=\Sigma[a]_n \cdot H_l \cdot A \cdot R \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots (5)$$

where, A=Piston area sq. in.
R=Crank radius in.

$$\therefore f_e=\frac{\Sigma\,[a]_n.H_l.A.R}{Z} \times \frac{1}{\Sigma \dfrac{(a_n-a_{n-1})^2}{l_n-l_{n-1}}} \times \frac{a_1-a_0}{l_1-l_0} \quad \ldots\ldots\ldots (6)$$

Now, $A=0{\cdot}785\times20^2=314$ in.^{2}, $R=20$ in., $Z=\pi 12^3/16=340$ in.^{3}

$$\frac{a_1-a_0}{l_1-l_0}=\frac{0{\cdot}192}{0{\cdot}83}=0{\cdot}232$$

$$\Sigma\frac{(a_n-a_{n-1})^2}{(l_n-l_{n-1})}=\frac{0{\cdot}180^2}{0{\cdot}78}+\frac{0{\cdot}04^2}{0{\cdot}174}+\frac{0{\cdot}192^2}{0{\cdot}83}+\frac{0{\cdot}223^2}{0{\cdot}83}+\frac{0{\cdot}223^2}{1}+\frac{0{\cdot}275^2}{1{\cdot}96}$$
$$+\frac{0{\cdot}14^2}{1}+\frac{0{\cdot}07^2}{1}=0{\cdot}208$$

$$\therefore f_e=\Sigma[a]_n \,.\, H_l \,.\left(\frac{0{\cdot}232}{0{\cdot}208} \times \frac{314}{340} \times 20\right)=20{\cdot}6\ \Sigma[a]_n \,.\, Hi \,.$$

which is equation 3.

The relation—vibration stress/equilibrium stress is the "resonance factor" which depends partly on the internal work done in the shaft itself on account of elastic hysteresis. The relation given by Fig. 80 has been deduced from torsiograph measurements. For vibration stresses up to about 5000 lb./sq. in. the resonance factor is nearly constant at about 80, but decreases as the vibration stress is increased.

The expedients available for removing dangerous critical speeds from the working range are :—

1. Altering the distribution of mass moment of inertia.
2. Altering the order of firing.
3. Altering the diameter of the shaft.

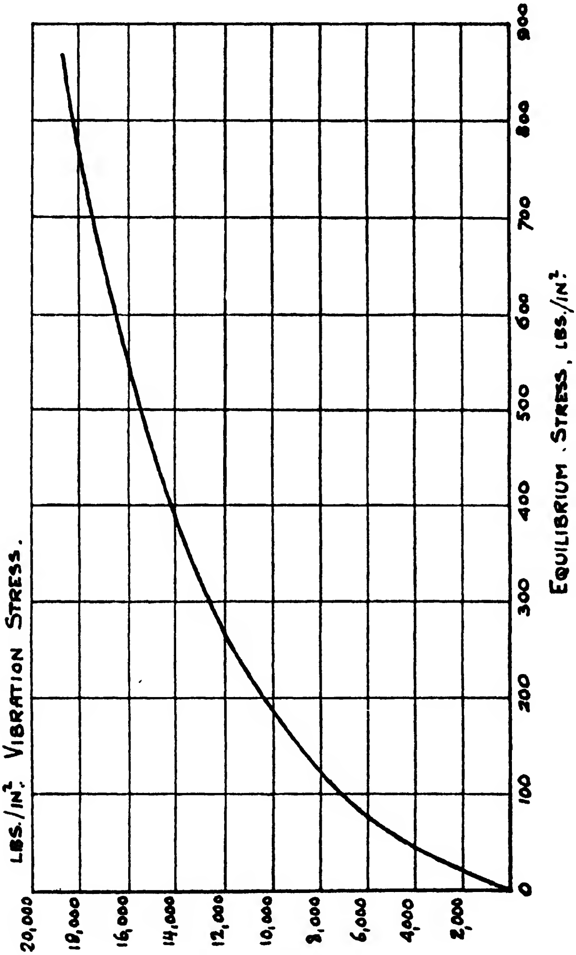


Fig. 80.

When the working range is a wide one it may be impossible to avoid dangerous critical speeds, and in this case a vibration damper as, for example, that devised by Lanchester, may be used.

Critical Speed of Marine Tunnel Shafting—As a second example suppose that the engine referred to in the previous section is coupled to a screw propeller 12 ft. dia. weighing 2·8 tons by means of a thrust shaft 8 ft. long by $9\frac{1}{2}$ in. dia., a tunnel shaft 60 ft. long (neglecting coupling widths) by 9 in. dia., and a propeller shaft 15 ft. long by $9\frac{3}{4}$ in. diameter. B.H.P. 800 @ 120 R.P.M. (full speed).

It is sufficiently accurate to regard the engine as a single mass of $WK^2 = 10^6\,(33 + 6 \times 2{\cdot}2)$

$$= 10^6 \times 46{\cdot}2 \text{ in.}^2 \text{ lb}$$

located at the centre of mean position of the fly-wheel and crank-mass moments of inertia.

The equivalent distance from the centre of the engine to the fly-wheel (see Fig. 77) is,

$$35{\cdot}5\,(0{\cdot}98 + 1 + 1 + 0{\cdot}78) = 133 \text{ in.}$$

The distance from the fly-wheel to the position of the equivalent mass is then :—

$$133 \times \frac{6 \times 2{\cdot}2}{33{\cdot}0 + (6 \times 2{\cdot}2)} = 133 \frac{6}{15 + 6} = 38 \text{ in.}$$

We now reduce the shafting between the equivalent engine mass and the propeller to an equivalent uniform 9″ shaft of equal flexibility as under :—

Crank-shaft (part of)	$38\left(\frac{9}{12}\right)^4$	$= 12$ in.
Thrust shaft	$8 \times 12\left(\frac{9}{9{\cdot}5}\right)^4$	$= 78$ in.
Tunnel shaft	60×12	$= 720$ in.
Propeller shaft	$15 \times 12\left(\frac{9}{9{\cdot}75}\right)^4$	$= 130$ in.
Total length of equivalent 9 in. shaft . .		940 in.

The radius of gyration of a solid propeller is usually about 0·41 of the extreme radius. Experience shews that it is necessary to add about 25% to the actual weight of the propeller to allow for the mass of water associated with the motion of the propeller ; hence :—

$$WK^2 \text{ of propeller} = 1{\cdot}25 \times 2{\cdot}8 \times 2240\left(\frac{0{\cdot}41 \times 12 \times 12}{2}\right)^2$$

$$= 6{\cdot}85 \times 10^6 \text{ in.}^2 \text{ lb.}$$

We have now a system consisting of two masses connected by 940 in. of 9-in. shafting. The distance of the node from the propeller :—

$$=940\frac{46{\cdot}2}{46{\cdot}2+6{\cdot}85}=820 \text{ in.}$$

By formula (1) the natural frequency is,

$$\sqrt{\frac{204\times 9^2}{6{\cdot}85\times 820}}=221 \text{ /min.}$$

Since the 6 cylinder four stroke engine gives three main impulses per minute the principal critical speed will occur at

$$221\div 3=74 \text{ R.P.M.}$$

i.e. at about 60% of full revolutions.

This is quite a usual and safe position for the critical speed of a marine engine arranged amidships.

Fig. 81 shews the approximate variation of torque in the tunnel shaft at various revolutions. The mean torque at full power :—

$$=\frac{800\times 33{,}000\times 12}{2\pi\times 120}=420{,}000 \text{ lb. in.}$$

The mean torque at other revolutions varies roughly as the $\left(\frac{\text{R.P.M.}}{120}\right)^2$

The cyclical fluctuation of engine torque requires separate investigation but will be of the order $=\pm 400{,}000$ in. lb. At slow speeds a fraction of this, viz. $6{\cdot}85\div(6{\cdot}85+46{\cdot}2)$, i.e. $12{\cdot}9\%$ or $=52{,}000$ in. lb. only will be transmitted to the propeller, the remainder being absorbed in accelerating and decelerating the engine masses, the fly-wheel in particular. At speeds not too close to the critical, the fluctuation of torque in the tunnel shafting will be given approximately by the formula,

$$\pm 52{,}000\left(\frac{1}{1-\left(\frac{\text{R.P.M.}}{74}\right)^2}\right)$$

At the critical speed itself, the fluctuation of torque in the tunnel shaft is limited by propeller damping and in circumstances similar to those postulated in this example, torsiograph measurements indicate that it may amount to about $\pm 600{,}000$ in. lb.

There is usually no special inconvenience involved in avoiding the critical speed in normal running.

It is easy to see that with long shafting as in this example, the propeller has little influence on the frequency or magnitude of the crank-shaft vibration considered in the previous article. It will be recalled that the frequency of these vibrations is

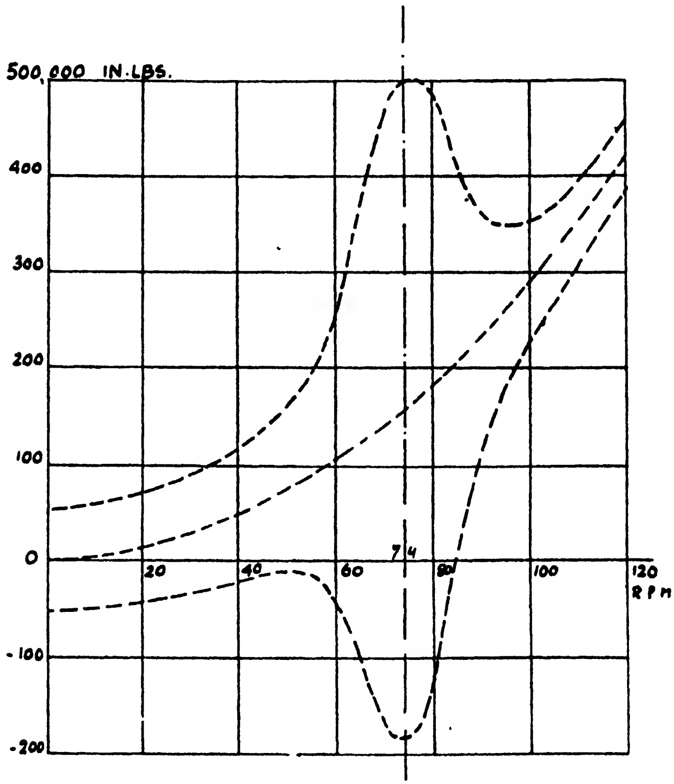


FIG. 81.

875∼/min. The vibration form for the complete system will be as shewn in Fig. 82. Let l = distance of the second node from the propeller; then approximately,

$$875=\frac{204\times 9^2}{\sqrt{6\cdot 85\times l}}$$

from which l=52·5 inches, less than 6% of the total shaft length. With these proportions a severe vibration stress in the crank-shaft is associated with a negligible stress in the tunnel shaft.

When Marine Diesel Engines are arranged aft as in tankers it is generally not satisfactory to deal separately with the crank-shaft and tunnel shaft vibration; it is desirable to

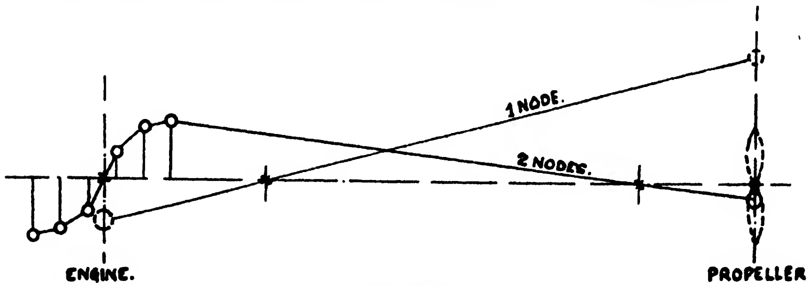


FIG. 82.

consider the system as a whole with respect to one and two noded vibrations. It is usually necessary to make the intermediate shafting above the requirements of Lloyd's rules in order to raise the chief critical speed above the working range.

Reduction of Shafting.—We proceed to consider in greater detail some of the problems of torsional vibration calculations. As indicated above it is convenient to reduce all length of shafting to an equivalent uniform shaft of same standard diameter. If a stepped shaft consists of lengths l_1, l_2, l_3.... having diameters d_1, d_2, d_3....then the equivalent length l_0 of uniform shaft of diameter d_0 is given by :—

$$l_0 = \Sigma l_n \left(\frac{d_0}{d}\right)^4 \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (4)$$

since a shaft of length l_0 and diameter d_0 will twist through the same angle under a given torque as the stepped shaft. If at the end of the calculation it is desired to calculate the stress at any section of the shaft from the calculated torque at that section, the modulus of the actual shaft section must be used.

A shaft with a straight taper from smaller diameter d_1, to larger diameter d_2 is equivalent to a uniform shaft of the same length having a diameter d_3 given by the following table :—

$\frac{d_2}{d_1}=1$	1·1	1·2	1·3	1·4	1·5	1·6	1·7	1·8	1·9	2·0
$\frac{d_3}{d_2}=1$	0·95	0·91	0·87	0·84	0·81	0·78	0·75	0·73	0·70	0·68

With regard to bolted flange couplings the writer's practice is to ignore the flanges (i.e. treat them as infinitely rigid) and to ignore the fillets measuring the plain shaft diameter as though it extended to the rear face of the coupling. Any errors so involved are usually small and act in contrary directions.

Cases of uncertainty arise in connection with shafts keyed or shrunk into relatively rigid bosses. The only general rule which can be given is to split the difference between the limits of greatest and least rigidity which appear possible in the circumstances. The equivalent length of a crank-shaft from one cylinder centre to the next is a fundamental quantity in calculating the torsional vibration of crank-shafts. If the journal and crank-pins are of the same diameter and the webs conform to Lloyd's rules for built up shafts the writer finds that the equivalent length may be taken as the actual length minus 30% of the length occupied by the webs. This simple rule has given excellent results in a large number of instances of marine main and auxiliary engines and stationary engines. Flange couplings and radii are ignored. Axial holes for forced lubrication if less than one-third the shaft diameter also ignored except in so far as they affect the mass of the crank-pin. Several formulæ have been given by different writers* for calculating the equivalent length of a solid forged crank and sometimes the result is in excess of the actual length in contrast with the above. Such cases appear to arise when the webs are relatively weak compared with the journals and pins. If the webs are in accordance with Lloyd's rules for built up shafts, but journals and pins have been increased beyond Lloyd's requirements for the sake of torsional stiffness or otherwise, the above rule can be applied after first reducing the lengths of crank-pins and journals to their equivalents of Lloyd's diameter and finally reducing to the actual diameter again.

For example: if the crank-shaft referred to on page 190 is increased in diameter from 12″ to 15″ (the webs being unchanged) the equivalent length of one crank-pin and journal reduced from 15″ to 12″ is

$$(40''-15'')\left(\frac{12}{15}\right)^4=10{\cdot}2''$$

to which add 70% of 15″ = 10·4″

making total 20·6″

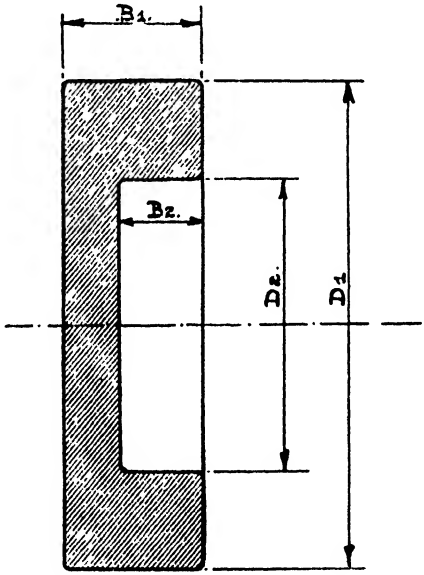
* Geiger, Constant, Carter, Jackson and others.

Reduced to 15″ dia. again this becomes :—

$$20{\cdot}6\left(\frac{15}{12}\right)^4 = 49{\cdot}5$$

which is 1·24 times the actual centres.

Calculation of Mass Moments of Inertia.—A graphical method has already been given (p. 184) for finding the moment of inertia of a solid of revolution. A convenient formula for cast iron or cast steel disc wheels is given in Fig. 83.



$$\text{Weight} = \begin{matrix}0{\cdot}204 \text{ (cast iron)} \\ 0{\cdot}221 \text{ (steel)}\end{matrix} \left\{B_1D_1^2 - B_2D_2^2\right\}$$

$$\begin{matrix}\text{Moment} \\ \text{of inertia}\end{matrix} = \begin{matrix}0{\cdot}0255 \text{ (cast iron)} \\ 0{\cdot}0276 \text{ (steel)}\end{matrix} \left\{B_1D_1^4 - B_2D_2^4\right\}$$

FIG. 83.

Screw propellers with elliptical blades may be calculated approximately for the following data :—

	Ratio of $\dfrac{\text{Radius of Gyration}}{\text{External Radius.}}$
Built propellers	0·35 to 0·37
Solid propellers	0·41 to 0·43

Add 25% to actual weight to allow for effect of water.

For convenience in calculating crank-pins, crank-webs, balance weights, etc., the radii of gyration[2] and moments of inertia of a number of useful shapes are tabulated in Fig. 84.

Connecting rods are dealt with by dividing into the conventional revolving and reciprocating parts as in balancing problems. The contribution of the revolving part to the moment of inertia is taken as the revolving mass × (crank radius)^2. The contribution of the total reciprocating mass,

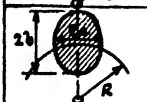
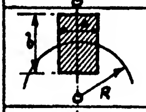
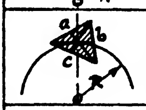
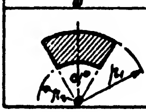
FIGURE.	AREA A.	K^2 ABOUT O.	AK^2.
	$\pi(r_1^2-r_2^2)$	$R^2+\frac{r_1^2+r_2^2}{2}$	$\pi\left\{\frac{r_1^4-r_2^4}{2}+R^2(r_1^2-r_2^2)\right\}$
	πab	$R^2+\frac{a^2+b^2}{4}$	$\pi ab\left\{R^2+\frac{a^2+b^2}{4}\right\}$
	ab	$R^2+\frac{a^2+b^2}{12}$	$ab\left\{R^2+\frac{a^2+b^2}{12}\right\}$
	$\sqrt{s(s-a)(s-b)(s-c)}$ $s=\frac{a+b+c}{2}$	$R^2+\frac{a^2+b^2+c^2}{36}$	$\sqrt{s(s-a)(s-b)(s-c)}\left\{R^2+\frac{a^2+b^2+c^2}{36}\right\}$
	$\frac{\pi\alpha}{360}(r_1^2-r_2^2)$	$\frac{r_1^2+r_2^2}{2}$	$\frac{\pi\alpha}{360}\left\{\frac{r_1^4-r_2^4}{2}\right\}$

FIG. 84.

consisting of piston, piston rod and crosshead (if any) + the reciprocating part of the connecting rod, is taken as *half* the total reciprocating mass × (crank radius)^2. This takes account of the fact that at the top and bottom dead points, the reciprocating parts contribute nothing to the moment of inertia, whereas in the neighbourhood of half-stroke, up or down, the whole reciprocating mass is equivalent to an equal mass concentrated at the crank-pin. The actual moment of inertia at each crank fluctuates to this extent.

Any revolving masses connected to the main shaft by gearing so as to revolve " N " times as fast as the main shaft, are to be multiplied by the factor N^2 to reduce them to equivalent masses revolving with the main shaft. Likewise any masses reciprocating with amplitude (semi-range) m times the crank radius and frequency "n" cycles per revolution of the main shaft are to be multiplied by $\frac{1}{2}n^2m^2$ to give the equivalent average moment of inertia referred to the main shaft.

Vibration Forms.—An arithmetical method of finding the relative displacements of the elements of a system oscillating

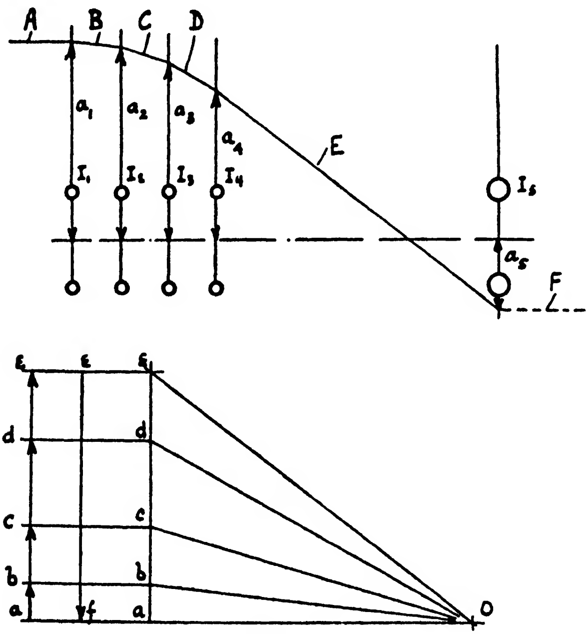


FIG. 85.

about a single node has already been given. A graphical construction valid for one or more nodes and known as Gumpels method is shewn in Fig. 85. The pole o is found by trial; heights ab, bc, cd, etc., are proportional respectively to I_1a_1, I_2a_2, I_3a_3, etc., with due regard to + and − signs of a_1, a_2, etc. Lines oa, ob, oc, etc., are parallel to lines A B C, etc. The pole is correctly chosen when the construction concludes with a horizontal line (shewn dotted). It is preferable to start the construction with a_1=unity to same convenient scale and to work with relative values for I_1, etc.

The vibration form for two masses is a straight line, the deflections of the end points being inversely as the corresponding masses.

The two noded vibration form for 3 masses is easily found by trial from the relations (see Fig. 86),

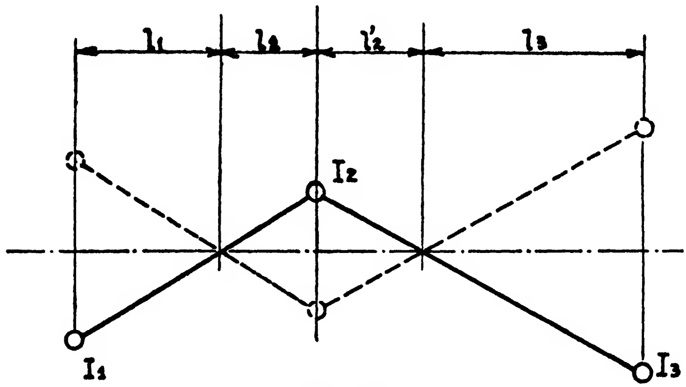


Fig. 86.

$$I_1 l_1 = I_2 \div \left(\frac{1}{l_2} + \frac{1}{l^1_2}\right) = I_3 l_3$$

in which $I_1 = W_1 K_1{}^2$, $I_2 = W_2 K_2{}^2$, $I_3 = W_3 K_3{}^2$

guess l_1 then $l_3 = l_1 \frac{I_1}{I_3}$; evaluate,

$I_2 \div \left(\frac{1}{l_2} + \frac{1}{l^1_2}\right)$ and compare with $I_1 l_1$; repeat with new value of l_1 until equality is obtained.

Systems with equal and equally space masses are readily calculated and some results are collected in Fig. 87. These results refer in each case to systems anchored at one end, but they apply equally to symmetrical systems having double the number of masses or systems in which the masses shewn are balanced by another system having the same value of ΣIa. For example, the two mass system shewn is immediately applicable to the 10 mass system with 2 nodes indicated in Fig. 88. Against each system is written :

(A) The fraction of the total mass (ΣI) which if concentrated as a single mass at the C.G. of the given masses will give a single mass system of the same frequency.

(B) The relative frequency of the systems shewn.
(C) The reciprocal of B, i.e. the relative periodic time.

It will be noted that C tends ultimately to be directly proportional to the number of masses. Also that for 3 or more masses (A) differs little from 0·82.

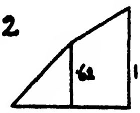
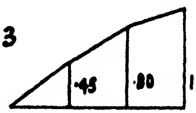
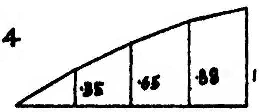
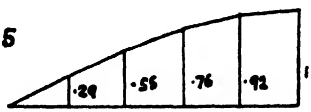
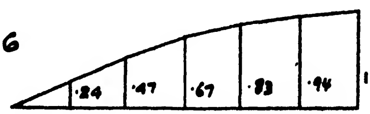
	A	B	C
2 	0·87	0·525	1·40
3 	0·84	0·264	2·75
4 	0·83	0·276	3·62
5 	0·82	0·222	4·50
6 	0·82	0·184	5·42

FIG. 87.

Applying this result to the data of page 191 we replace the crank masses by a single mass,

$$0{\cdot}82 \times 6 \times 2{\cdot}2 \times 10^6 = 10{\cdot}8 \times 10^6 \text{ in.}^2 \text{ lb.}$$

The fly-wheel moment of inertia is 33×10^6 in.^{2} lb., and the distance from the engine centre to the fly-wheel $= 0{\cdot}98 + 2 + 0{\cdot}78 = 3{\cdot}76$ units of 35·5″.

The distance of engine centre to node is therefore

$$3{\cdot}76\left(\frac{33}{33+10{\cdot}8}\right)=2{\cdot}83 \text{ units.}$$

According to Fig. 78 this distance is

$$0{\cdot}98+1+0{\cdot}83=2{\cdot}81 \text{ units,}$$

in close agreement.

The frequency is found to be

$$\frac{204\times 12^2}{\sqrt{2{\cdot}83\times 35{\cdot}5\times 10{\cdot}8}}=893\sim/\text{min.}$$

within 2% of the previous result.

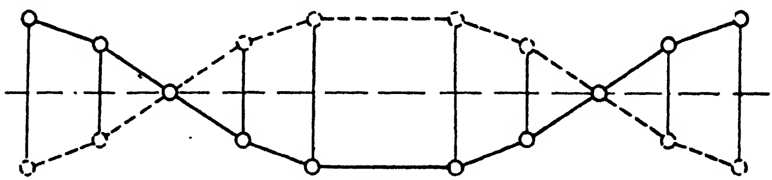


FIG. 88.

The Natural Frequency.—Having determined the vibration form by one or other of the methods described the natural frequency is readily determined by a formulæ for the natural frequency of a part system to one side of a node.

One such formula (formula 2, with reference to Fig. 78) has already been given and follows immediately from consideration of the magnitude and deflection of a mass on the other side of the node capable of equilibrating the given masses. The formula suffers from the defect that the quantities l_1 and a_1 are frequently small and therefore not determined to a high degree of accuracy.

An equivalent formula is as follows :—

$$F=\frac{204\ d^2}{\sqrt{\dfrac{W_1k_1{}^2a_1l_1+W_2k_2{}^2a_2l_2\ldots\ldots W_nk_n{}^2a_nl_n}{a_n}}}$$

Applying this to the previous data we obtain :—

$$F=\frac{204\times 12^2}{\sqrt{\dfrac{2{\cdot}2\times 35{\cdot}5}{1}\left(\begin{array}{c}5{\cdot}79\times 1{\cdot}00+4{\cdot}79\times 0{\cdot}93+3{\cdot}79\times 0{\cdot}79\\ +1{\cdot}83\times 0{\cdot}415+0{\cdot}83\times 0{\cdot}192\end{array}\right)}}$$

$$=889\sim/\text{min.}$$

The Harmonic Components of Torque.—Each cylinder gives rise to a variable applied torque which repeats itself for every cycle of operations, i.e. once per revolution with 2 cycle engines, and once every two revolutions for 4 cycle engines. In order to eliminate all reference to any particular bore and stroke it is convenient to express the torque variation as a pressure per sq. in. of piston area, tangential to the crank circle. To convert this to torque it is only necessary in any specific instance to multiply by the piston area and the crank radius.

For application to vibration problems it is necessary to analyse the variable tangential pressure (as given by a table or graph) into its harmonics of various orders, in accordance with the principles laid down by Fourier. A point of nomenclature arises here. In this book we take the order number of an harmonic to be the number of times the harmonic repeats itself in an engine cycle, whether two stroke or four stroke cycle. Some writers adopt the revolution and not the cycle as the basis and call, for example, a $4\frac{1}{2}$th harmonic (in a 4 cycle engine) that which we call a 9th harmonic.

For most purposes it is not necessary to know the epochs and the separate sine and cosine terms; all that is necessary is in general, the resultant amplitude $c=\sqrt{a^2+b^2}$ where "a" and "b" are the amplitudes of the sine and cosine components.

We need not describe the numerous graphical, mechanical and tabular methods of harmonic analysis; suffice it to say that if a tangential pressure curve is cut into N sections by N+1 equally spaced ordinates (including the two end ordinates) and the N sections are superimposed and added algebraically, the result is an approximation to a sine curve the amplitude (semi-range) of which is N times the Nth harmonic, provided the 2Nth, 3Nth......etc., harmonics (which are also present) can be neglected. This method has the merit of requiring no special apparatus or formulæ.

Harmonics of orders 1, 2 and 3 (2 cycle), 2, 4 and 6 (4 cycle) are very seriously affected by the inertia pressure due to the reciprocating parts. This explains why the resultant twisting moment curves of 2 and 3 cylinder 2 cycle engines and 2, 4 and 6 cylinder 4 cycle engines depend greatly on revolutions even if the indicator diagrams remain the same. Harmonics of 4 and upwards (2 cycle), and 3, 5, 7 and upwards (4 cycle) are practically unaffected by inertia forces; the resultant turning moment diagrams of engines having these numbers of cylinders (with equally spaced ignitions) are practically independent of revolutions if the indicator cards remain the same. In vibration problems harmonics of these low orders seldom come in question as resonance with them must be avoided at all cost. The values of the above harmonics due to *inertia only* are tabulated under expressed as fractions of the pressure obtained by multiplying the mass of the reciprocating parts (weight/g) by the acceleration in the crank-circle and dividing by the piston area. We may call this the mean inertia pressure.

Order of Harmonic (inertia only).		Amplitude of Harmonic as fraction of mean inertia pressure. Length of Con. Rod ÷ crank radius				
2 Cycle	4 Cycle	4·0	4·5	5·0	5·5	Epoch
1	2	0·062	0·055	0·050	0·045	0°
2	4	0·50	0·50	0·50	0·50	90°
3	6	0·186	0·165	0·150	0·135	60°
4	8	0·015	0·012	0·010	0·008	45°

The "epoch" is the crank angle reckoned from top dead centre at which the harmonic is zero passing from − to +.

As a slight digression it is worth mention that the 2nd, 3rd and 4th harmonics (2 cycle) and 4th, 6th and 8th (4 cycle) due to cylinder gas pressure are nearly in opposition as regards phase to the above harmonics due to inertia. Now the resultant turning moment curve of say a 6 cylinder engine with equally spaced ignitions consists of the 6th, 12th, 18th, etc., harmonics of which the 6th predominates. Accordingly the fluctuation in turning moment and excess energy are a minimum at about those revolutions which make the 6th harmonic due to gas pressure equal and opposite to that due to inertia. The 6th harmonic in a 4 cycle engine due to gas pressure has a value of about 25 lb./in.^{2}; if the connecting rod is, say, 4·5 cranks long the mean inertia pressure giving minimum torque fluctuation is about $25 \div 0{\cdot}165 = 150$ lb./in.^{2}

The 4th harmonic in a four stroke engine due to gas pressure is about 33 lb. and the inertia pressure for minimum fluctuation is about $33 \div 0{\cdot}5 = 66$ lb./in.^{2}

The value of the harmonics of gas pressure depends on the compression pressure, maximum pressure, mean pressure and the shape of the diagram in other respects ; if H is the value of any harmonic the following may be suggested, viz. :—

$$H=p_c f\left(\frac{p\ max}{p_c}, \frac{p\ mean}{p_c}, q\right)$$

where p_c=compression pressure and q defines in some way the degree of flatness or sharpness of the top of the diagram.

Wydler has worked out the values of H for p_c=p max= 482 lb./in.^{2} and square top four stroke diagrams for mean pressures from 0 to 128 lb./in.^{2}

Lewis has given graphs of the sine and cosine components and resultants for similar shapes of diagram. Values for two stroke S.A. engines have been given by Sulzer. Series of values for S.A. four stroke as well as S.A. and D.A. two stroke harmonic coefficients are given on page 182 for orders up to 12. For higher orders the values appear to have a tendency to decrease as the cube of the order and in the absence of more exact information this rule may be used to extend the lists, by plotting on log. paper.

The harmonics of double-acting engines are complicated by the reduction of area of the underside of the piston due to the presence of a piston rod and the reversed effect of connecting rod obliquity. Apart from these two effects there is a tendency for the odd harmonics 1, 3, 5, etc., of a 2 cycle engine to cancel out and the even harmonics 2, 4, 6, etc., to be doubled as compared with a single-acting two stroke engine, but the actual facts are not so simple and the only sound procedure is to draw a combined tangential pressure diagram and analyse it. The case of a four stroke D.A. engine is further complicated by the two alternative firing order, viz. top-bottom or bottom-top.

Damping.—If there were no damping, the amplitude of oscillations at a critical speed would theoretically increase indefinitely. In practice the amplitude is limited, even in severe instances by at least two circumstances :—

(1) The restraints of the bearings which introduce reactions tending to limit the scope of the vibrations.

(2) Marked deviations from Hookes' Law when the skin stress in the shaft exceeds the fatigue limit.

Vibrations which bring into action the second of those limitations are too severe to be tolerated as they are accompanied by noise, loss of power, generation of heat and eventual fracture.

It remains to explain the origin of the damping forces which limit the vibrations (due to relatively weak harmonics) which can be tolerated.

The earlier investigators assumed the existence of frictional forces proportional to the piston area and the vibration velocity referred to each crank circle. This assumption was no doubt prompted by the form of the classical equations of damped vibration of systems of single or multiple freedom. Unfortunately the damping constant was found to be different for different engines and for the same engine in different circumstances. Furthermore if the system consists of an engine with a heavy fly-wheel coupled by a relatively flexible shaft to a dynamo, then the cylinders contribute very little damping and if the shaft stresses are calculated with a damping coefficient determined in different circumstances, the stresses so calculated may be enormous, whereas the actual stress may be quite moderate. A further objection is that the damping factor which gives correct results for a single noded vibration, gives calculated stress for a two noded vibration which are much higher than those observed.

It has been suggested that the most important cause of limitation is the departure from uniform velocity of rotation, i.e. the degree of irregularity, particularly that due to the cyclical fluctuation of equivalent moment of inertia. A strict mathematical treatment of this problem is difficult. Approximate considerations appear to indicate that inequality of rotation is effective in limiting vibrations of which the frequencies are high in comparison with the frequency of fluctuation of speed, but is not of primary importance in most practically interesting cases.

Other investigators following Lewis have assumed that elastic hysteresis of the shaft material is the most important source of damping. This assumption can be shewn to lead to the simple result that vibration stress is proportional to equilibrium stress whatever the shape of the vibration form provided the hysteresis work varies as the square of the stress. This is roughly correct for low stresses only. A natural generalisation consists in assuming that vibration stress is a function of equilibrium stress (see Fig. 80) and this in the author's experience is the most successful working hypothesis so far advanced. The chief objection to it is that experiments by Dorey on specimens of crank-shaft steel do not exhibit enough elastic hysteresis to account for the observed limitations of critical vibrations. The discrepancy may be cleared up later ; meanwhile the hysteresis theory whether right or wrong seems to give the *form* of solution which accords best with experience without involving special coefficients for different classes of vibration form.

Damping due to propellers, fluid fly-wheels, etc., requires special treatment and when strongly in evidence other sources of damping may be neglected in comparison.

Damping due to Marine Propeller.—The limitation of the torsional vibrations of a marine shafting system due to propeller damping may be calculated approximately from the following considerations.

Under constant conditions of draft, weather, etc., the mean propeller torque varies as the $(\text{R.P.M.})^2$, and the speed of the vessel as the R.P.M. directly. With a constant speed of advance of the vessel against variable resistance the propeller torque varies approximately as the $(\text{R.P.M.})^3$. These statements may be taken as roughly correct for ordinary mercantile vessels.

Let Tmc = mean propeller torque at the critical speed.

$\pm$Te = "Equilibrium" fluctuation of torque at the propeller, i.e. the total fluctuation of engine torque multiplied by the moment of inertia of the propeller and divided by the sum of the moments of inertia of engine and propeller.

n = R.P.M. @ critical speed.

F = Vibration frequency per minute.

$\pm\theta$ = Angular swing in radians of the propeller at resonance.

Ip = Polar moment of inertia of the section of the shafting.

l = length of shafting from propeller to the node.

C = Modulus of rigidity of shaft material $= 12 \times 10^6$ lb./in.^{2}

At resonance θ increases until the variation of propeller torque due to the fluctuation of angular speed of the propeller just balances the "equilibrium" torque fluctuation Te.

θ may be found as follows :—

The fluctuation of angular speed of propeller is $\pm 2\pi F\theta$; the mean angular speed is $2\pi n$ $\therefore$ the percentage speed fluctuation is $\pm 100\ \theta F/n$. Since the speed of the ship remains sensibly constant the percentage fluctuation of propeller torque $= \pm 300\ \theta F/n$.

Applying this to the data of page 198, the mean torque at full speed is 420,000 lb. in. and since the critical speed occurs at 74 R.P.M., the mean torque at the critical speed is :—

$$420{,}000\left(\frac{74}{120}\right)^2=160{,}000 \text{ lb. in.}$$

Te (see p. 199) is $\pm 52{,}000$ lb., so the fluctuation of propeller torque is 32·5%. Substituting above with $F/n=3$.

$$\theta=\frac{32{\cdot}5}{300\times 3}=\frac{1}{27{\cdot}7}$$

The corresponding fluctuation of torque in the shafting is :—

$$\pm\frac{IC\theta}{l}=\frac{\frac{\pi}{32}\cdot 9^4\times 12\times 10^6}{27{\cdot}7\times 820}=\pm 340{,}000 \text{ lb. in.}$$

This should be compounded with T_e in quadrature giving, $\sqrt{340{,}000^2+52{,}000^2}=\pm$*345,000 lb. in.*

Torsional Vibration Dampers.—With relatively slow speed engines it is generally possible to avoid dangerous critical speeds by making the crank-shaft diameter sufficiently large. Large journal diameter is more useful than large crank-pin diameter since the latter involves increased weight of revolving parts thus partially defeating the object in view, i.e. to increase the natural frequency. As speeds are increased it becomes impossible to avoid serious criticals within the working range and dampers are used to keep the vibration within safe limits. Several types are in use.

The original Lanchester type consists of a light fly-wheel attached by means of a multi-plate clutch to the end of the crank-shaft remote from the fly-wheel. When vibration begins, vibratory relative moment takes place at the clutch and frictional forces are brought into existence which act in opposition to the forces exciting vibration. The plate clutch works in oil and the friction between the plates is supposed to be of the viscous fluid type. Variations of this scheme include dry plate or cone clutches either metal to metal or with brake lining material.

In another class of damper the light fly-wheel is connected to the crank-shaft hydraulically by means of pump elements filled with lubricating oil. At resonance relative motion between the fly-wheel and crank-shaft sets up a pumping action through narrow openings resulting in forces in opposition to the forces causing vibration.

All these forms of damper result in the dissipation of energy to a greater or less extent. Other forms of damper have been suggested in which the light fly-wheel is connected to the crank-shaft by flexible members of periodically varying stiffness, or in which the virtual mass of the fly-wheel is varied periodically.

Dampers of the frictional type have been made in which the light fly-wheel is connected to the crank-shaft by a spring connection of such stiffness that the natural frequency of vibration of the damper wheel relative to the crank-shaft is the same as that of the crank-shaft system without damper. At resonance the damper vibrates somewhat violently and the inertia forces so brought into play practically annul the forces which would otherwise cause the crank-shaft to vibrate. Some friction damping appears to be desirable in this case also. A similar scheme has been proposed by Inglis for preventing transverse vibration of ship members.

The theory of torsional vibration dampers has been discussed by Carter.

Literature.—Bauer and Robertson, "Marine Engines and Boilers," Chapter III.—Griffin.

Carter, B. C., "Torsional Vibration in Engines."—Aero Research Committee, R. and M. 1053. Also, "An Empirical Formula for Crank-shaft stiffness in Torsion."—*Engineering*, July 13th, 1928.

Geiger, J., "Mechanische Schwingungen."—Springer.

Hopkinson, B., "Vibrations of Systems having one degree of Freedom."—Camb. Univ. Press.

Lewis, F. M., "Torsional Vibrations in the Diesel Engine." —Soc. N.A. and M.E., New York, November 12/13, 1925.

Kjaer, V. A., "Torsional Vibration in Crank-shafts."—*The Motor Ship*, August, 1930.

Jackson, P., "The Vibrations of Oil Engines."—Diesel Users Assoc., April 26th, 1933.

Thorne, A. T., and Calderwood, J., "Torsional Oscillations in Propeller Shafts."—N.E. Coast Inst. Engs. and Shipbuilders, November 17th, 1922.

Sulzer, R., "Causes and Prevention of Vibration in Motor Ships."—I.N.A., April 11th, 1930.

Dorey, S. F., "Limits of Torsional Vibration Stress in Marine Oil Engine Shafting."—I.N.A., March 27th, 1947.

Wilson, W. Ker, "Practical Solution of Torsional Vibration Problems."—Chapman and Hall.

Tuplin, W. A., "Torsional Vibration."—Chapman and Hall.

CHAPTER XI

FRAMEWORK

A LARGE number of different types of framework have been employed in Diesel Engine construction, and a complete classification will not be attempted here. The outstanding types in successful practice may, however, be broadly divided into a few well-defined classes, as under :—

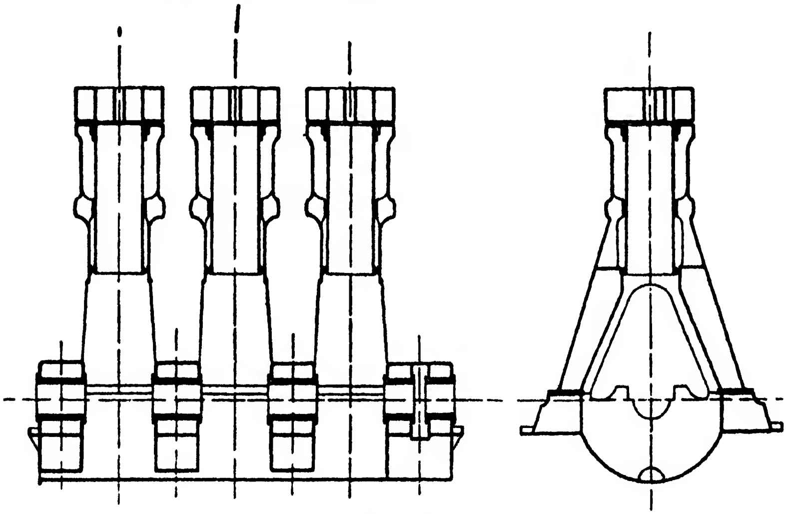


FIG. 89.

"A" Frame Type.—This is the earliest type of Diesel Engine construction, and for slow speed engines is still very occasionally used. Referring to the diagrammatic drawing Fig. 89, it will be seen that a stiff bedplate of box section is provided, and that each cylinder stands on its own legs without support from its neighbours. The legs of the column are cast integrally with the cylinder jacket, into which a liner is fitted. The breech end of the cylinder is closed by means of a deep cylinder cover of box section.

The main tensile load due to the cylinder pressure is transmitted from the cover through the jacket and legs to the bedplate. The reaction corresponding to this load occurs, of course, at the main bearings, and consequently that part of the bedplate between the column feet and the main bearing housings must be designed to deal with the bending moment occasioned by the fact that the tensile load in the columns and the reaction at the bearings are not in the same plane. Casting the cylinder jacket and column in one piece reduces fitting and machining operations to a minimum and the independence of the individual cylinders would appear to have no disadvantages so far as slow speed engines are concerned.

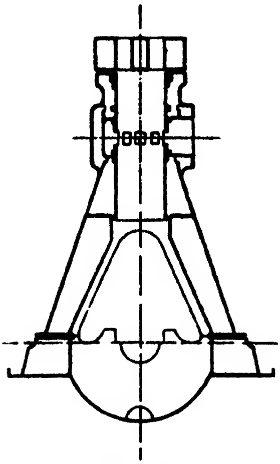


Fig. 90.

Fig. 90 shews the same type of construction applied to a two stroke land engine and Fig. 93 to a four stroke marine cylinder.

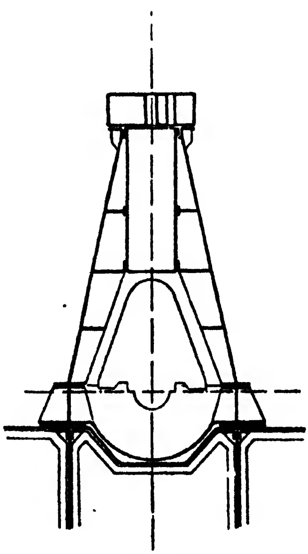


Fig. 93.

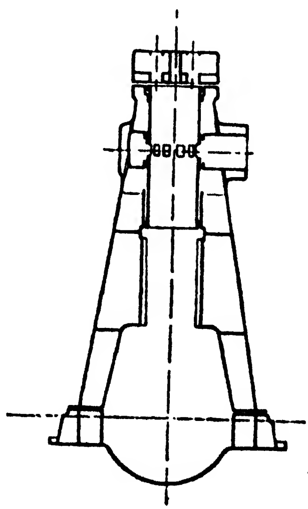


Fig. 94.

Slightly lengthening the column legs enables crosshead and guides to be fitted (see Fig. 94, which represents a large two cycle land engine). Occasionally one of the column legs takes the form of a steel tie rod, with a view to giving greater accessibility to the running gear and to enable the crank-shaft to be replaced, if necessary, without dismantling the whole engine. Unfortunately this arrangement nullifies many advantages of the "A" frame construction, as special splash guards must now be fitted to retain the lubricating oil, which office they do not always perform very efficiently, and also additional

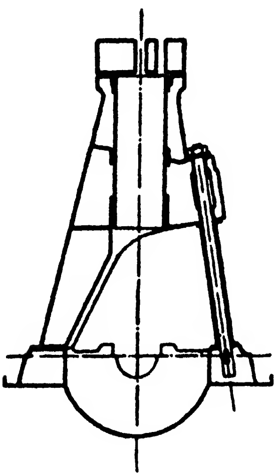


FIG. 91.

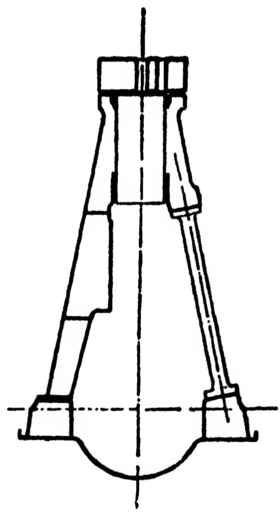


FIG. 92.

machining and fitting operations are introduced which add to the cost of production without increasing the efficiency of working. Figs. 91 and 92 shew this construction applied to trunk and crosshead engines respectively.

Crank-case Type.—The crank-case type of Diesel Engine was introduced when a desire was felt for higher speeds, necessitating forced lubrication. The crank-case bears external resemblance to that of a high speed steam engine (see Fig. 95). On the other hand, the high pressures dealt with in the cylinder of a Diesel Engine necessitate the crank-case being strengthened internally to an extent which is not found necessary in steam practice. Sometimes the box or girder construction of the crank-case is relied upon to transmit the tensile stresses from the cover to the bedplate; more frequently, however, steel staybolts are provided for this purpose (see Fig. 96). The latter procedure, however, does not justify flimsy construction

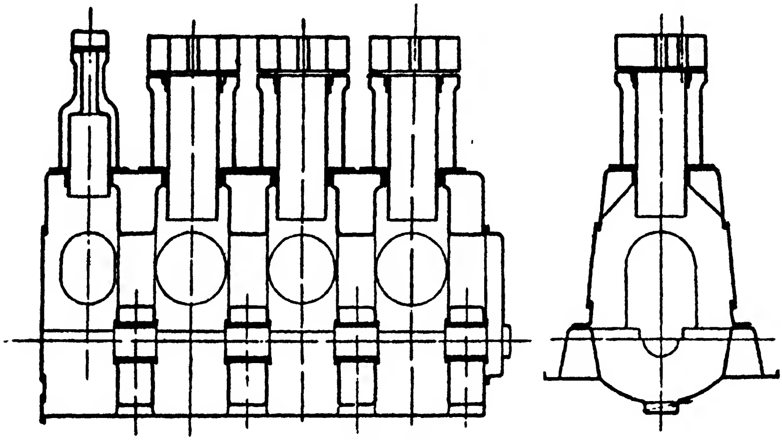


Fig. 95.

or a careless distribution of metal in the crank-case, as the guide pressure has still to be reckoned with and the pressure caused by tightening up the staybolts may be relied upon to cause serious distortion of a poorly ribbed case.

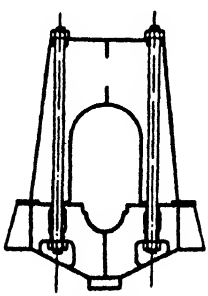


Fig. 96.

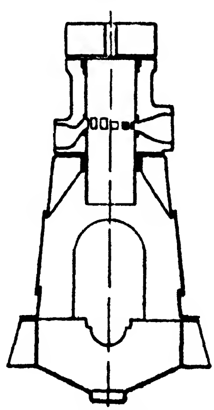


Fig. 97.

The cylinders being separate are secured either by a round, studded flange or by passing the staybolts through each corner of a deep, square flange of hollow section cast at the lower end of the cylinder jacket for this purpose. The latter arrangement requires four staybolts for each cylinder, whereas with the former it is usual to arrange a pair of staybolts only at each main bearing girder. With the crank-case construction it is not necessary to make the side girders of the bedplate so strong as for an "A" frame type of engine, as the bending action referred to above is avoided and the bedplate and crank-case when bolted together form a girder construction of great rigidity. On the other hand, the upper part of the crank-case is clearly subject to bending actions similar to those which

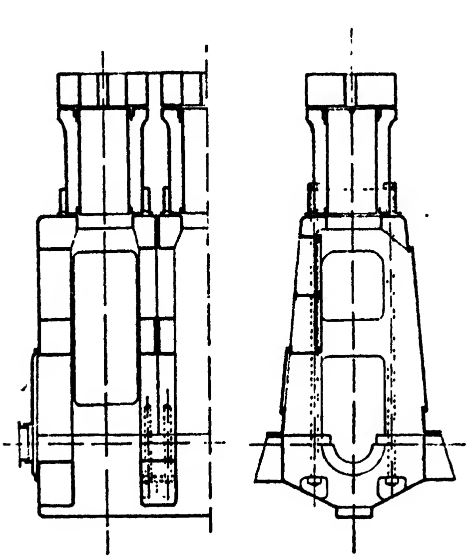


Fig. 98.

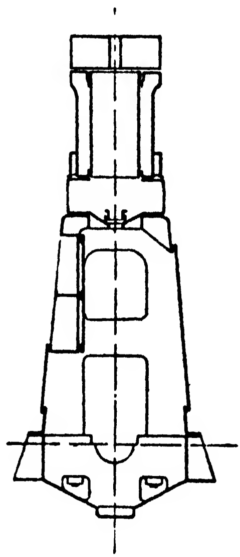


Fig. 99.

occur in the bedplate of an "A" frame engine and must be designed with this fact in view. Fig. 97 shews a section through a two stroke trunk engine of the crank-case type. Figs. 98 and 99 shew the crank-case construction applied to crosshead engines. The suitability of this type of framework for marine service has been amply proved in practice. In some cases the crank-case is common to two or more cylinders, and in others the case for each cylinder is a separate casting, the individual cases being bolted together to form a virtually continuous box of great strength and rigidity.

Trestle Type.—With this construction, the frames over the main bearings and midway between the cylinder centres are each cast separately, machined on two parallel athwartship faces, and connected together at the top by distance pieces. This plan has several advantages for large engines. The separate trestle frames being only about 12" in depth, even for a marine engine developing about 1000 B.H.P. per cylinder, lend themselves to easy and expeditious moulding operations in the foundry, take up less space in the machine shop than the crank-case type, and are conveniently handled during erection.

The frames must be sufficiently stiff in a transverse direction to avoid resonance with the forces of guide reaction (see Chapter XII); this consideration may determine both the outline and the metal thickness, quite apart from considerations of tensile stress due to gas pressure or compression due to tie bolt tension.

Fig. 8 gives a side view of such a trestle frame for a double acting engine. Above the main frames are arranged upper frames for supporting the cylinders. The frames may be of cast iron or fabricated steel. The spaces between the main frames are closed by oil-tight removable doors of sheet steel with reinforced edges and leather jointing strips. Fig. 8 also shows the guide for the crosshead. The guide in this engine is of the single type with astern plates secured by fitted bolts. The guide plate also serves as a distance plate between frames.

Fig. 152 shows an engine fitted with double guides located in the frames for the accommodation of double crosshead shoes similar to those fitted to paddle steam engines. With either type abundant pressure lubrication renders water cooling of guides unnecessary.

The crosshead slipper surfaces are usually whitemetalled and the guide surfaces made of cast iron.

Framework of the trestle type is frequently but not necessarily associated with the use of vertical tie bolts. If opposed pistons of equal diameter are used, the vertical components of gas pressure loads are balanced, and tie bolts would appear to be superfluous.

Design of Bedplates.—The design of a suitable bedplate involves consideration of the following points, which will be dealt with in order, viz. :—

(1) The provision of a suitable main bearing.

(2) A girder construction under each main bearing, capable of supporting the full bearing load without central support.

(3) A sufficiently strong and stiff connection between the main bearing girders, forming at the same time an oil-tight tray (unless a sheet steel tray is fitted).

(4) Suitable studding or staybolt arrangements for carrying the tensile pull of the columns.

(5) Adequate bearing surface on, and correction to, the foundation.

(6) Means for collecting drainage of lubricating oil to some convenient sump, whence it can readily be drawn off with a view to filtration and repeated use.

(7) Facings for barring gear, auxiliary pumps, etc.

Main Bearings.—The earlier types of Diesel Engines were fitted with ring lubricated main bearings having a projected area of 50 to 75% of the piston area and the peripheral speed was limited to about 500 or 600 ft./min. This method is suitable for simple types of slow speed engine, for example, certain types of horizontal engines and vertical crank-case compression engines. For most classes of Diesel Engines forced lubrication is preferred and the cylinders are placed as close together as the design will allow, with the result that main bearings having a ratio of length to diameter as low as about 0·5 are not uncommon. In such cases the end bearings and the bearing between two symmetrical groups of say 3 or 4 cranks are made about 50 to 100% wider. The projected area of the narrower bearings may be as little as 25% of the piston area in the case of small high speed engines; 35% to 45% is more usual in larger engines. Peripheral speeds range up to 1500 ft./min. or over.

Ring Lubricated Main Bearings.—These are similar to the bearings fitted to electrical machinery and need not be described in detail. The arrangements for catching the oil squeezed out of the bearings and conveying it back to the oil well merit careful attention, as inefficiency in this direction leads to unnecessary waste of oil. In particular, the oil spaces and holes should be as large as possible, to avoid congestion. Fig. 100 shews a very usual form.

Forced Lubricated Bearings.—Figs. 101 and 102 show typical designs. The shells are usually of low carbon steel with manganese not exceeding about 0·9% and free of nickel and chromium which are detrimental to adhesion of white metal.

The bearing cap should be designed as a beam capable of

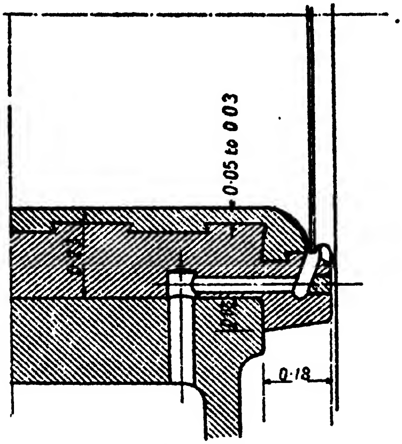


FIG. 100.

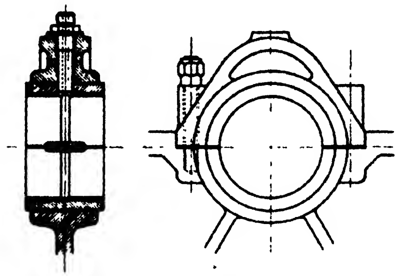


FIG. 101.

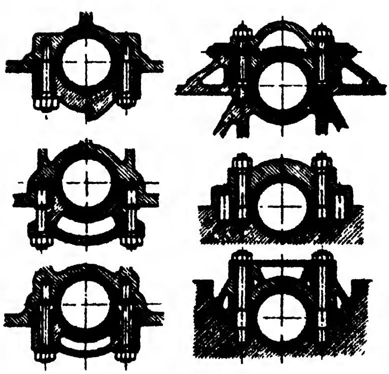


FIG. 102.

carrying a central load equivalent to the full inertia and centrifugal load due to one set of running gear. This is possibly a little on the safe side, but reference to Chapter VIII will shew that the margin is not large in the case investigated there.

Main Bearing Girder.—Where forced lubrication is used, the main bearing girder may conveniently be of I section, the bottom flange being formed by the oil tray; for ring lubrication a box section lends itself more conveniently to the formation of the oil reservoir. The depth of the girder is determined by that of the oil tray required to give an inch clearance or so to the connecting rod big end at the bottom of its path. Referring to Chapter VIII, it will be seen that the maximum reaction at a bearing for the case considered is equal to 0·8 of the resultant load due to pressure, inertia and centrifugal force, and this is the load for which the girder must be designed. In other cases the load may be less than this, but it is doubtful if in any case it approximates to the conventional load frequently assumed, viz. one-half the resultant cylinder load.

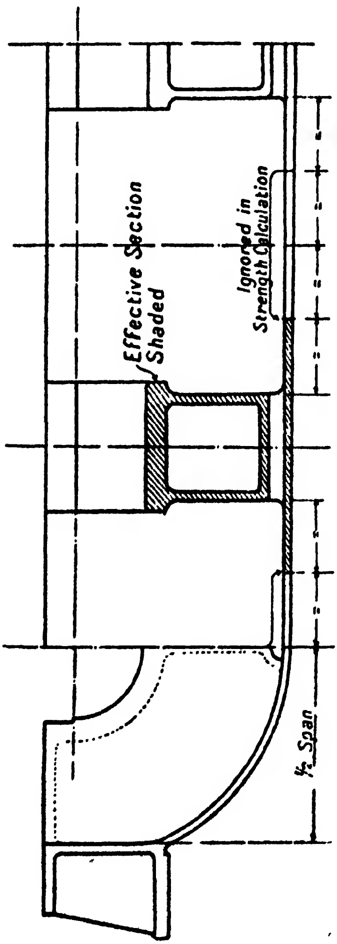


Fig. 103.

A very debatable point is the extent to which the oil tray can legitimately be regarded as a part of the tensile flange of the girder. The author's practice in this respect is to ignore the middle half of that part of the tray lying between two bearing girders (see Fig. 103). The span of the girder is the distance between the two points at which it meets the side girders.

If W = Load on girder in lb.
l = Span in in.
M = Bending moment in in. lb.
Then $M = 0{\cdot}2\ Wl$—approximately.

The assumption being that the fixing moments at the ends are negligible (which if not correct is on the safe side) and that the load is distributed over the journal. Allowable stress 1500–2500 lb. per sq. in. for cast iron.

Side Girders.—With "A" frame engines the bending moment on each side girder may be taken as :—

$$\frac{\text{Half pressure load} \times \text{Distance between centres of bearings}}{6}$$

The usual stress allowance being about 1500 lb. per sq. in. Where the trestle or crank-case type of frame is used the side girders may be of lighter section.

Arrangements for Carrying Tensile Pull of Columns.—With the "A" frame construction the foot of each column is secured by a row of studs, the stress in which when referred to the normal maximum working pressure of 500 lb. per sq. in. in the cylinder amounts to about 5000 to 10,000 lb. per sq. in., according to the size of the stud. It is very convenient to have a list of the loads which studs and bolts of different sizes can conveniently carry, and such a list is given below :—

Size of Bolt or Stud (Whitworth).	Stress (Core) allowed, lb./in.^{2}	Working Load, lb.
½″	2000	240
⅝″	2850	550
¾″	3550	1080
⅞″	4250	1800
1″	5000	2750
1⅛″	5250	3650
1¼″	5500	5000
1⅜″	6000	6300
1½″	7100	9300
1¾″	8500	15,000
2″	,,	20,000
2¼″	,,	24,000
2½″	,,	32,000
2¾″	,,	37,000
3″	,,	46,000

The table (p. 225) refers to mild steel bolts for large or medium sized engines in which the smaller bolts and studs are liable to rough handling. The design of small engines would be seriously hampered by such lowly stressed bolts and higher stresses are therefore adopted. For example, $\frac{5}{8}''$ cylinder studs for a small engine might well be used with nominal stresses of about 6000 lbs./in.^{2} for well specified and tested mild steel, 8000 lb./in.^{2} for 3% nickel steel and 12,000 lbs./in.^{2} for heat treated nickel chrome steel.

Care must be taken that none of the studs are at any considerable distance from adequate supporting ribs. This is best obtained by judicious spacing of the studs rather than the provision of special ribs for the purpose.

With the crank-case and trestle types staybolts are usually fitted, and in land work at any rate these should terminate within the bedplate and not penetrate to the underside of the latter for fear of oil leakage, which would destroy the concrete. The studs or bolts used to secure the crank-case to the bedplate may be disposed more with a view to making an oil-tight joint than to carry any definite load. If staybolts are not fitted, then a sufficiency of effective bolt or stud area must be arranged in the neighbourhood of each column foot, and some of the bolts or studs must be inside the crank-case.

Cam-shaft Driving Gear.—The motion required by the valve gear is derived from the crank-shaft by spiral or spur gearing or by a roller chain drive. Fig. 104 shews a very common arrangement of spiral drive, with the driving-wheel between the two sections of a divided main bearing. It is good practice to make the combined length of the two sections about 50% greater than the length of a normal bearing. There would appear to be nothing against having the spiral wheel outside the bearing altogether, provided the gear is at the fly-wheel end. This position for the valve gear drive is preferable to the opposite end as the weight of the fly-wheel tends to keep the journal in contact with its lower bearing shell, whereas the forward journal has freedom of motion to the extent of the running clearance, and is subject to greater torsional irregularity.

In six cylinder engines the spiral gear is frequently arranged at the centre of the engine, where it is very easily accommodated. There seems to be some feeling that the cam-shaft would whip unduly if driven from the end. This difficulty (if any difficulty can be said to exist) is easily overcome by making the cam-shaft about 10% larger in diameter than would be considered sufficient for a four cylinder engine.

Where spur gearing is used for the valve gear drive, facings must be provided for the support of the first motion shaft.

Oil Drainage.—With land engines of the non-forced lubricated type the oil which drips down from the cylinders and is thrown from the big ends is drained periodically from the

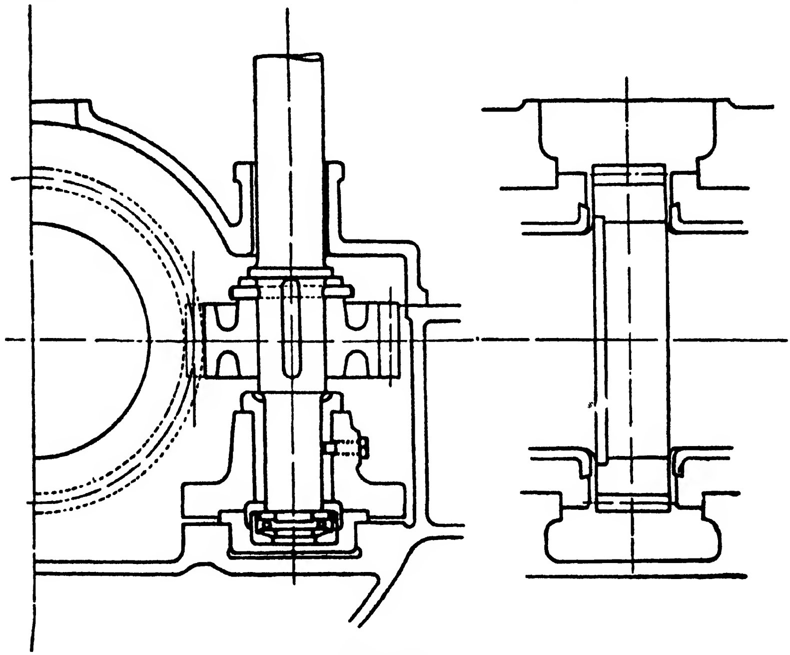


Fig. 104.

forward end of the bedplate and holes are cored through the main bearings girders to give the oil free passage. Perhaps the best arrangement is a rectangular duct about four inches square running down the centre of the oil tray. Small holes are useless as they are easily choked. With forced lubricated engines the same arrangements are made with the addition of a collecting sump of good capacity, a pump for forcing the oil into the bearings and filters in duplicate. These features being familiar in steam engine practice need not be described in detail. It must be borne in mind, however, that where trunk engines are being considered the oil is contaminated with carbon, so that the filtering arrangements require to be on a more liberal scale than is necessary with engines in which the cylinder is isolated from the crank-case.

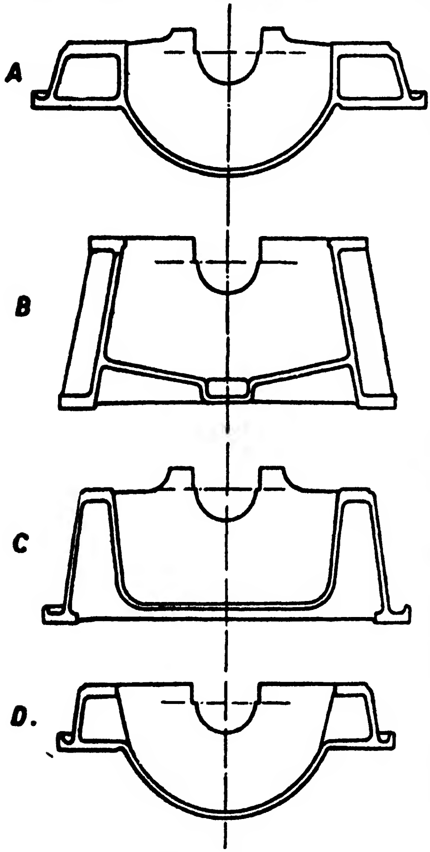


Fig. 105.

Proportions of Bedplate Sections.—Fig. 105 gives approximate proportions for various types of bed-plate sections, Type "A" is usually associated with the "A" frame construction. Type "B" is a useful one for main or auxiliary marine engines as it enables the engine to be bolted direct to a tank top or to a deck without building up a special seating.

Type "C" is preferable to type "A" for land generating sets as the extra depth of bedplate enables the generator to be flush with the engine room floor without the necessity of building the engine on an unsightly plinth. A deep bedplate is also very desirable with six cylinder engines as the cancellation at the centre of the engine of the inertia and centrifugal couples gives rise to vibrations, the amplitudes of which are reduced by increasing the stiffness of the framework.

The general thickness of metal may be about 6% of the cylinder bore increased to about 8% or 10% on machined surfaces. These figures are usually exceeded on small engines on account of foundry considerations.

Bore of Cylinder, in.	10	12	15	18	21	24	27	30
General Thickness of Metal for Bedplate.	$\frac{5}{8}$	$\frac{3}{4}$	$\frac{13}{16}$	$\frac{15}{16}$	$1\frac{1}{16}$	$1\frac{3}{16}$	$1\frac{1}{4}$	$1\frac{3}{8}$

Design of Crank-cases.—Sometimes the crank-case is cast in one piěce with the bedplate as in Fig. 106, in which case some provision must be made for inserting the crankshaft :

(*a*) through vertical gaps in the transverse frames as in Fig. 106,

(*b*) through side gaps in the frames,

(*c*) or endways ; in which case circular housings of large diameter may be bored for the accommodation of bearings.

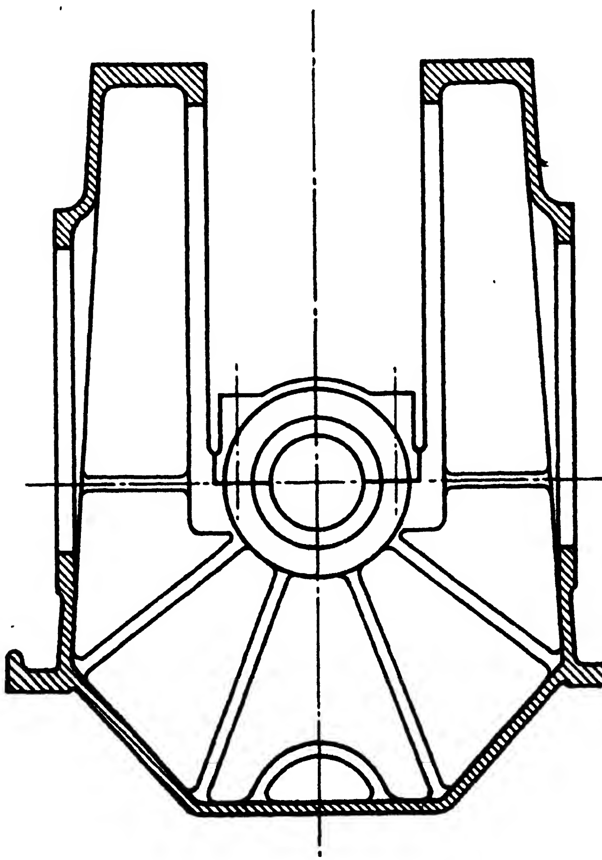


Fig. 106.

More usually the crank-case is a separate casting as in Fig. 107, which shows a typical crank-case with accommodation for circular camshaft bearings and supporting individually cast cylinders. In the example illustrated the crank-case legs or columns carry the tensile load due to the firing pressures without the aid of tie bolts and must be given a sectional area suitable for this duty ; also the studs which secure the cylinders to the crank-case must be suitably connected by bosses and possibly ribs to enable the tensile loads to be transferred from the studs to the column without setting up excessive local bending stresses. The feet of the columns are secured to the bedplate by a number of studs and the same precaution applies here also.

If vertical tie bolts are used between the bedplate and the top of the crank-case, as in Fig. 96, or from the bedplate to the cylinders, the studded connections need not be so strong and the crank-case casting may be somewhat lighter, but the saving in weight is small since a certain stiffness is required to keep transverse vibration within a tolerable limit.

The legs or columns of a cast crank-case are usually made of H section to facilitate coring. Good fillet radii and transition tapers between walls of different thickness are in general to be preferred to ribs and bosses. The table below is a rough guide to average practice :—

Bore of Cylinder.	General Thickness of Crank-case Metal (no Tie Bolts)	Same (with Tie Bolts)	Diameter of Tie Bolts.
10″	$\frac{1}{2}$″	$\frac{7}{16}$″	$1\frac{5}{8}$″
12″	$\frac{5}{8}$″	$\frac{1}{2}$″	2″
15″	$\frac{3}{4}$″	$\frac{9}{16}$″	$2\frac{1}{2}$″
18″	$\frac{7}{8}$″	$\frac{3}{4}$″	3″
21″	$1\frac{1}{16}$″	$\frac{7}{8}$″	$3\frac{1}{2}$″
24″	$1\frac{3}{16}$″	1″	4″
27″	$1\frac{3}{8}$″	$1\frac{1}{16}$″	$4\frac{3}{8}$″
30″	$1\frac{1}{2}$″	$1\frac{3}{16}$″	$4\frac{3}{4}$″

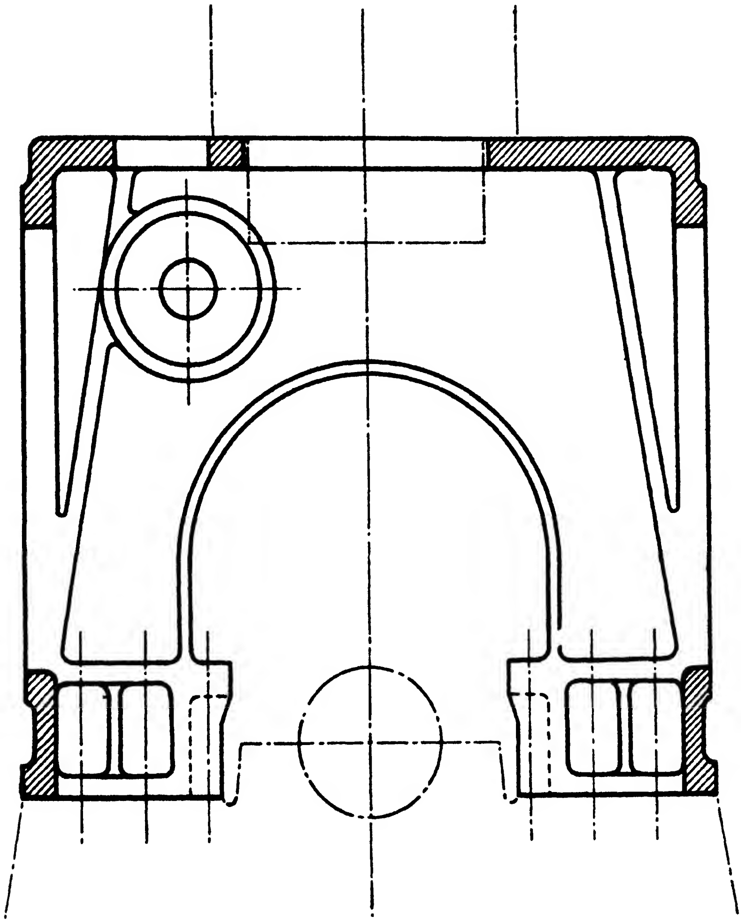


Fig. 107.

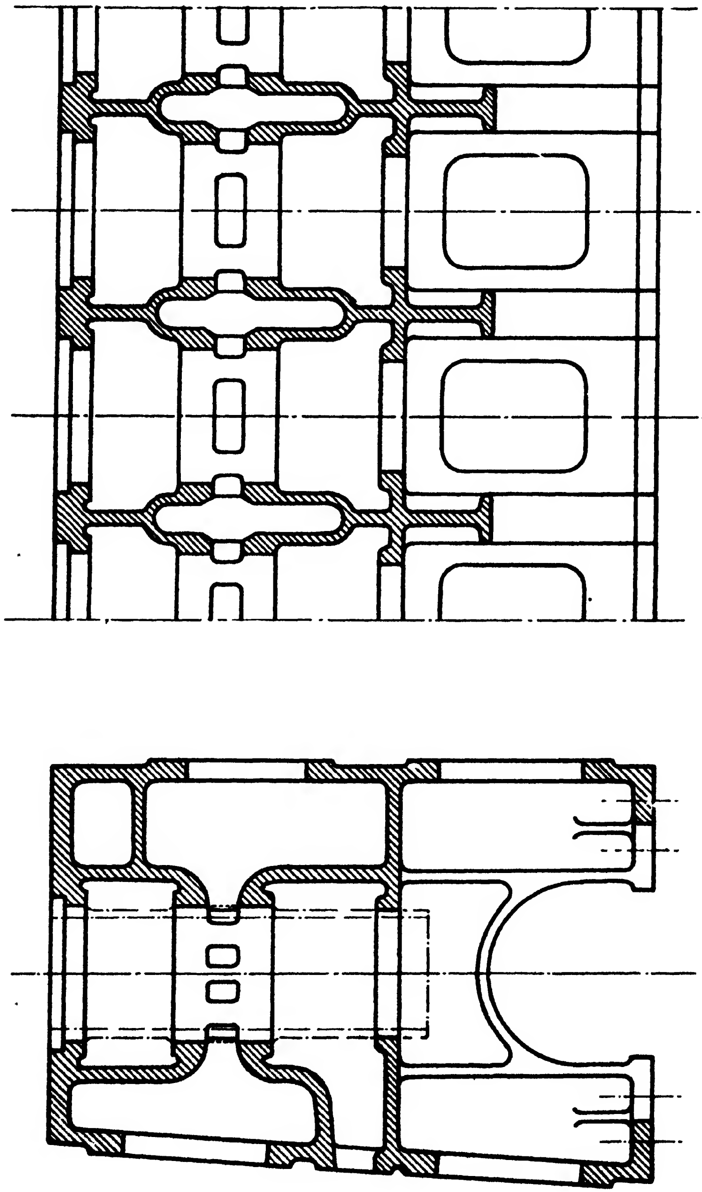


FIG. 108.

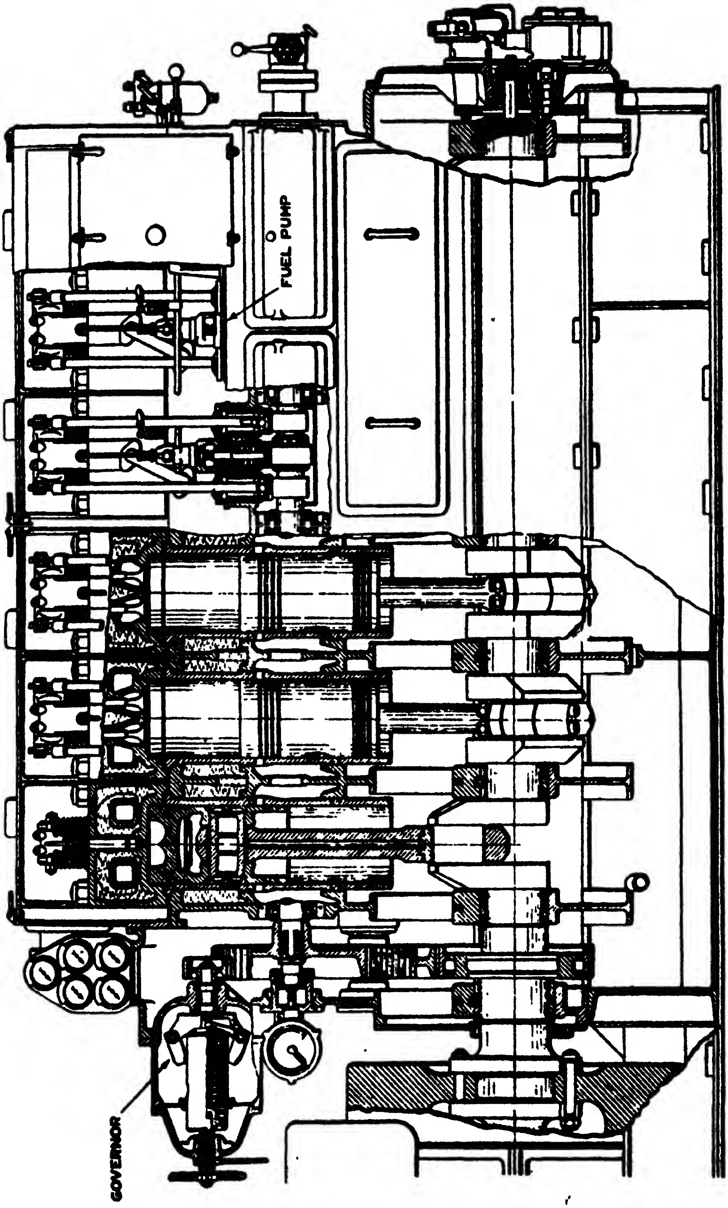


Fig. 108a.

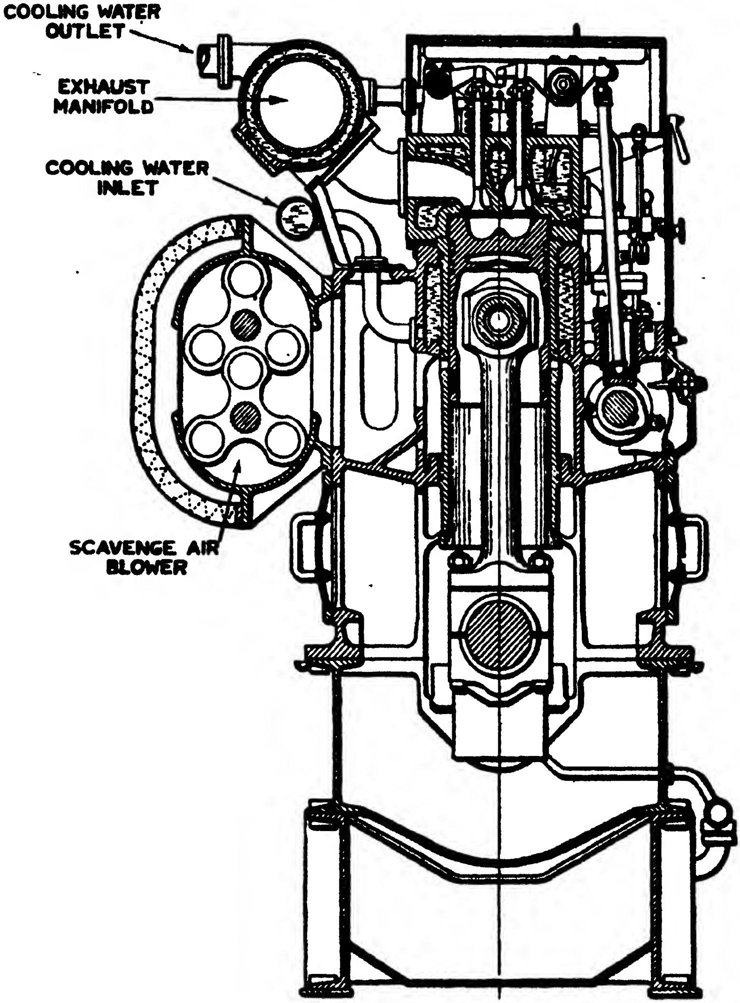


Fig. 108B.

This diagram and figure 108A illustrate sections of Harland and Wolff 2 cycle engine with poppet exhaust valves, showing fabricated steel bedplate and cast iron crankcase.

For cylinders up to, say, 12″ diameter, the crank-case may be cast in one piece for as many as eight cylinders; alternatively cases for eight or even six cylinders may be cast in two pieces and bolted together at the middle of the engine. It is largely a matter of convenience in the foundry and machine shop. The end of the case next the flywheel may include a compartment for the chain or gear drive for the camshaft; or this compartment may take the form of a special casting or welding of different outline and thinner metal (see Figs. 318, 319). Other points to be considered in designing a crank-case are :—

(1) The provision of oil-tight access doors of ample size for overhauling the bottom ends.
(2) End casings provided with oil flingers, stuffing boxes, or other means of preventing the escape of oil.
(3) Facings, and other necessary accommodation for valve gear, etc.
(4) Bosses to carry lubrication oil connections to the main bearings.
(5) Facings for platform brackets.
(6) A vent pipe or valve of large area, to relieve pressure in the event of an explosion in the crank-case without loss of lubricating oil during normal working.
(7) Steady pins to each section of the case, to fix correct location.

Machining the Framework generally.—In designing all parts of an engine the designer will keep in mind the capabilities and limitations of the manufacturing plant and the operatives. This is especially necessary in the case of the framework, on account of the relatively large size of the parts. Where the most modern type of face milling plant is available the element of size offers no difficulties, and bedplates of 60 feet in length may be faced in one operation. Where planing must be resorted to the capacity of the machines must be studied in the early stages of the design. Machined faces should be arranged in as few different planes as possible, and ribs or flanges projecting beyond those planes are to be avoided as much for convenience in machining as for the sake of appearances. The simpler forms of girder or box-girder construction are to be preferred to those designs in which alternate perforation by lightening holes and reinforcement by ribbing mutually defeat each other's object. The lightest, strongest and cheapest forms are to be attained with a minimum of holes and ribs when cast iron is used. Large steel castings, however, are preferably lightened out almost to the extent of lattice-work, in order to facilitate rapid stripping of the cores after solidification and to minimise initial stresses.

Monoblock Crank-cases.—A very rigid framework is obtained by casting the crank-case and cylinder jackets of a group of cylinders in one piece. This construction facilitates complete enclosure of the valve gear which is in many cases desirable in the interests of cleanliness and suppression of noise; it also leads to a neat external appearance and economical manufacture. Petrol engine designers have led the way in this class of design which is now applied to Diesel Engines developing 200 B.H.P. or more per cylinder.

Typical examples are shewn in Figs. 7 and 10 (*ante*). The first refers to a four stroke stationary engine of substantial build, the latter to a light high speed engine. With small engines anything up to 8 cylinders may be accommodated in a single block. Larger engines may be divided into groups of 3 or 4 cylinders, to facilitate casting, the two or more blocks being bolted together. The design of such monoblocks requires careful scheming to reconcile the claims of,

(a) Strength and rigidity.
(b) Accessibility of working parts.
(c) Economical foundry work and machining.
(d) Tightness of enclosure plates and covers.

When cost is more important than weight the material used is grey cast iron. In lighter designs alloy cast irons can be used to advantage. Further weight reduction is possible by the use of cast steel or bronze; yet further by alloys of aluminium and magnesium.

Welded Steel Framework.—The rapid advance in the technique of electric welding has opened up new possibilities for the lightening of Diesel Engine framework without the use of costly materials. The designs which result from the use of this method do not differ greatly in external appearance from good examples of monoblock castings; plane surfaces predominate in both, but in designing a welding careful consideration must be given to the order of assembling and welding the various components and the incidence of stress on the lines of welding. In some designs it is claimed that the main stresses are carried in shear of interpenetrating members independently of the welding. In others the main tensions are carried by tie bars of sufficient section to relieve the welded structure of all but subsidiary stresses.

The restrictions which the use of welded plates as compared with the use of castings, places on the designer's fancy are in the main beneficial, and effectually rule out extravagances which all too often spoil the design of castings.

Framework of Large Engines.—Examples of framework suitable for large engines have been shown diagrammatically in Figs. 98 and 99; see also Figs. 8 and 152, showing sections of large marine engines, for further detail.

In very large engines it is possible with some advantage to effect a subdivision which would be uneconomical in smaller sizes; for example, the bedplate cross girders may be cast separately from the side girders and secured thereto by fitted bolts. This diminishes foundry risks and may enable the parts to be machined with smaller and quicker tools than would otherwise be possible. Similarly the columns or trestles may be divided horizontally or vertically as may be convenient from the points of view of machining or erection.

The use of steel castings in place of iron castings holds out the inducement of reduced weight and increased safety. The higher cost per ton of steel castings as compared with iron is partly offset by the reduced weight.

A further alternative is the use of weldings instead of castings.

Literature.—Jackson, P., "Design of Diesel Engine Castings in Relation to Foundry Practice."—*Gas and Oil Power*.

"Applications of Aluminium in Diesel Engines."—*Oil Engine*, Nov., 1940.

Strub, R. E., "Influences of Production on Design."—N.E. Coast Inst. of Engineers. and Shipbuilders.

CHAPTER XII

VIBRATION OF FRAMEWORK AND SUPPORTS

Types of Vibration considered.—A four cylinder four stroke engine mounted in the frame of a road vehicle vibrates vertically under the influence of unbalanced secondary forces with a frequency equal to twice the R.P.M. If the number of R.P.M. equals half the natural frequency of the engine on its supports, the vibrations have a very appreciable amplitude. The flexibility of front springs, tyres, etc., is such that this critical speed occurs at very low revolutions and over the normal range of R.P.M. the vibrations are small in amplitude. Another critical speed occurs when the main component of torque reaction synchronises with the natural frequency in respect of transverse vibration. The engine mounting and springs, etc., are so designed that this critical speed also is passed through at low revolutions below the normal running range.

Other examples may be taken from marine practice. Engines having 6, 8 or 10 cylinders are frequently arranged in such a way that equal and opposite primary and secondary couples in a vertical plane mutually cancel. The effect of these pairs of couples is to cause a vertical vibration of the centre of the engine room floor.

In such cases if the natural frequency of the double bottom including the mass of the engine is equal to the R.P.M. or twice the R.P.M., such vibrations are liable to be magnified by resonance. The calculation of such natural frequencies is complicated by questions of the degree of fixation of the double bottom at the sides of the ship and the bulkheads and also to what extent the vibrating mass is increased by water under the ship's bottom.

In a twin screw vessel, unless the engines are tied together by athwartship stays, the natural frequency may be lowered by transverse bending of the engine frames resulting in the cylinder tops vibrating symmetrically to and from each other when the engines come into step. The same type of vibration can occur when the natural frequency for this type agrees with that of the main component of torque reaction.

Twin engines tied together or single screw engines are also liable to transverse vibrations under the influence of torque reaction. The gravest natural frequency is something less than the natural frequency in respect of transverse vibration of the engine on a rigid bed.

In yet another type of vibration the end cylinder tops

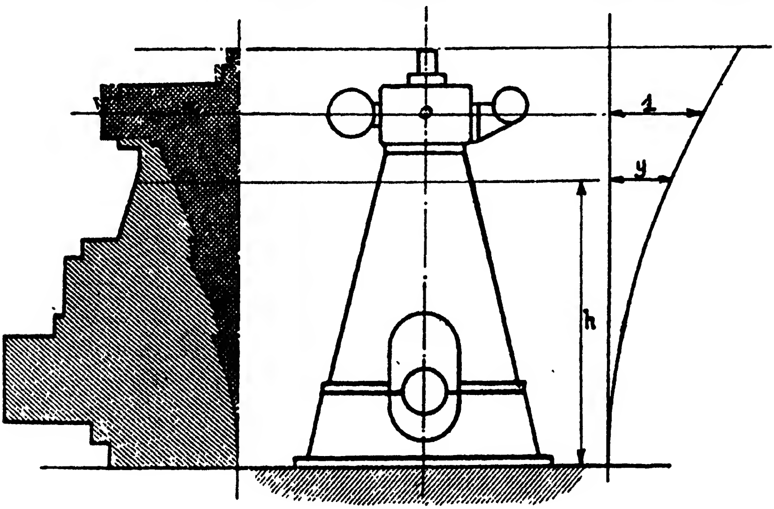


Fig. 109.

oscillate in opposite phase. For example, suppose the firing order of an 8 cylinder two cycle engine is 18347256, then the phase relations of the 5th harmonics of the several cylinders are given by the order 4 1 2 3 6 7 8 5, and if these are plotted round a circle it is found that the resultant of 1 2 3 is nearly opposite that of 6 7 8. If the natural frequency for this type of vibration is 5 times the R.P.M. vibration of this type will be magnified by resonance.

Three and 5 cylinder engines usually have unbalanced pitching couples which in unfavourable circumstances may cause vibration of the supporting structure.

Transverse Vibration on Rigid Bed.—The first step in the approximate method to be described is to divide the total weight W of the engine into the equivalent masses

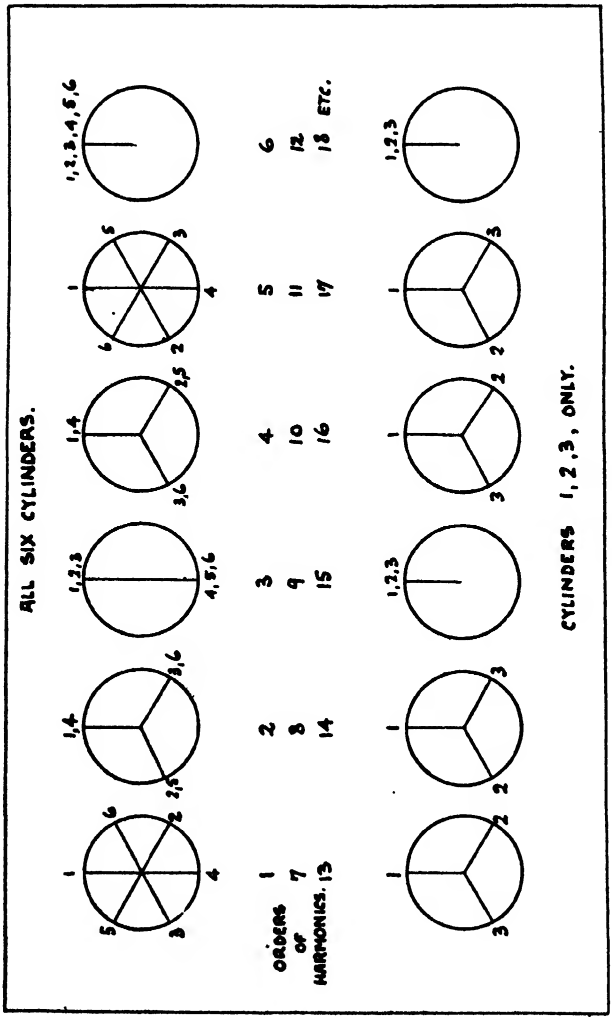


Fig. 110.

W_1 and W_2 located at the middle of the cylinder cover (S.A. engines) or the mean position of the piston (D.A. engines), and the underside of bedplate respectively. This may be done as in Fig. 109. The full line graph to the left represents the weight " w " per inch of height and its area is a measure of W. The curve to the right represents an assumed deflection curve during vibration with $y=1$ at the position of W_1 and $y \propto h^3$ elsewhere. The dotted curve to the left represents wy^2 and its area is a measure of W_1.

Next treat the engine as a cantilever of varying section and find graphically or otherwise the deflection d (inches) of W_1 due to a force W_1 applied horizontally (including deflection due to shear).

The frequency F in vibrations per minute is then given by :—

$$F=\frac{60}{2\pi}\sqrt{\frac{386}{d}} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (1)$$

The element of arbitrariness in the assumed deflection curve can be minimised by finding the deflection curve due to a distributed load wy per inch of height and repeating the process until agreement is obtained. Verification by direct experiment, though very desirable, is not easy as test beds are usually far from rigid.

A single cylinder two cycle engine with a transverse natural frequency of 1200$\sim$/min. will exhibit transverse vibrations due to the various harmonics of torque reaction at the following critical speeds, viz. :—

Order of harmonic .	10	11	12	13	14	15	16 etc.
Critical R.P.M. .	120	109	100	93	86	80	75 etc.

A well braced group of six such cylinders with cranks at equal angles will have transverse criticals (all cylinders vibrating in phase) at 100 R.P.M. (12th harmonic), 67 R.P.M. (18th), 50 R.P.M. (24th), etc. Harmonics of orders not divisible by 6 cancel out, in the vector diagram (Fig. 110), since the displacements of all cylinders are equal.

If the cylinders are arranged in two groups of three with firing order 1 6 2 4 3 5 the vector diagrams for cylinders 1, 2 and 3 take the form shewn in Fig. 110 and criticals occur as follows :—

Order of harmonic .	9	10	11	12	13	14	15	16	17	18
Cylinders in phase .	—	—	—	100	—	—	—	—	—	67
Cylds. 1, 2 and 3, 180° out of phase with cylds. 4, 5 and 6 . . .	134	—	—	—	—	—	80	—	—	—

Other numbers and groupings of cylinders may be investigated similarly.

Transverse Vibrations on Yielding Supports.—A typical example is afforded by a single screw marine engine bolted to a ship's tank top. The method already described can be extended to this case also by drawing the deflected form of engine and tank top on a single diagram and extending the curve of w to include the weight per inch of double bottom plus water, etc.

The weight W_1 so obtained is the equivalent total mass referred to the position of W_1.

The various types of vibration of which twin screw engines (whether tied together or not) are capable, can be dealt with on similar lines.

In each case the natural frequency of the engine on a rigid bed is the *upper* limit of the possible frequency of the *gravest* mode of vibration. In addition to the gravest modes of vibration there are other modes of higher frequency in which the engine or engines vibrate in opposition to the double bottom, that is the engine is bent by inertia in the opposite direction to that in which it is tilted by the deformation of the double bottom.

Literature.—Lewis, F. M., "Vibration and Engine Balance in Diesel Ships."—Soc. N.A. and M.E. New York, November, 1927.

CHAPTER XIII

BALANCING

Preliminary Notes.—The whole subject of the Balancing of Internal Combustion Engines contains groups of ideas which may be summarised somewhat as follows :—

(a) Balance of revolving masses ; primary and secondary balance of reciprocating masses ; balance of torque reaction.

(b) Influence of number of cylinders, spacing of cylinders, grouping of cylinders and angular spacing of cranks.

(c) Influence of the cycle whether four stroke or two stroke, single acting or double acting, on suitable crank arrangements.

(d) Influence of considerations of regularity of turning moment and torsional resonance on suitable crank and cylinder spacings.

(e) Special arrangements of revolving and reciprocating masses for counteracting the effects of lack of balance.

(f) Influence of the nature of the supporting structure on the types of lack of balance, if any, which are tolerable in various circumstances.

For a full discussion the reader is referred to complete treatises on Balancing, some of which are mentioned in the bibliography. The simplest procedure in the following brief survey appears to be to deal first with a few essential preliminaries and then to deal in order with straight-in-line engines of 1 2 3 ....12 cylinders.

Revolving Masses.—A system of masses M_1, M_2, M_3....M_n, see Fig. 111, revolving as a rigid body about a common axis can be completely balanced by two masses B_1 and B_n rigidly attached to the system and revolving in two non-coincident planes. For purposes of calculation it is convenient to take these two planes as the planes of rotation of M_1 and M_n respectively. If the centre of gravity of the original system lies on the axis of rotation then B_1 and B_n, if equally distant from the axis, are equal in magnitude and 180° apart with respect to the axis. If $\rho_1, \rho_2, \rho_3 \ldots\ldots \rho_n$ are the radii vectores of $M_1, M_2, M_3 \ldots\ldots M_n$, then the condition for the C.G. to lie on the axis is :—

$$\Sigma\,[M_r \rho_r]=0 \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (1)$$

If this condition is fulfilled the system is said to be in static balance and the reactions on the bearings during rotation (apart

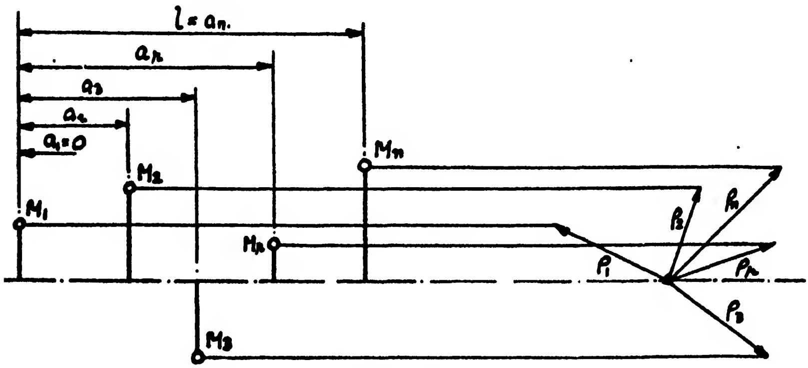


Fig. 111.

from dead weight) reduces at most to a couple. The further condition for dynamic balance is :—

$$\Sigma\,[M_r\,\rho_r\left(\frac{l-a_r}{l}\right)]=0 \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (2)$$

The square brackets in (1) and (2) represent vector addition ; l and a have the significance indicated in Fig. 111. For purposes of calculation it is convenient to reduce M_1, etc., to a constant numerical value of ρ by simple proportion, since the centrifugal force varies as the radius directly. In any given case the mass required at either reference plane (both as regards magnitude and angular position) to balance the given system is found by performing graphically or numerically the operation indicated by the left-hand side of (2) and turning the vector so found through 180°.

Primary Balance of Reciprocating Masses.—Primary balance deals with the fundamental harmonic component of the inertia forces due to the motion of the reciprocating parts. The forces so dealt with are such as would exist if the connecting rod were infinitely long. The primary forces have the same period of alternation as the engine revolutions and vary in simple harmonic manner; accordingly they may be regarded as the vertical projections of centrifugal forces brought about by the revolution of ideal masses equal to the reciprocating masses, about an axis parallel to the crank-shaft. The conditions of balance are therefore the same as for revolving masses and equations (1) and (2) are therefore applied, bearing in mind when interpreting the results that the horizontal components have no real existence.

Secondary Balance of Reciprocating Masses.—Secondary balance deals with the second harmonic components of the inertia forces. The third and higher harmonics are small and usually considered to be of no practical importance. The secondary inertia forces having double the frequency of the crank-shaft rotation and being simple harmonic functions may conveniently be visualised as vertical components of the centrifugal forces of masses rotating at double the crank-shaft speed. The angular relationship of the corresponding vectors for use in equations (1) and (2) are very simply determined by taking any one crank vector as a standard of reference and doubling the angles (in either direction) of the remaining vectors with respect to the standard. The maximum value of the secondary inertia is r/l times the value of the corresponding primary inertia where r=crank radius and l=length of connecting rod.

Primary inertia has maximum value upwards at top dead centre, downwards at bottom dead centre and zero value at crank angle 90°, and $1/\sqrt{2}=\cdot702\times$maximum value at 45° to either dead centre. Secondary inertia has maximum value upwards at top and bottom dead centre, maximum value downwards at crank angle 90°, and zero at 45° to either dead centre and $0\cdot702\times$maximum value at odd multiples of $22\frac{1}{2}$° from top or bottom dead centre.

Torque Reaction.—The nett effect of the mechanical actions between the parts of a reciprocating engine consists in general of a fluctuating periodic torque superimposed on a steady torque. Whilst the use of fly-wheels or balance weights may prevent the fluctuating part from being transmitted (for the most part) to the drive, the full effect of the fluctuating torque is applied to the engine foundation. The only way of completely eliminating the fluctuating torque reaction appears to be to design the engine in the form of a pair of equal and oppositely rotating engines running in step, as in an early motor car engine due to Lanchester. In this case the elimination of torque reaction was independent of the behaviour of individual cylinders, since the opposed cylinders had combustion chambers in common. Practically effective measures against the consequences of variable torque reaction are sometimes carried out in the form of rotating or reciprocating mass systems which at any one speed may be arranged to neutralise the greater part of the objectionable effect.

Angular Spacing of Cranks.—Two stroke engines fire at every top dead centre if single acting and at every top and bottom dead centre if double acting, consequently the spacing of cranks determines the firing sequence.

With four stroke engines alternative firing orders are possible with the same angular spacing of cranks. Some orders of firing are more favourable than others from the point of view of torsional vibration of the crank-shaft and the selection of a suitable spacing of cranks and ignition sequence to give the best results from all points of view may offer many alternatives.

In the interests of good balance it is sometimes necessary to adopt simultaneous firing of cylinders or unequal spacing of ignitions in circumstances in which considerations of uniformity of turning moment would suggest equal spacing of ignitions or non-simultaneous firing.

The angular spacing of the cranks and the sequence of firing have an important influence on the possibilities of torsional resonance. The usual procedure in developing a new design is first to fix upon a spacing of cranks which gives a good balance and then to investigate the system with respect to torsional vibration, trying one after another all the orders of firing to which the system is susceptible. If none of these orders are satisfactory the spacing of cranks must be altered unless it is found possible to alter the natural frequency of the system or introduce a damping device. The problem is thus essentially one of trial and error.

Single Cylinder Engine.

Let W_r = weight in lbs. of revolving parts referred to crank radius.

W_o = weight of reciprocating parts in lbs.

r = radius of crank in inches.

l = length of connecting rod in inches.

n = revolutions per minute.

$g = 386$ in./sec.^{2}.

The centrifugal force of the revolving parts :—

$$F_v = \frac{W_v}{g}\left(\frac{2\pi.n}{60}\right)^2 r = \frac{W_v n^2 r}{3530} \text{ lbs.} \qquad \ldots\ldots\ldots\ldots(3)$$

The maximum value of the primary inertia force is :—

$$F_{c1} = \frac{W_c n^2 r}{3530} \text{ lbs.} \qquad \ldots\ldots\ldots\ldots(4)$$

The maximum value of the secondary inertia force is :—

$$F_{c2} = \frac{W_c n^2 r^2}{3530.l} \text{ lbs.} \qquad \ldots\ldots\ldots\ldots(5)$$

The revolving parts may be completely balanced by a pair of weights, one attached to each crank web, each balance weight being equal to $\frac{1}{2}W_v$ when referred to crank radius. (In future the words "referred to crank radius" will be assumed to be understood.)

The primary and secondary forces can only be balanced either by arrangements of masses reciprocating at engine speed and double engine speed respectively or by pairs of masses revolving at these speeds. Such arrangements are usually prohibitive on single cylinder engines on account of cost.

By fitting additional balance weights equivalent to W_c the primary inertia forces may be eliminated, but horizontal forces equal to the pre-existing vertical forces are thereby introduced. The choice of evils will depend upon whether vertical or horizontal forces are likely to be more objectionable. A compromise is sometimes made by fitting balance weights equivalent to the revolving masses plus $\frac{1}{2}$ to $\frac{2}{3}$ the reciprocating masses.

Two Cylinder Engine.—(*a*) Cranks on the same centre.

This arrangement gives equal spacing of ignitions in a four stroke engine and simultaneous firing in a two stroke. As regards balance it is similar to a single cylinder engine. Equal balance weights on all four crank webs are indicated.

(*b*) Cranks at 180°.

Equal spacing of ignitions in two stroke engine but very unequal in four stroke. Secondary forces are fully out of balance as in (*a*) or a single cylinder engine. The revolving forces and primary reciprocating forces produce couples F_v L and F_{c1} L where L=cylinder centres. The revolving couple can be balanced by a pair of weights each equal to

$$\frac{W_v L}{2(L+2a)} \dots\dots\dots\dots\dots\dots\dots\dots (6)$$

attached to the outer crank webs only, where (a) is the distance from a balance wt. C.G. to the nearest cylinder centre line. This arrangement has the advantage of relieving the centrifugal loading of the outside bearings. The centre bearing is already relieved by opposing forces.

The reciprocating couple can be balanced wholly or in part by a suitable increase of the above balance weights at the expense of introducing a horizontal couple. Whether this is worth while depends on whether a vertical or horizontal couple is likely to be the more objectionable. This remark applies in all similar cases and need not be iterated.

Three Cylinder Engine.—The usual arrangement of cranks at 120° gives equally spaced impulses both for two stroke and four stroke engines. Primary and secondary forces are balanced but there then are primary and secondary unbalanced couples. Referring the revolving (or primary reciprocating) masses to reference plane passing through the axis of No. 1 cylinder, the referred masses are in the ratio 2, 1, 0 or 1, $\frac{1}{2}$, 0 for Nos. 1, 2 and 3 respectively. Combining these with respect to their direction by a vector polygon the resultant has a magnitude $=\sqrt{1^2-(\frac{1}{2})^2}=0{\cdot}866$ lying in a plane $\perp$ to crank No. 2. The appropriate balance weights for balancing the revolving masses therefore lie in this plane and make angles of 120° with cranks 1 and 3 as shewn in Fig. 112. If a balance weight is fixed to each web of cranks 1 and 3 their magnitudes are each $0{\cdot}433\ W_v$. These are indicated by B_1B_1 and B_3B_3 in diagram I. The state of balance is not disturbed by adding the self balanced system of opposed pairs of equal balance weights indicated in diagram II. If each of these weights are given a magnitude $0{\cdot}25\ W_v$ and combined with the above, the result is a system indicated in diagram III. The inner balance weights of cranks 1 and 3 remain as before; the outer balance weights of 1 and 3 are now at 180° to these cranks and each have a magnitude of $0{\cdot}5\ M_v$. Both weights attached to No. 2 crank are at 180° thereto and are of magnitude $0{\cdot}25\ W_v$. This arrangement has advantages from the point of view of local balance and relief of inertia loading of bearings.

An obvious alternative consists of fixing a balance weight

0·5 W_v to every crank web at 180° to the crank-pin, but this involves a greater total weight of balance weights and decreases the natural vibration frequency of the system which is usually disadvantageous from the point of view of torsional vibration.

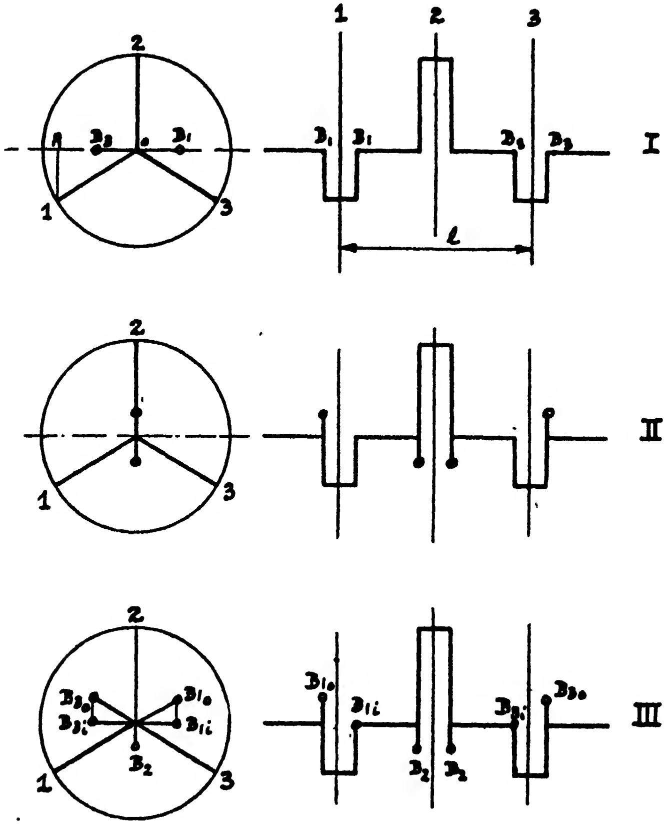


FIG. 112.

Four Cylinder Engine.

(a) Cranks @ 180°.

The arrangement of four cranks in one plane with the outer cranks @ 180° to the inner is a common one for four stroke engines, since it gives equal spacing of ignitions. It is balanced for primary forces as well as for couples both primary and secondary, but the secondary forces are fully unbalanced.

The secondary forces may be balanced by an arrangement consisting of a pair of masses revolving at double engine speed in opposite directions, disposed symmetrically on either side of the engine at the centre. The magnitude of each such mass referred to crank radius is,

$$\frac{W_c r}{2l} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (7)$$

These weights should be on top dead centre when the engine cranks are horizontal and on bottom dead centre when the engine cranks are in a vertical plane.

The local balance of revolving parts may be improved by balance weights on the outer webs of Nos. 1 and 4 and the inner webs of Nos. 2 and 3 at 180° to their respective crank-pins, the magnitude of each being $\frac{1}{2}\ M_v$ or $\frac{1}{2}\ (M_v+\frac{1}{2}M_c)$. There appears to be no object in attaching balance weights to the inner webs of Nos. 1 and 4, and the outer webs of 2 and 3 as such weights would practically cancel out so far as their effect on adjacent bearings is concerned.

(b), (c), (d) Cranks @ 90°.

Four cylinder two stroke engines usually and four stroke engines sometimes, have their cranks arranged at 90°. Three arrangements denoted by (b), (c) and (d) in Fig. 113, are possible. In all these there is balance of primary and secondary forces, but all three involve unbalanced primary couples and arrangement (c) is the only one free from an unbalanced secondary couple.

The revolving masses referred to the plane of No. 1 crank are in the ratio, 1, 2/3, 1/3, 0 for cranks 1, 2, 3, and 4 respectively. Their resultants in the three cases are as follows :—

(B) $\frac{\sqrt{2}}{3} W_v \qquad = 0{\cdot}471\ W_v$

(C) $\sqrt{1^2+(\frac{1}{3})^2} W_v = 1{\cdot}052\ W_v$

(D) $\frac{2\sqrt{2}}{3}\ W_v \qquad = 0{\cdot}942\ W_v$

The diagram for secondaries is given at the foot of the figure, from which it results that the secondary force in the reference plane bears the following ratio to that of a single cylinder in the three cases, viz. :—

(B) 4/3.

(C) 0.

(D) 2/3.

Local balance in respect of revolving masses may be improved, as in the case of three cylinder engines, by the addition of equal and opposite pairs of balance weights forming a self-balanced system.

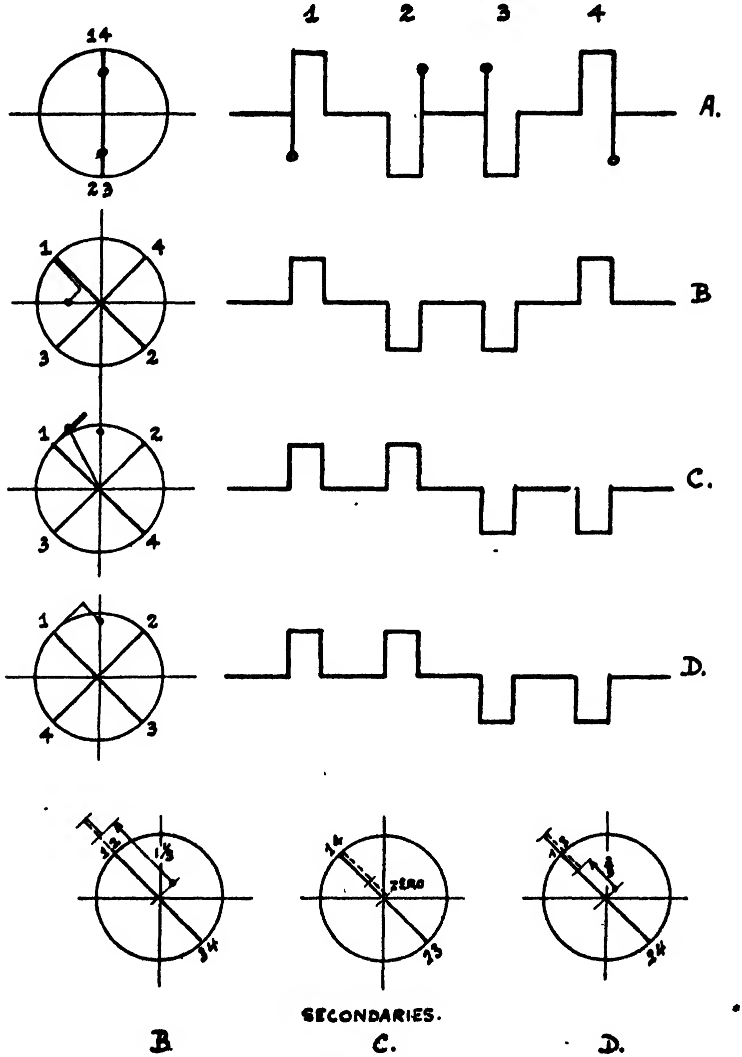


Fig. 113.

By "overbalancing" arrangement C it is possible to make the unbalanced couple wholly horizontal and obtaining complete balance vertically.

The possible orders of firing of a 4 cylinder four stroke engine with cranks at right angles are numerous and all involve rather irregular turning effort diagrams, which necessitate a rather heavy fly-wheel if a fine degree of regularity is required. If a, b, c, d represents the sequence of cranks the best sequence

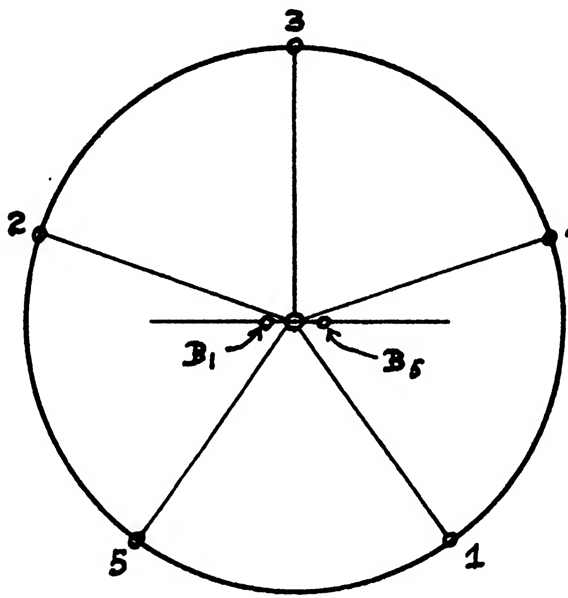


FIG. 114.

of firing from the point of view of regularity is ab—d——c— where the dashes represent non-firing dead centres.

Five Cylinder Engines.—The usual arrangement of cranks at 72° gives uniform spacing of ignitions both with two and four stroke engines. Primary and secondary forces are balanced but there are unbalanced couples of both kinds. There are 12 possible numbered crank sequences and the best of these from the point of view of reducing the unbalanced primary couple to a minimum is 1 5 2 3 4. With this sequence the unbalanced revolving masses referred to No. 1 and No. 5 crank planes are $0{\cdot}11W_r$ at the angle indicated in Fig. 114, and can be balanced by quite small counter weights. The corresponding unbalanced primary reciprocating couple is not usually serious, but it can of course be eliminated by over-balancing at the expense of introducing an equal unbalanced horizontal couple. The unbalanced secondaries referred to planes Nos. 1 and 5 have a value equal to 1·245 that due to a single cylinder.

The alternative sequence 1 3 5 4 2 interchanges the above two values for the primary and secondary unbalanced reciprocating masses respectively.

Six Cylinder Engines.—The almost universal arrangement for 6 cylinder four stroke engines consists of two symmetrically placed arrangements of 3 cranks at 120°. Cranks 1 and 6 are on the same dead centre, and likewise with cranks 2 and 5, 3 and 4. Each system of 3 cranks is balanced for forces both primary and secondary, and the symmetry of the arrangement cancels opposing couples. There are 4 orders of firing, giving equal spacing of ignitions, viz. :—

1 5 3 6 2 4, 1 5 4 6 2 3, 1 2 3 6 5 4, 1 2 4 6 5 3.

The first of these has been commonly used, but the last gives a wider region of speed free from serious torsional vibration in the neighbourhood of resonance with the 9th harmonic if a heavy fly-wheel is fitted, and is sometimes preferred on that account.

Local refinement of balancing can be used, as described, under 3 cylinder engines.

The above symmetrical arrangement of 6 cylinder crank-shaft can also be used with two stroke engines if no exception is taken to the cylinders firing in pairs.

The more usual arrangement of 6 cylinder crank-shaft for two stroke engines consists of cranks arranged at 60°, giving equal spacing of ignitions without simultaneous firing. With all such arrangements primary and secondary forces are balanced, but the couples depend on the crank sequence. The sequence 1 6 3 2 5 4 was at one time in common use; it will be noted that this arrangement consists of three pairs of adjacent cranks at 180°, so that the crank-shaft can be built up of three interchangeable sections of simple form from a manufacturing viewpoint. Primary couples are balanced, but the secondary force referred to a reference plane at No. 1 or 6 crank amounts to $4/5.\sqrt{3}=1{\cdot}38$ times that of a single cylinder. The separate halves of the engine are not in good balance.

A sequence now preferred is 1 6 2 4 3 5. Primary couples are balanced and the secondary force referred to a reference plane at No. 1 or 6 amounts only to $2/5.\sqrt{3}=0{\cdot}69$ times that of a single cylinder. Each half of the engine is self balanced for primary and secondary forces and the local balance of each half may be improved as described for a 3 cylinder engine.

Seven Cylinder Engines.—Cranks spaced at $360\div7=51^\circ-26'$ give equally spaced ignitions both for two and four stroke engines with balance of primary and secondary forces. There are 360 possible crank sequences and the best of these appears to be 1 6 5 3 2 7 4, giving complete balance of primary couples. The secondary inertia force referred to a reference plane at No. 1 or 7 crank amounts to 1·53 times that of a single cylinder.

Eight Cylinder Engines.

(a) Cranks at right angles.

Symmetrical duplications about the centre plane of the 4 cylinder arrangements, shewn in Fig. 113 give complete balance of primary and secondary forces and couples. Ignitions are equally spaced with four stroke engines, and may be so with two stroke engines if simultaneous firing of symmetrically placed cylinders is allowed.

With four stroke engines the crank sequence 18, 45, 27, 36 giving the smallest primary couple (but the greatest secondary couple) in each half of the crank-shaft is commonly preferred. With this sequence there are eight possible firing orders, the choice between which may be influenced by considerations of torsional vibration. The two following appear to be favourable in typical instances, viz.: 1 4 7 3 8 5 2 6 and 8 4 7 3 1 5 2 6.

(b) Cranks at 45°.

The customary arrangement for two stroke engines is for the cranks to be spaced at 45°, giving equal spacing of ignitions without simultaneous firing and balance of primary and secondary forces. Eighteen crank sequences have been investigated in detail by Mr. P. Cormac (see p. 256). The sequence 1 8 2 6 4 5 3 7 gives balance of secondary couples, but there is a primary unbalanced couple corresponding to referred forces at the planes of cranks 1 and 8 amounting to 0·064 times that due to one cylinder. This can be eliminated by increasing the spacing between cylinders 2 and 3, and 6 and 7 by the factor 1·414. The sequence 1 8 6 2 4 5 7 3 gives balance of both primary and secondary couples if the pitch between cylinders 4 and 5 is 1·828 times the pitch of the remaining cylinders, which is a very convenient allowance of space for the accommodation of the crank-shaft couplings and valve driving gear.

Nine Cylinder Engines.—An arrangement of cranks at 40° gives equal spacing of ignitions both for two and four stroke engines. The crank sequence 1 9 2 7 4 5 6 3 8 results in unbalanced primary and secondary couples, the effects of which when referred to the planes of cranks 1 and 9 amount to 0·024 and 0·068 times the effect due to one cylinder.

Ten Cylinder Engines.—A symmetrical arrangement consisting of two groups of five cranks at 72° gives complete primary and secondary balance and equal spacing of ignitions for a four stroke engine. Each half crank-shaft is preferably arranged in the sequence 1 5 2 3 4, giving a minimum primary couple. There are numerous alternative orders of firing and the choice is determined by considerations of torsional vibration. If the node is at the centre the best orders of firing appear to be :—

	1	2	4	5	3	10	9	7	6	8
or	1	2	4	6	3	10	9	7	5	8

For equal spacing of ignitions of a two stroke engine (excluding simultaneous firing) the cranks must be arranged at 36° ; the possibilities are very numerous. One way of arriving at possible arrangements is to combine a five cylinder unit with cranks at 72° with another such unit turned through 180°. This cancels out any primary couples but leaves the secondary couples unbalanced. With this procedure it is evidently desirable to select the type of 5 cylinder unit which has the smallest unbalanced secondary couple.

Twelve Cylinder Engines.—For equally spaced ignitions of four stroke engines the cranks are at 60°. Two 6 cylinder balanced engines end to end with the cranks of one engine 60° or 180° in advance of, or behind, the corresponding cranks of the other provide possible solutions. Simultaneous firing of symmetrically placed cylinders (cranks at 120°) has been adapted on engines without fly-wheels in order to avoid torsional resonance. For two stroke engines with equal spacing of ignition and no simultaneous firing the cranks are at 30°. A 6 cylinder unit with cranks at 60° combined with another such unit turned through 90° provides a possible solution which automatically conceals any secondary couples existing in the two units separately. Mr. P. Cormac (*Engineering*, October 11th, 1929, p. 460) has enumerated 36 firing orders (counting both ways) which give complete primary and secondary balance. It is a matter for trial in any given instance to ascertain which order is most favourable from the point of view of avoiding torsional resonance.

Flat Opposed Engines.—If two equal cylinders are arranged coaxially mouth to mouth with their connecting rods connected to the same crank-pin, then clearly one piston will arrive at top dead centre when the other arrives at bottom dead centre. The primary inertia forces due to the two pistons are additive but the secondary inertia forces mutually cancel out. The conditions for complete balance of flat opposed engines are therefore the same as that for primary balance of straight line engines having the same number of cranks. In particular the 2, 4 and 6 cylinder flat opposed engines are subject to an unbalanced primary force or couple, but the 8 cylinder is in complete balance.

Vee Engines.—If two equal cylinders are arranged with axes in a plane and intersecting at 90° and with connecting rods connected to a common crank-pin, the primary inertia forces due to the two pistons combine to form a rotating vector which can be balanced by a counter weight opposite the crank-pin. Secondary forces remain out of balance. This principle may be used to build up completely balanced arrangements from a pair of straight line units which separately are unbalanced in respect of primary couples, but balanced in respect of secondaries. For example, 4 cranks @ 90° with sequence 1 2 4 3 give balance of secondaries in a straight line engine, but there is an unbalanced primary couple which can be eliminated by combining with another block of four cylinders in a plane at 90° to the plane of the first block and adding a balance weight opposite each crank. Two cylinders arrive at T.D.C. every 90°, so that ignitions are equally spaced in a four stroke engine, but there would be simultaneous firing in a two stroke engine.

As another example a Vee 12 two stroke engine could be arranged with two banks of 6 cylinders at an angle of 90°, the cranks being arranged at 60° with sequence 1 3 5 6 4 2 or 1 3 2 6 4 5.

If the number of cylinders in each bank is such that complete balance is possible for a straight line engine having that number of cylinders then the angle of the V may have any value which may be convenient from considerations other than those of balance, e.g. torsional resonance or space considerations.

Literature.—Wilson, W. K., "The Balancing of Oil Engines"—Griffin.

Cormac, P., "The Design of Dynamically Balanced Crankshafts for Two Stroke Cycle Engines"—*Engineering*, Oct. 11, 1929.

CHAPTER XIV

CYLINDERS AND COVERS

General Types.—The great majority of Diesel Engines are provided with cylinder liner, jacket and cover as separate pieces, as in Fig. 115, which refers to a four cycle trunk engine. Different arrangements have, however, been used successfully, and deserve mention. With small engines, simplification is achieved by casting the jacket and liner in one piece, as in Fig. 116. Remembering that the bulk of the jacket wall remains

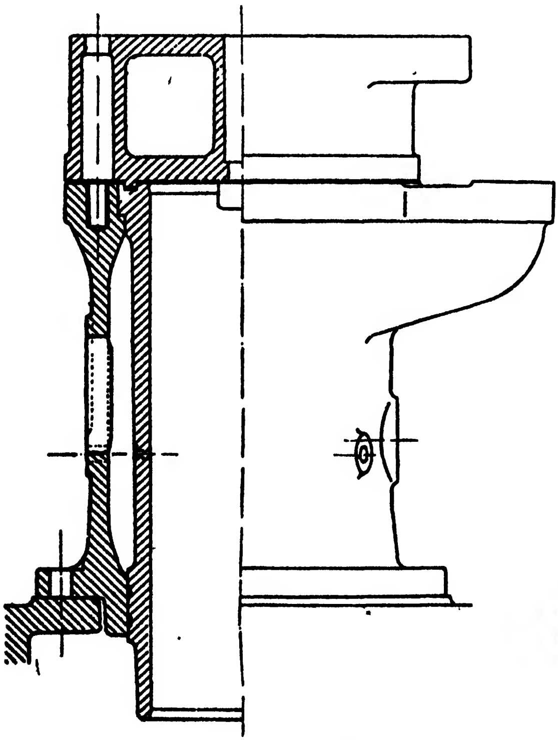


Fig. 115.

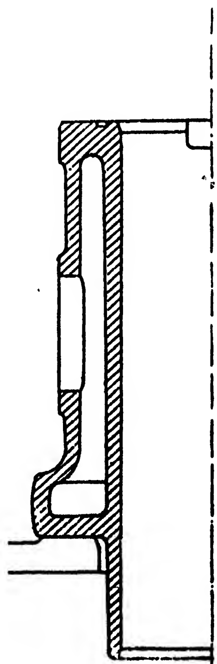


Fig. 116.

stone cold, it will be appreciated that this construction involves increased tensile stresses on the jacket, due to the tendency of the liner to expand, and jackets of this kind have been known to crack circumferentially. In cases where staybolts have been fitted to carry the tensile stresses from the cover downwards little damage has resulted. On the other hand, when the jacket has been relied on for this function, rupture during working may easily occur, and has sometimes resulted in the cylinder being projected towards the roof. These considerations would appear to indicate that the use of this construction, without staybolts or other safeguards, is not lightly to be attempted without serious consideration of the capabilities of the foundry. In Fig. 117 is shewn a construction in which the cover is incorporated with the cylinder casting in motor-car style. In this case the tensile stresses are mainly carried by the liner, and the jacket is made relatively thin and flexible. This design, though probably safer than that of Fig. 116, also makes some demand on the skill and care of the foundry people. In this connection it is worth while bearing in mind that many failures might possibly have been avoided if it had been realised that certain special designs, in which lightness has been the primary consideration, were only practicable if the greatest care were exercised in the selection of material and in making the castings. There are other types of cylinder in successful use, notably those in which the liner and cover are cast together apart from the jacket, but this chapter will be very largely devoted to the consideration of the details of the more common construction, in which the liner jacket and cover are separate pieces. Unless the contrary is stated, cast iron is understood to be the material in each case.

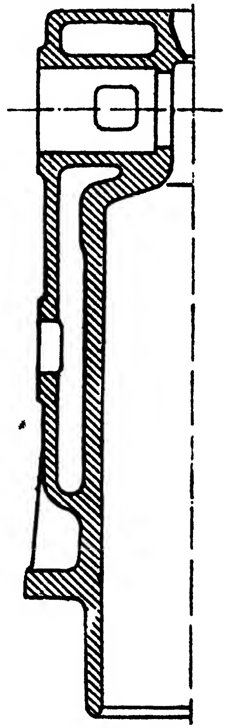


FIG. 117.

Liners.—Special cast iron is used for liners, but there is little unanimity of opinion as to the most desirable properties beyond the obvious requirements of soundness and homogeneity. The greatest difficulty to be overcome is abrasion by the piston rings. At present it seems open to question whether the problem is most influenced by the material of the liner or the piston rings themselves. Four stroke liners very seldom crack except on the occasion of the seizure of an uncooled piston. This immunity is traceable to the very moderate heat flux to which four stroke cylinders are generally subject. Two stroke liners of large size are liable to crack in course of time at the breech end if the flange is unduly heavy. The subject of the

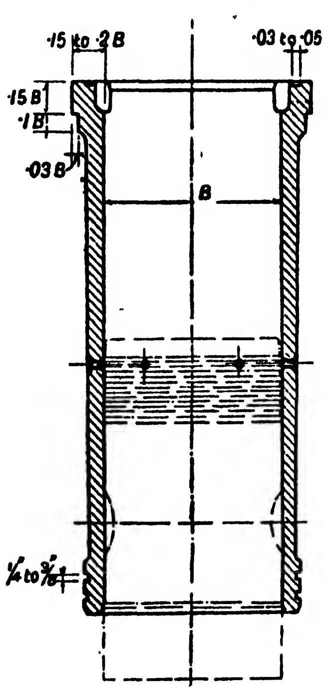


Fig. 118.

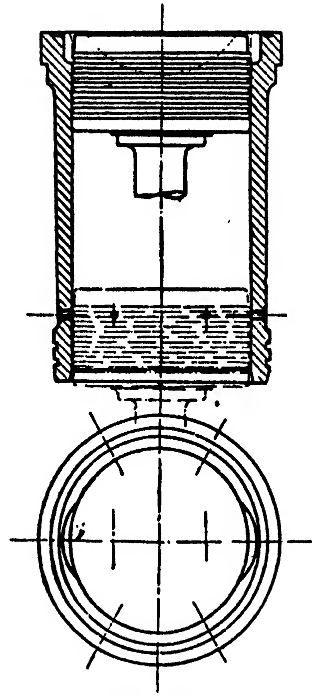


Fig. 119.

conduction of heat through the walls of the combustion space and the stresses induced thereby, is of great importance in connection with large engines particularly and will be considered later.

Typical liners for four stroke engines of the trunk and crosshead types respectively are shewn in Figs. 118 and 119. With the latter the piston is only of sufficient length to carry the rings, and the length of the liner is determined by the position of the bottom ring at the bottom of the stroke. With the trunk engine the liner must be long enough to embrace a sufficient length (about equal to the bore) of the parallel part of the piston when at the bottom of its stroke, in order to avoid a piston knock at the bottom dead centre. It is therefore necessary to determine the clearance volume and complete the design of the piston before the length of the liner can be fixed finally.

The bore is usually parallel with four stroke engines and slightly barrelled in way of the ports in two stroke engines to allow for the restraints which are inevitably placed at that position against free expansion of the liner. Probably the best bore is produced by finishing with a reamer in a vertical machine. Grinding is frequently adopted, but there is a question if this process does not to some extent destroy those properties of cast iron which facilitate good lubrication. The outside surface is sometimes left unmachined in competitive work, and there is probably no serious objection to this practice for four cycle work. For two cycle engines it seems reasonable to take advantage of the increased heat conductivity obtainable by removing the skin.

Strength of Liners.—The upper end of the liner is subject to a working pressure of about 600 to 700 lb./in.^{2} and the thickness at this part measured under the heavy top flange may be found by the following formula, which represents average practice for substantial engines :—

$$\text{Thickness} = B/12{\cdot}5 \text{ to } B/11$$

The working stress being about 3000 lb. per sq. in. in the case of a 30 in. cylinder, and less in smaller sizes ; explosions at starting, etc., may nearly double this stress occasionally. Unfortunately the available information on the effects of repeated stress is not sufficiently complete at present to enable one to say definitely whether or not these excesses of stress have any influence on ultimate failure by fatigue, but the writer is inclined to believe (on the strength of such evidence as has come before his notice) that the elimination of these occasional excess pressures would not enable any substantial reduction of thickness to be effected with the same margin of safety.

On account of the diminution of pressure on expansion the liner may be tapered to a thickness of about 0·04 bore at the open end.

The breech end of the liner requires to be reinforced by a heavy flange, to avoid distortion due to the pressure of the cover on the spigot joint. Proportions are given in Fig. 118.

Points of Detail.—The difficulty of accommodating the valves in the limited space available in the cover of a four stroke engine usually renders it necessary to make recesses in the top of the liner to clear the air and exhaust valve heads (see Fig. 118).* Two or more tapped holes are provided in a circumferential line round the liner to accommodate the lubricating fittings, these being drilled when the liner is in position in the jacket. The holes are located at about the level of the second piston ring (counting from the top), when the piston is at the bottom dead centre. The fittings themselves will be described later. The water-joint between liner and jacket at the lower end may be made by one or more rubber rings. The joint between cover and liner may be of copper or ground metal to metal.

Two stroke liners are complicated by exhaust, and sometimes air ports (see Fig. 122). In the earlier designs the bars between the latter were always provided with water passages, which introduced difficulties in manufacture, and the value of which seems doubtful, and these are now frequently omitted.

Cylinder Jackets.—A simple and effective form of jacket for a four cycle engine is shewn in Fig. 115, and in this example the jacket takes the pressure pull without the assistance of staybolts. The chief points to be observed are :—

(1) A heavy flange at the top to carry the liner and to enable the tensile forces concentrated at the studs to distribute themselves uniformly round the jacket without producing high local stresses.

(2) A nearly plain cylindrical barrel, as nearly as possible in line with the pitch circle of the cover studs and provided with sludge doors, bosses for lubricating fittings, and a bracket for supporting the cam-shaft bearings.

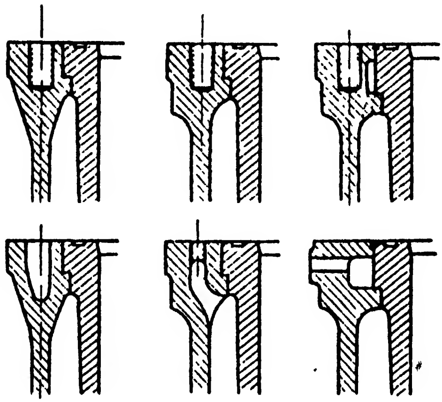
(3) A circular flange at the bottom for securing to the crankcase.

The remarks *re* tensile forces under heading (1) apply here also, but to a less degree, as the studs are pitched closer together than would be feasible on the cover. On these considerations the thickness of the jacket for equal strength should

* This statement applies to flat-bottomed cylinder covers; with dished covers (see Fig. 129) recesses in the cylinder liner may be avoided.

taper gently towards the middle, and the form shewn in the figure is the practical compromise. Some of these points will be considered in greater detail.

Top Flange of Cylinder Jacket.—In small engines this may be solid, but with larger sizes, say from 15 in. bore and upwards, difficulty is sometimes experienced in obtaining sound metal at this point, and coring of the flange between the studs is resorted to in order to accelerate cooling in the mould. Different constructions are shewn in Fig. 120. Schemes A and



C. B. A

FIG. 120.

B have the additional advantage of increasing the cooling surface. Where four cycle engines are concerned the importance of this consideration is probably negligible. Scheme B requires a water outlet connection between each stud if air pockets are to be avoided. On the other hand, the expense of coring is less than with scheme C.

Barrel of Cylinder Jacket.—This is sometimes conical, instead of cylindrical, and in this case it is reasonable to provide a vertical internal rib under each stud to discount the additional stress involved. Consideration of manufacturing costs, and of the good appearance of the engine, rule out of court any form of external ribbing. The brackets supporting the valve gear take many forms in different designs. That shewn in Fig. 115 is a simple form, and considered in conjunction with the gear it supports appears to combine most advantages, including that of elegance.

Bottom Flange of Jacket.—If four staybolts are provided for each cylinder, these may conveniently be used to secure the latter to the crank-case. The concentration of the tensile load at four points necessitates a heavy flange, preferably of box form, as shewn in Fig. 121. The corners of this flange being each subject to a load of one-quarter of the maximum working pressure load, deserve attention in the form of a calculation of the bending stress involved. A plain, square shape would appear to be preferable to some of the more elaborate shapes which have occasionally been used, the flat sides lending themselves well to the provision of facings for various purposes.

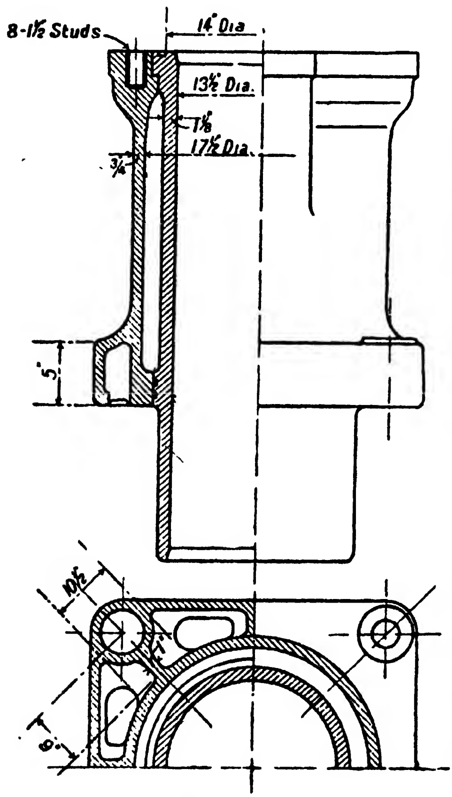


FIG. 121.

Frequently the flange is spigoted into the top of the crank-case, but as this involves an unnecessary machining operation on the latter and makes cylinder alignment more difficult, the better practice is to core the aperture in the crank-case sufficiently large to allow for adjustment of the position of the cylinder and to locate the latter by means of two steady pins.

Strength of Four Stroke Cylinder Jackets.—The considerations of strength which enter into the design of a cylinder jacket are illustrated by the following check calculations relating to the cylinder shewn in Fig. 121.

Bursting stress in liner

$$=\frac{600\ (\text{lb. per sq. in.})\times 6{\cdot}75}{1{\cdot}125}=3600\ \text{lb. per sq. in.}$$

Nominal pull in each cover stud $=\dfrac{600\times 0{\cdot}784\times 14^2}{8}=11{,}600$ lb.

Permissible nominal load for 1¾ in. stud, according to table on page 225, 15,000 lb.

Maximum working pull in jacket $=0{\cdot}784\times 13{\cdot}5^2\times 600=$ 86,000 lb.

Tensile stress in jacket $=\dfrac{86{,}000}{\pi\times 18{\cdot}25\times 0{\cdot}75}=2000$ lb. per sq. in.

Owing to the peculiar shape of the bottom flange the calculation of its strength presents a difficulty which is easily evaded by substituting for the actual section a simpler one of obviously inferior strength.

Nominal load at each corner, 86,000 lb. $\div 4=\sim 21{,}000$ lb.

Moment from centre of bolt to jacket wall $=21{,}000\times 9$ in. lb.

Modulus of hypothetical section :—

$$z=\left(\frac{10{\cdot}5\times 5^3}{12}-\frac{9{\cdot}5\times 3{\cdot}5^3}{12}\right)\div 2{\cdot}5=30{\cdot}2\text{ in.}^3$$

$$\therefore\ \text{Stress}<\frac{21{,}000\times 9}{30{\cdot}2}\quad \text{i.e.} <6300 \text{ lb. per sq. in.}$$

In view of the unfavourable assumptions this is probably not excessive for first-class cast iron.

Jackets for Two Stroke Engines.—The necessity for providing exhaust passages or belts, and in some cases passages for scavenge air as well, introduces considerable complication into the design, renders the stresses in certain parts more or less indeterminate, and makes greater demand on the skill of the manufacturing departments, in comparison with that required by four cycle construction.

Referring to Fig. 122, it will be seen that the exhaust belt interrupts the vertical line of the jacket wall, and if the latter has to carry the main tensile stresses internal ribbing becomes a necessity. The arrangement shewn is perhaps as good as any, but the attachment of ribs to the exhaust belt has a restraining influence on the temperature expansion of the latter which can only result in mutual stresses. It appears, however, that these are not very serious, as cylinders which have failed in other respects have remained intact at this point. Fig. 123 shews a construction in which a good attempt is made to secure continuity of the vertical wall of the jacket. Either of these systems is probably satisfactory for cylinders of medium size. Large cylinders, however (and this applies to

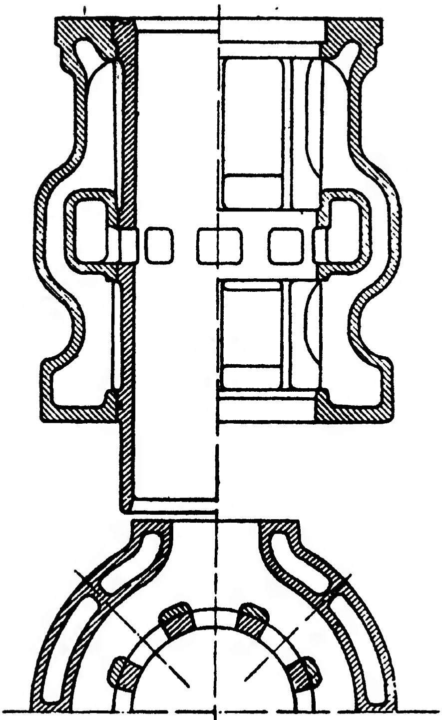


FIG. 122.

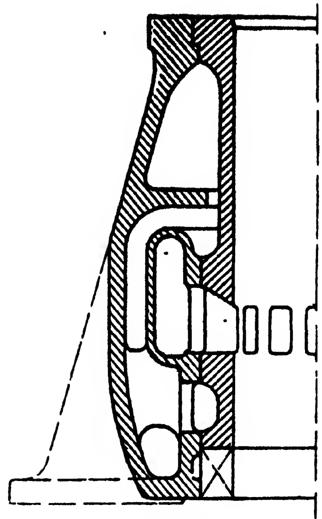


FIG. 124.

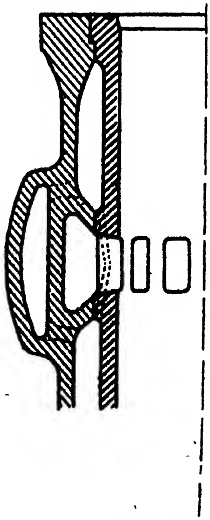


FIG. 123.

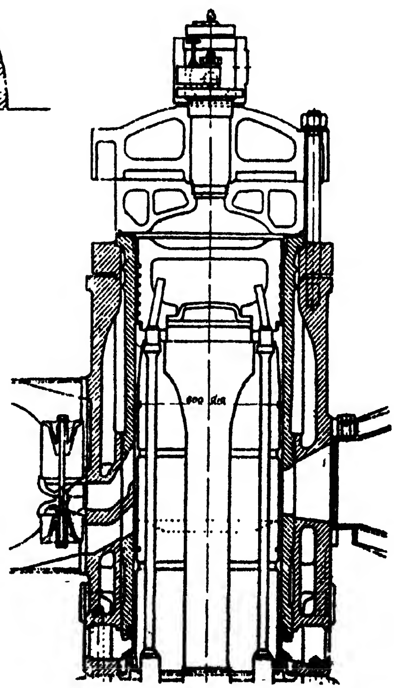


FIG. 124A.

other parts as well), are known to be subject to greater temperature differences than smaller ones (though not to the extent sometimes suggested), and the leading designers have had recourse to other expedients when faced with the problem of constructing cylinders of large size.

In Fig. 124 the jacket wall may be described as similar to a honey-pot in shape and of abnormal thickness, to allow for the bending stresses caused by the curvature of the walls and the fact that the tensile supporting forces are localised at two feet.

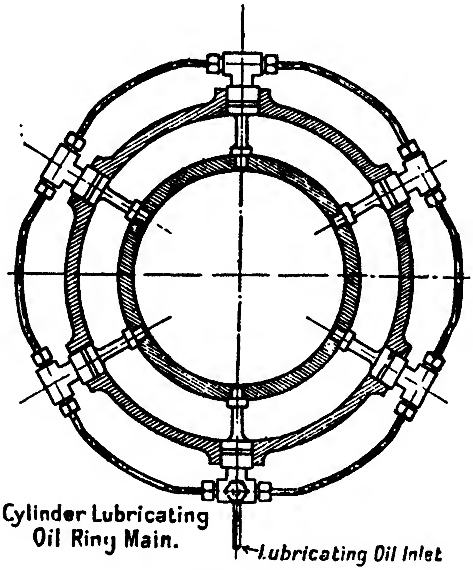
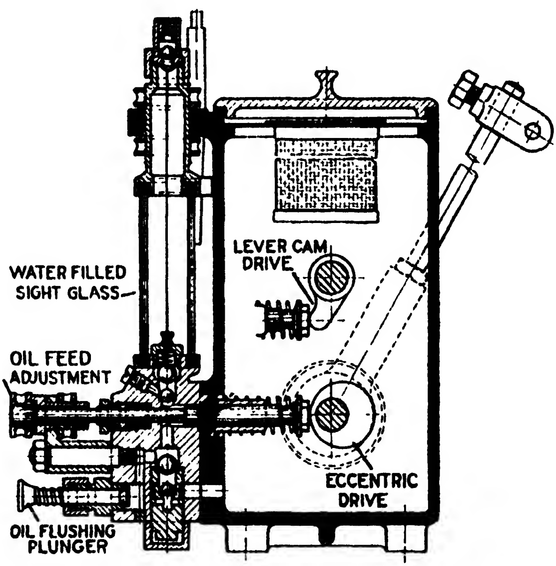


FIG. 125A.

The exhaust belt is of relatively thin metal, with comparatively small support from the walls. It will be evident that the strength of the jacket is very slightly influenced by the exhaust belt, and that the latter is free of all but temperature stresses. This construction, therefore, attains a good approximation to the correct allocation of the respective duties of jacket and exhaust belt.

A further example (Sulzer crosshead type engine) is shown in Fig. 124A. See also sectional drawing Fig. 9 and page 285 *et seq.*

Another and perhaps better way out of the difficulty is to connect the cylinder cover to the bedplate by means of stay-bolts, thus relieving the jacket of all stresses except those induced by temperature differences. The jacket in this case virtually hangs from the cylinder cover, and only requires to be attached thereto by studs proportioned to a load based on the cylinder pressure and the annular area lying between the cylinder bore and the spigot at which the cover joint is made. The upper flange is preferably made fairly substantial, but other thicknesses may be made a practical minimum.



WAKEFIELD 'DP' TYPE MECHANICAL LUBRICATOR

FIG. 125B.

Cylinder Lubrication.—The problem of cylinder lubrication in Diesel Engines consists in effecting uniform distribution of minute quantities of oil. The quantity of oil admitted must be the minimum necessary to effect satisfactory lubrication, as the oil " cracks " in service, leaving a gummy deposit, which in course of time causes the piston rings to stick. Under favourable conditions this may take several months, even two years. Every drop of superfluous oil reduces this period, hence the importance of uniform distribution so that every part may have sufficient, but none a superfluity. These conditions are best secured by a forced feed lubricator discharging at two to eight points round the circumference of the cylinder. A typical lubricating fitting is shewn in Fig. 125 (pages 267–269), and the point to be observed is that the fitting must adapt itself to slight relative movement between the liner and jacket. The small hole at the end which leads to the surface of the liner reduces to a minimum the chances of the fitting becoming choked with carbon. With forced lubricated engines, in which the cylinder is not isolated from the crankpit, it frequently happens that more than sufficient oil reaches the cylinder, apart from any arrangements made for the purpose. In this case the problem may be to devise scraper rings, vent holes, or other devices, to remove the superfluous oil.

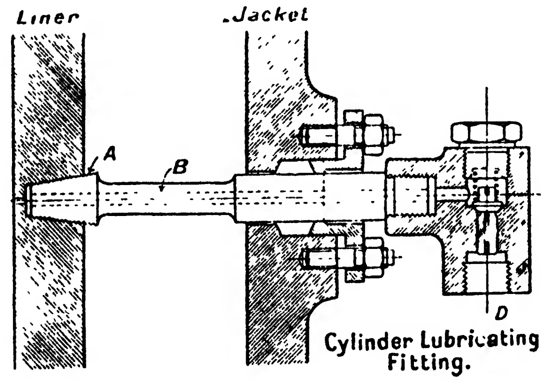


FIG. 125c.

Cylinder Covers.—Owing to a considerable number of failures in service and difficulties experienced in manufacture, cylinder covers for both four and two cycle Diesel Engines have come to be regarded as difficult pieces of design, and it may perhaps be instructive to review the subject in a more or less historical manner.

The earlier type of four cycle cover is shewn in Fig. 126, from which it will be seen that the internal coring is complicated and that a few core-holes of small diameter only are provided to vent the core in the mould. In spite of these disadvantages, such covers have given good service when made by the most skilful and experienced manufacturers. Dismissing for the moment the question of manufacturing costs, these covers have the following shortcomings :—

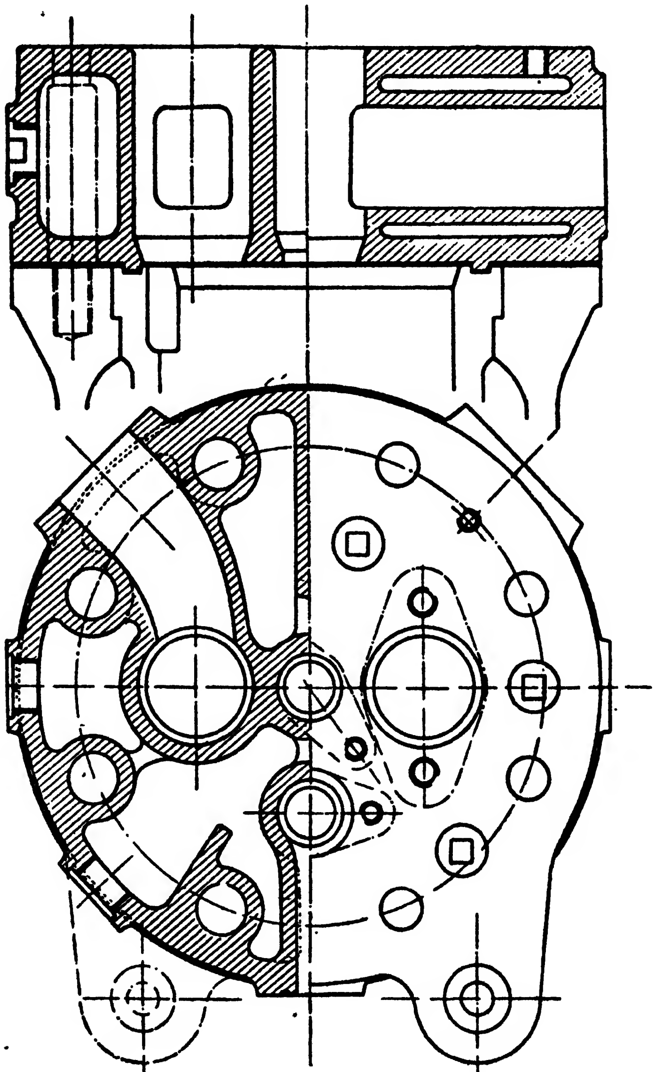


FIG. 126.

(1) The thin walls of uncooled metal between the recesses for fuel and exhaust valves are liable to crack on overloaded engines.
(2) The hot exhaust passage is too rigid to permit of much expansion and leads to cracking of the bottom plate.
(3) The small core-holes give poor access to the interior for purposes of cleaning away accumulated scale.

Assuming first-class foundry work, the two latter considerations are perhaps the most important. Modern development is on the following lines :—

(1) The provision of large doors, which serve the double purpose of providing good access for cleaning and affording better support and venting for the core when casting.
(2) Elimination of all internal ribs, as experience seems to shew that the tubular walls provided to accommodate the valve casings provide all requisite support between the top and bottom plates.
(3) The use of square instead of conical seats for the valve casings.

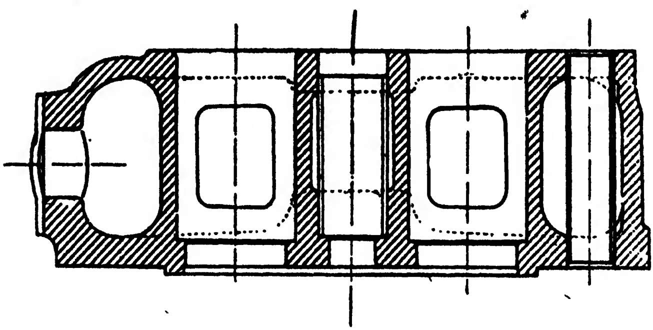


Fig. 127.

A cover designed on these lines is shewn in Fig. 127. Some makers have simplified the question of casting at the expense of introducing extra machining and fitting operations by making the top plate a separate piece (see Fig. 128).

Another innovation which is becoming increasingly common is to place the fuel valve off centre. This arrangement enables the cooling space around the fuel valve to be increased, but too great a displacement of the fuel valve from the centre position necessitates a special shape of combustion space.

Some modern developments in four stroke cylinder cover construction are shewn diagrammatically in Figs. 129 to 132.

That of Fig. 129 follows closely on traditional lines with the exception that the lower plate is dished upwards to allow

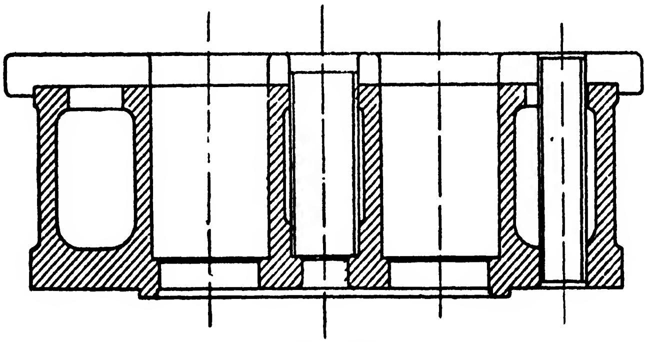


Fig. 128.

more freedom for expansion and to give more room for the valves. Fig. 130 represents a combined cover and liner as used by the Werkspoor Co. Fig. 131 shews the kind of construction adopted in large numbers of Burmeister & Wain engines. In this case the liner is secured to the cover by a

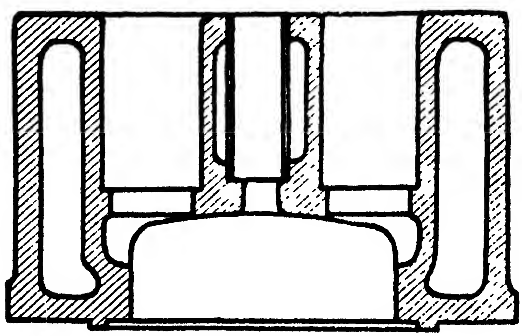


Fig. 129.

flange and setscrews. Fig. 132 shews a cover fitted with a water cooled pad to receive the heat which would otherwise reach the cover proper.

Progress in metallurgy and foundry technique during the past decade has greatly improved the quality of iron castings for the vital parts of Diesel engines.

Points of Detail.—Owing to the large recesses for the valve cages, a four cycle cover is relatively weak, considering the amount of metal in it, and on this account all stud holes should be well bossed under and all inspection openings well reinforced by compensating rings like a boiler.

The under face of the cover is machined all over, but on the

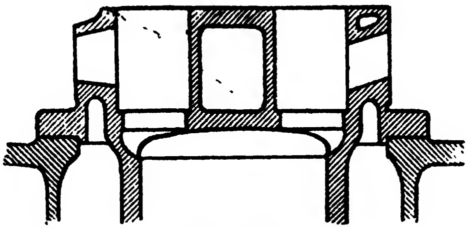


FIG. 130.

top face machining is sometimes restricted to those parts which are occupied by valves, etc. This enables the corners to be given a liberal radius, which in addition to improving the appearance facilitates moulding (see Fig. 127). From all considerations, all internal angles should be well radiused.

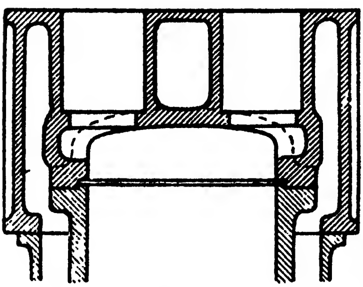


FIG. 131.

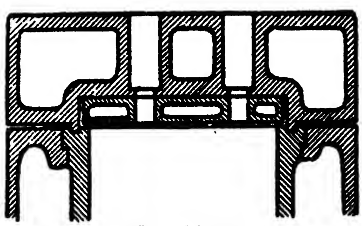


FIG. 132.

Water is led to the cover by one of two methods :—

(1) By one or more tubular fittings screwed into the top of cylinder jacket and passing through holes in the under face of the cover (see Fig. 133). With the type of jacket shewn in Fig. 120B it is desirable to fit one such fitting between each pair of cover studs.

(2) By means of an opening in the side of the cover (see Fig. 134).

Whatever means be adopted, it is advantageous to fit internal pipes or baffles to encourage flow towards the fuel valve, as accumulation of deposit at this point is to be avoided at all cost. It is usual to arrange the outlet above the exhaust branch, as stagnation at this point is also undesirable.

Proportions of Cylinder Covers.—The depth of a four stroke cover generally works out to about 0·7 of the cylinder bore, the limiting factors being the size of the exhaust passage and the water space around it. The former should be at least equal in area to the exhaust valve at full lift. The passage starts by being rectangular in shape at the valve end, and gradually becomes circular at the outlet where the diameter is

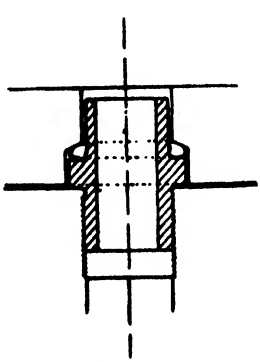


Fig. 133.

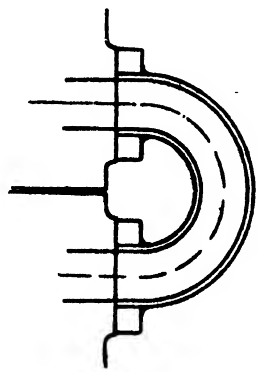


Fig. 134.

about 0·31 of the cylinder bore. The same applies to the air inlet passage. The thicknesses of metal vary considerably in different designs, and the proportions shewn on the sketches represent average practice. The bottom plate should either be fairly thick as shewn (small engines), or well supported internally round the spigot in larger sizes to prevent caving in.

Strength of Four Stroke Cylinder Covers.—The system of loads acting on the cover comprises the tightening stresses of the studs, the reaction at the spigot joint and the gas pressure on the lower plate. The effect of such a system is to produce tensile stress in the top plate and compression on the bottom. Considering the relative weakness of cast iron in tension and the fact that cracks in the top plate are of very rare occurrence, it would appear that covers proportioned in accordance with average practice have a good margin of safety so far as pressure stresses are concerned. In view of a few isolated failures, or rather as a matter of principle, the strength should be subject to calculation. Owing to the uncertainty as to actual conditions the method of calculation detailed below must be considered comparative rather than absolute.

The assumptions underlying the method are as follows :—

(1) That the severest conditions of stress are due to a cylinder pressure of 1000 lb. per sq. in., due to pre-ignition, careless starting or otherwise, and that this pressure causes the cover to lift to such an extent that the reaction at the joint is eliminated.

(2) That the stress is uniform across a diametrical section in the case of a cover of constant depth and proportional to the distance from the neutral axis of the section in the case of a cover of varying depth. This is not correct, but probably involves approximately equal percentage of error in different cases.

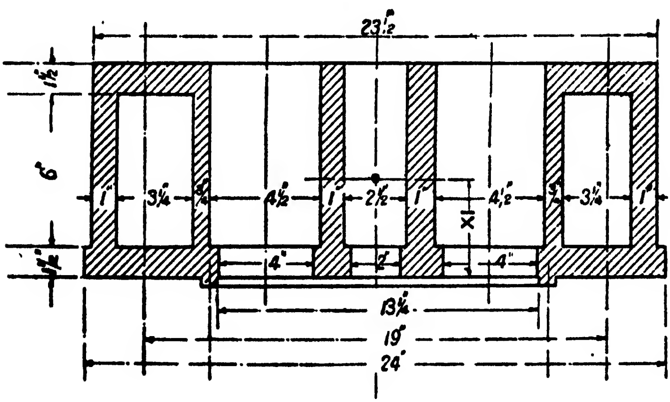


Fig. 135.

Example : Referring to Fig. 135, shewing the weakest section passing through the recesses for the air and exhaust valves, the section-modulus is 150 in.^{3}

Considering the forces to the right or left of this section, we have :—

(1) A downward force at the stud circle equivalent to a pressure of 1000 lb. per sq. in. over half the circular area, extending to the joint spigot, viz.: $\cdot 784 \times 13{\cdot}25^2 \times 1000 \div 2 = 69{,}000$ lb. This may be considered to act at

the centre of gravity of the pitch semicircle, that is at a distance of $9{\cdot}5 \times 2 \div \pi = 6{\cdot}02$ in. from the section under consideration.

(2) An equal and opposite force on the under side of the cover acting at the centre of gravity of the semicircular area extending to the joint spigot, i.e. at a distance of $\frac{6{\cdot}625 \times 4}{3\pi} = 2{\cdot}8$ in. from the centre.

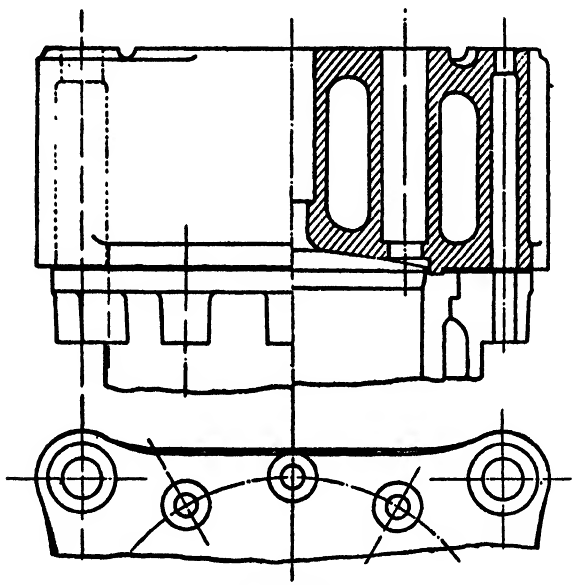


FIG. 136.

The stress is therefore :—

$$\frac{69{,}000\,(6{\cdot}02 - 2{\cdot}8)}{150} = 1500 \text{ lb. per sq. in.}$$

Putting the above into the form of a rule :—

$$f = \frac{1000\, R_1^2\, (R_2 - \frac{2}{3} R_1)}{z}$$

Where 1000 = Assumed maximum pressure.
f = Stress in lb. per sq. in.
R_2 = Radius of stud pitch circle in in.
R_1 = Inside radius of joint ring in in.
z = Section modulus in in.^{3}

Two Stroke Cylinder Covers.—Where port scavenge is adopted the cover has only to accommodate the following fittings :—

(1) Fuel valve.
(2) Starting valve.
(3) Relief valve (if fitted).
(4) Indicator tube fitting.

As all the above are relatively small the casting of the cover is much simpler than that for a four stroke engine. Fig. 136 shews a cover of this type arranged for four staybolts.

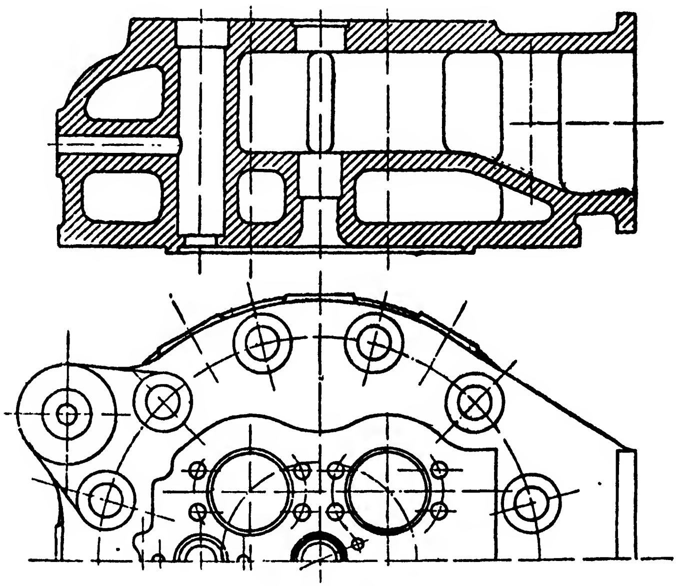


Fig. 137.

When valves in the cover are employed for scavenging purposes the construction depends on the number of valves. If two scavenge valves are used the cover may be similar to that of a four stroke engine. This arrangement seems to have fallen into disuse, no doubt on account of the difficulty of securing adequate valve area and efficient scavenging.

Fig. 137 shews a cover designed to accommodate four scavenge valves. It will be noticed that the interior is divided by a horizontal diaphragm separating the air space and the water-jacket. It appears that this diaphragm and the tubular connections to the bottom plate impose too great restrictions on the expansion of the latter, and fractures have been frequent (with both cast iron and cast steel), so that this type of cover, as hitherto designed, must be considered a failure.

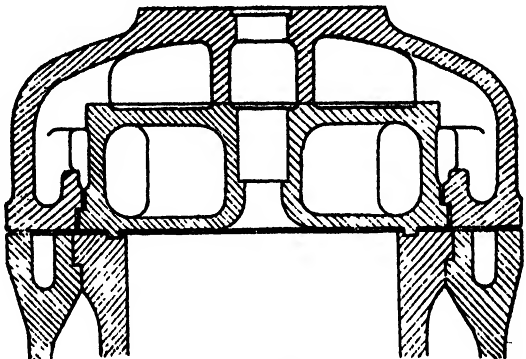


FIG. 138.

The writer understands that a modification of this design (patented by Mr. P. H. Smith), shewn in Fig. 138, has proved satisfactory, and failures have been greatly reduced in frequency. Apparently the additional depth of the water-jacket, and correspondingly increased freedom of expansion, minimise

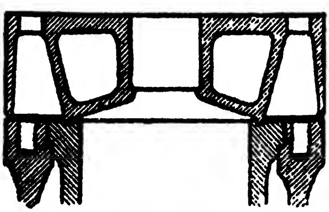


FIG. 139.

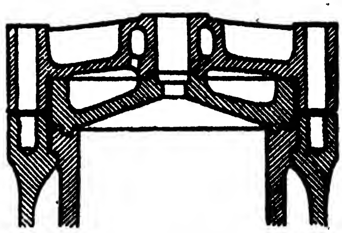


FIG. 140.

temperature stresses, and the support afforded by the external shell keeps the bending stresses to a moderate figure.

Some other types of two stroke cylinder covers are shewn diagrammatically in Figs. 139 to 141.

The first represents a Sulzer construction in which the fuel, starting and relief valves are accommodated in a central water-jacketed cage. The bottom plate is comparatively thin and ceases at the spigot. The symmetry of the casting and the flexibility of the walls are favourable conditions from the point of view of immunity from heat stresses. Fig. 140 shews a scheme used on a White two stroke engine. This appears to aim at freedom of expansion and easy replacement of the bottom plate should fracture occur. Fig. 141 shews another application of the cooled pad idea. These examples illustrate the diversity of the designs which have arisen very largely with a view to reducing the liability to failure on account of temperature stresses which are considered in the next chapter.

The Shape of the Combustion Space.—It has often been said that the ideal shape for the combustion space of an

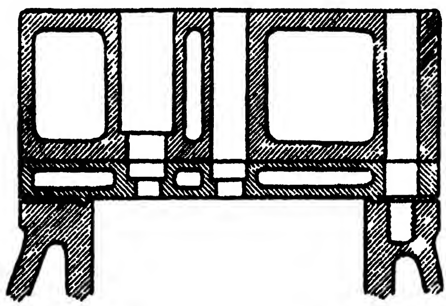


Fig. 141.

internal-combustion engine is that of a sphere, since for a given volume this shape offers the smallest surface for the dissipation of heat. Such an ideal cannot be realised in practice, as a combustion space which is spherical when the piston is on top dead centre is no longer so when the piston has moved outwards on the expansion stroke.

This consideration applies particularly to the Diesel Engine, since the maximum temperature of the Diesel cycle is only attained after a certain fraction of the expansion stroke has been performed. For this reason the spherical combustion space shewn in Fig. 142, 1, would probably shew little, if any, advantage over the flat shape of Fig. 2, so far as heat loss is concerned, though the distribution of fuel and resulting efficiency of combustion would probably be better with Fig. 1.

So far as the writer is aware, no such shape as that indicated in Fig. 1 has been used, but the approximation shewn in Fig. 3 has been adopted in several makes of small and moderate-sized engines. One advantage claimed for this shape is the turbulence caused by the expulsion of air from the annular space between the piston and the cylinder head on the compression stroke. The reverse flow on the expansion stroke is probably responsible for an increased rate of heat flow to the walls of the cover.

The first essential in a Diesel Engine is good combustion, and this can only be secured if the fuel is well distributed in the first instance. The combustion space should, therefore,

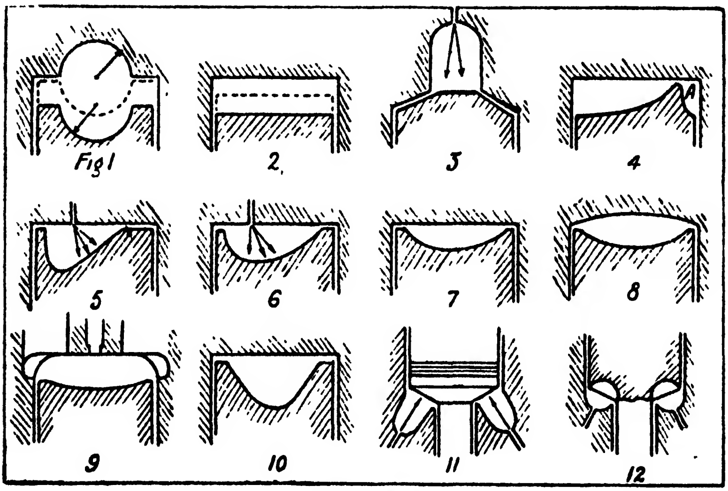


FIG. 142.

be fairly compact at the dead centre. The shape shewn in Fig. 4, for example, would render impossible good combustion at full load on account of the dead air at A to which the fuel could not find access. On the other hand, the peculiar shape shewn in Fig. 5 is capable of excellent results provided the fuel blast is properly directed as shewn. A similar shape (Fig. 6) sometimes gives inferior results if the fuel valve is offset too far from the centre. Moderate offsetting of the fuel valve seems to have little, if any, effect with large cylinders.

The earliest four stroke engines were provided with flat clearance spaces, as in Fig. 2. Better results are obtained with the plano-convex shape (Fig. 7) or the convexo-convex shape of Fig. 8. The flat shape is, however, well suited to opposed-piston engines, and perhaps to single-piston engines having a stroke to bore ratio of about 2 : 1 and upwards, since the combustion space is then fairly deep.

Sometimes the maximum diameter of the combustion space is made in excess of the cylinder bore in order to obtain more room for the valves, as in Fig. 9. In these cases care must be exercised to avoid pockets of dead air. Some designers aim at providing a considerable depth of air in the direction of the fuel blast, as in Figs. 3 and 10. The maximum temperature of the piston is probably reduced by this construction. Some forms of the pocket type of combustion space for double acting engines are shewn in Figs. 11 and 12.

The requirements of the combustion space in relation to good combustion may now be summarised as follows :—

(1) Compact shape of space at top dead centre.*

(2) Direction of fuel blast approximately through the C.G. of the combustion space ; if two fuel valves are fitted, each fuel blast should be directed at the C.G. of a symmetrically placed half space.

(3) Either (*a*) normal impingement of fuel blast (air injection), or (*b*) injection into a considerable depth of air.

Double-acting Engines.—The principle of employing both sides of a Diesel Engine piston for power purposes has been applied with varying degrees of success at intervals during the past two decades or more, but in recent years double-acting Diesels have definitely emerged from the experimental stage and taken their place in commercial fields, particularly in connection with marine propulsion.

With a few exceptions double-acting Diesel Engines have followed the development of reciprocating steam engines, inasmuch as the pistons are connected to the crossheads by single piston rods which pass through stuffing-boxes in the bottom cylinder covers. In most of these engines the top cylinder cover and its equipment of valves differ in no essential respect from those of single-acting engines and therefore require no further mention. The main problems in the design of double-acting Diesel Engines are :—

1. The design of a bottom cylinder cover combining the requisites of a good shape of combustion space, suitable accommodation of valves, and durability of the casting from the heat stress point of view. The provision of a piston rod stuffing-box may be regarded as a subsidiary problem long

* See Chapter XXVII.—Compression Space.

since solved in prior gas engines experience, but it is worthy of note that recent Diesel Engine designs shew advance in this respect on the score of simplicity.

2. Provision for expansion of the cylinder liner.

3. Provision for an adequate system of piston and piston rod cooling.

In addition, there are a number of questions in connection with the design of the running gear, the order of firing, etc., which differ somewhat from the analogous questions relating to single-acting engines.

Some of the problems enumerated above will be discussed with the help of diagrams representing examples from practice. These diagrams have been redrawn from published drawings

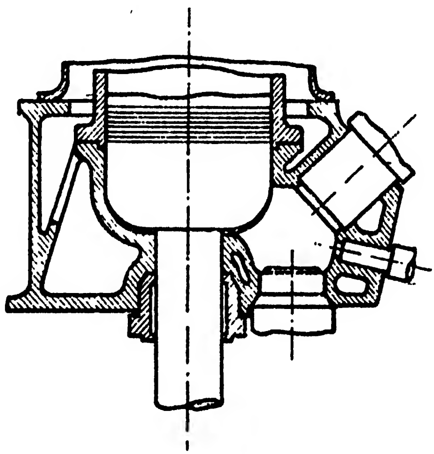


Fig. 143.

of well-known designs with a view to exemplifying a few ideas at a time and should be regarded as diagrammatic only as omissions of detail and exaggerations of thickness have been made freely wherever these appeared conducive to clearness.

The Bottom Cylinder Cover.—Various schemes for this part are illustrated in Figs. 143 to 148. The first three refer to four stroke engines, and in each case the combustion space takes the form of a single compact chamber provided with a single fuel valve. The latter three figures refer to two stroke engines and in each case the combustion space takes a more or less annular form and is provided with two fuel valves. In these two stroke engines the bottom ends of the pistons seem to have been designed primarily with regard to their functions as deflectors of the scavenge and exhaust streams, and pockets

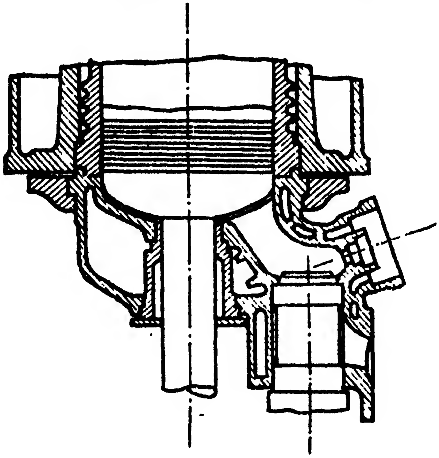


FIG. 144.

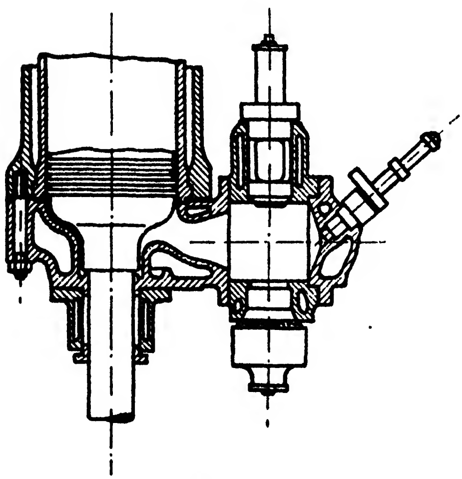


FIG. 145.

in the cover have been avoided. The resulting annular form of combustion space can hardly be adequately penetrated by sprays directed from one point, hence the duplication of fuel valves.

Returning to Fig. 143, which represents the type of bottom cover used by Burmeister & Wain, Harland & Wolff, and other B. & W. licencees, it will be noticed that the cover is a single piece casting of cuboidal shape externally and well

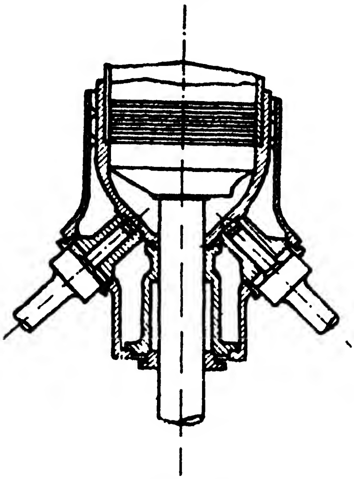


FIG. 146.

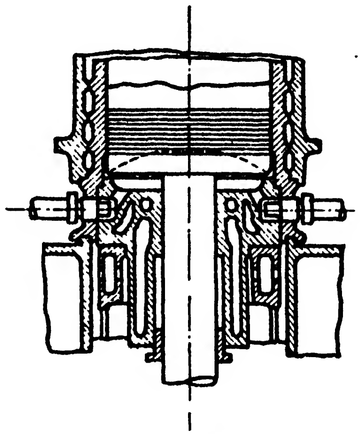


FIG. 147.

rounded internally to accommodate expansions. The liner is secured to the cover by a ring of set screws. The inlet, exhaust, fuel and starting valves are disposed approximately radially around the nearly spherical combustion space.

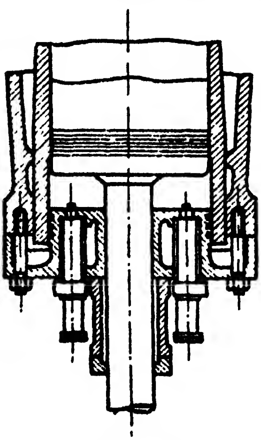


FIG. 148.

Fig. 144 represents the Beardmore-Tosi design, in which the cover is held against the liner by a clamping ring. A single valve is enabled to function as inlet and exhaust alternately by virtue of a deflector valve (not shewn) which puts the valve into communication with the inlet and exhaust manifolds respectively as the occasion demands. Fig. 145 represents the North-Eastern Werkspoor design, consisting of three main parts, viz.: the cover proper, the combustion chamber accommodating the inlet and exhaust valves and the combustion chamber door accommodating the fuel valve. In this design the compression ratio is lower than usual at the bottom of the cylinder with a correspondingly larger compression space.

Turning now to the two stroke designs, Fig. 146 represents a

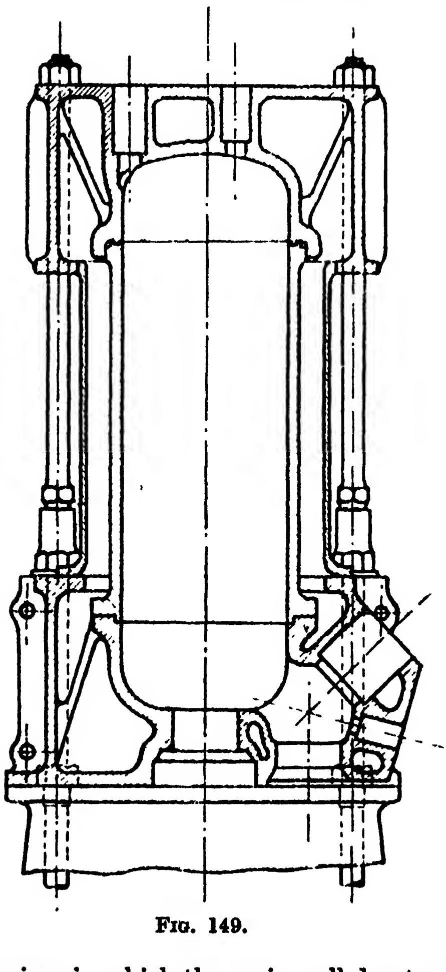


FIG. 149.

Worthington design, in which the main pull due to gas pressure is transmitted by an alloy steel shell of projectile shape, the parallel part being lined with a cast iron liner to the extent of the piston ring travel and the conical part forming the envelope of the combustion space. Twin fuel valves inclined

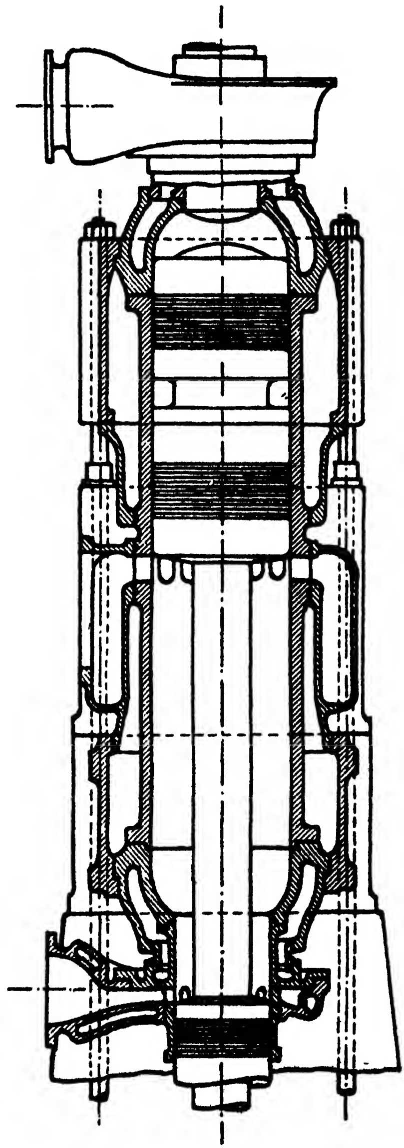


FIG. 150.

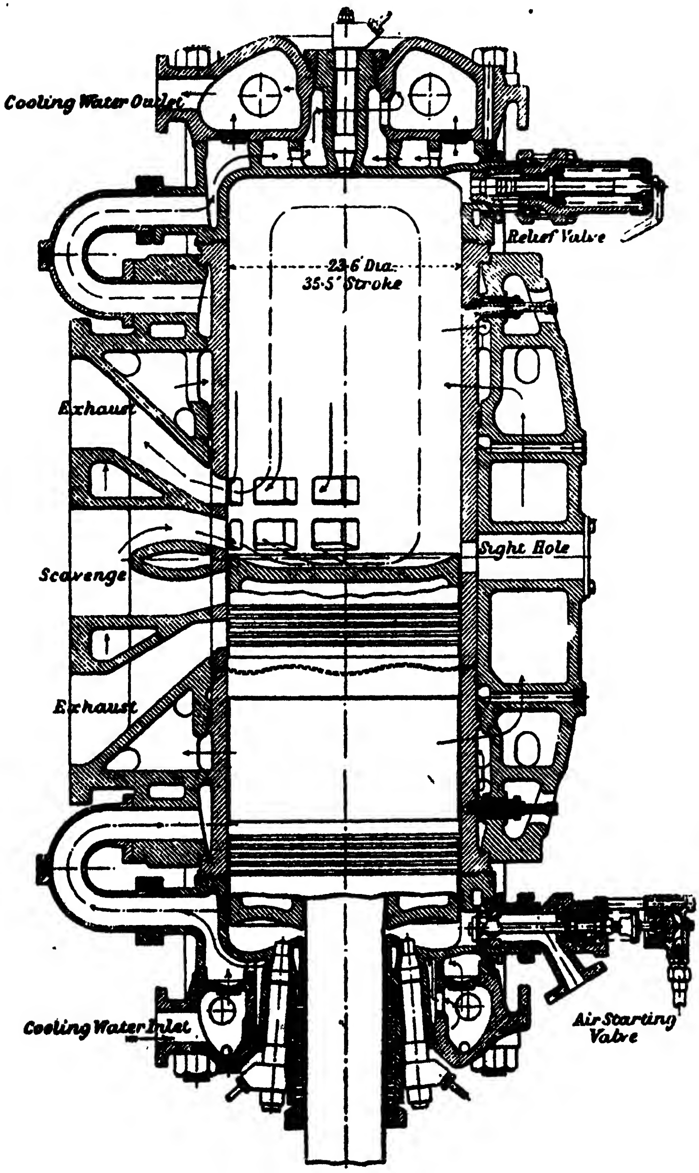

Cooling Water Outlet
Relief Valve
23·6" Dia.
35·5" Stroke
Exhaust
Sight Hole
Scavenge
Exhaust
Cooling Water Inlet
Air Starting Valve

Fig. 151.

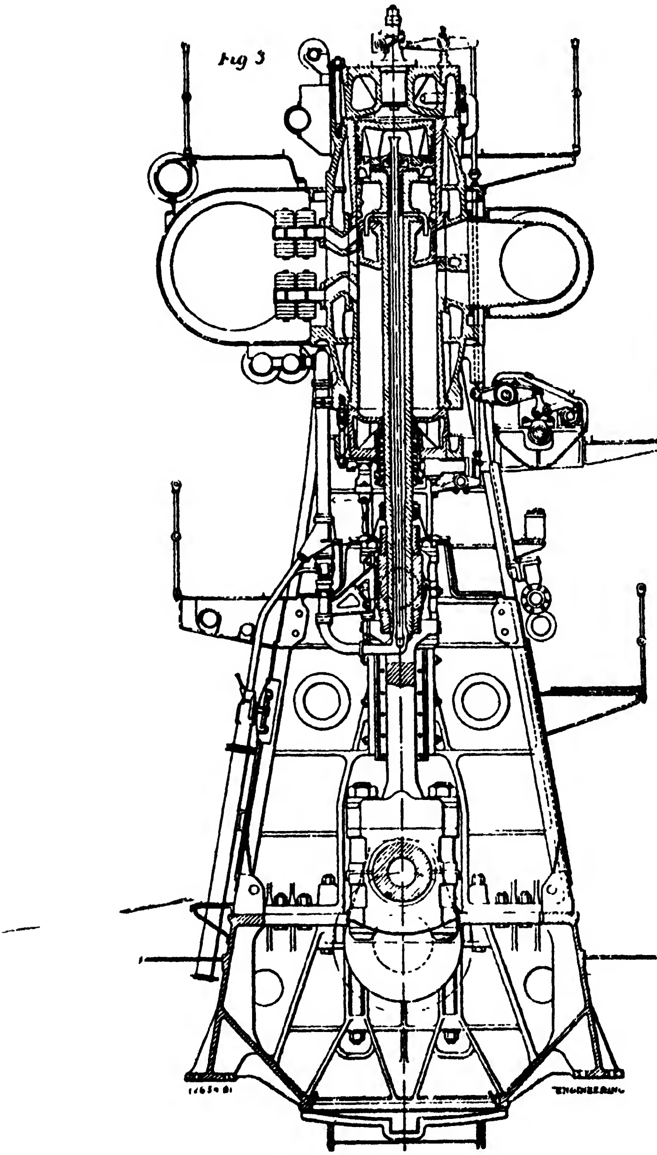


Fig. 152.

at about 45° are employed. Fig. 147 represents an M.A.N. design, in which the cylinder cover takes the form of a sort of water-cooled breech block held in position by an annular member provided with projections which abut against corresponding projections in a surrounding entablature. By twisting the annular member through an angle the projections are

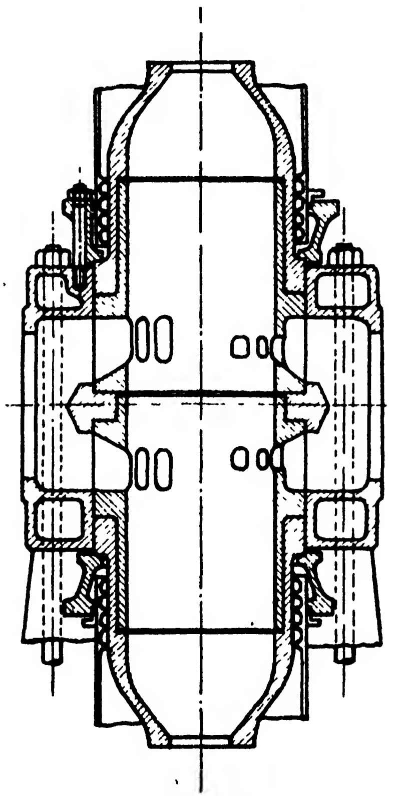


Fig. 153.

disengaged and the breech block can be lowered. Fig. 148 represents a Richardson-Westgarth design. The cover is secured by a ring of studs to the water-jacket which takes the gas pull. Other designs of bottom cylinder cover are shewn on the diagrams illustrating the next section.

Expansion of the Cylinder Liner.—Inadequate provision for the expansion of the cylinder liner has been responsible for the failure of many gas engines and some oil engines. Profiting by this experience designers of double-acting Diesel Engines have paid particular attention to this point. In the Burmeister & Wain four stroke engine (Fig. 149) the cylinder liner is sandwiched between the top and bottom covers which are drawn together by long through bolts extending from the top of the upper cover to the bottom of the bedplate. The intermediate nuts are so adjusted that the whole length of the bolts is available for stretching on account of the expansion of the liner. The latter is so small in relation to the length of the bolts that the increase in the tension of these bolts is quite small.

The same principle is used in the B. & W. D.A. two cycle engine cylinder shewn in Fig. 150.

In the M.A.N. and Sulzer designs shewn in Figs. 151 and 152 the water jacket forms the principal supporting member into which cylinder liners are inserted from above and beneath and secured by spigoted cylinder covers in the conventional way. The two half liners are separated at the centre by a narrow gap to allow for expansion.

In the Worthington design shewn in Fig. 153 the opposite arrangement is adopted. The liners are anchored at their inner extremities being pressed by studded ring members against a central abutment in a single entablature which accommodates the exhaust and scavenge in passages.

CHAPTER XV

TEMPERATURE STRESSES

Introductory.—It is fairly generally recognised that the stresses in the cylinders of internal combustion engines due to temperature differences are, at least, as important as those due to pressure, at any rate in the larger sizes of engine, and that the magnitude of these stresses sets an upper limit to the size of cylinder which is possible with any given method of construction.

Such considerations apply with even greater force to cylinder covers and pistons, but it is not necessary to agree with those who maintain that the maximum size of cylinder possible or the maximum power which can safely be developed in a cylinder of given bore and stroke is necessarily limited by the materials available.

It is true that every substantial advance in size and power rating accentuates the heat stress problem, but there is no reason to suppose that such problems cannot be solved satisfactorily. Some of the newer constructions which have been developed to overcome temperature difficulties have already been noticed, and it can hardly be doubted that demand will be met by supply.

In the present chapter the general heat flow problem as it affects Diesel Engine Design will be considered in detail with a view to throwing some light on the principles involved.

Temperature difficulties in Diesel Engines may be classified in three categories :—

(1) Local overheating, as for example of exhaust valve heads, piston crowns, rings and liner surfaces.

(2) Local temperature differences, as for example the temperature difference between the gas and water sides of a liner or cylinder cover plate.

(3) Extensive temperature differences, as for example the temperature difference between the centre and the edge of an uncooled piston crown.

The troubles coming under the first category are avoided or minimised by either avoiding uncooled surfaces altogether or by making the uncooled part of a special material capable of performing its function at a high temperature.

The stresses which arise on account of the second category can only be kept within safe limits by either limiting the heat flux to which the surface is subject or else limiting the thickness of metal through which the flux penetrates.

Extensive temperature differences (3) are, in general, only productive of temperature stresses in so far as the temperature differences refer to different parts of the same casting. In these cases the stresses are to be kept within safe limits by adopting forms which admit of free expansion. A study of the problems which arise out of these considerations must clearly start with an estimate of the distribution of heat flux to the various members which constitute the walls of the cylinder volume.

Heat Flow to the Jackets.—The heat received by the cylinder walls may be regarded as transferred thereto by convection and radiation. Any element of pure conduction may presumably be neglected except in so far as the process of convection culminates in the conduction of the convected heat through a thin layer of stagnant gas in contact with the walls. The relative importance of the two processes of radiation and convection in transferring the heat lost to the jacket is not known with any certainty, but a number of important facts have emerged as a result of the investigations, some of which are referred to at the end of this chapter; among these may be mentioned the following:—

(1) The heat lost by convection is increased by increasing the state of turbulence before and during ignition (Hopkinson and Clerk). This fact helps to account for the observed fact that increasing the piston speed beyond very moderate values does not very materially reduce the percentage of heat rejected to the jacket. Apparently the increased air speed through the inlet valve results in increased turbulence, which tends to speed up the loss of heat.

(2) A mass of flaming gas is semi-opaque, semi-transparent to its own radiations. That is to say, of the total radiation emanating from a small element of flame, a certain part (usually a small fraction in cylinder volumes of ordinary size)

is absorbed in passing through the outer layers (Callendar). The practical bearing of this fact is open to different interpretations in the present state of knowledge. According to one school of thought the semi-transparency of flame is fatal to large cylinders on account of the augmented radiation flux which results from increasing the depth of the flame. On the other hand, it may be argued that the partial opacity of flaming gas must lead to the rejection by radiation of smaller percentages of the heat liberated by combustion, in cases of large cylinders as compared with small ones. Both arguments are unsound, since both ignore the duration of the combustion period and the question whether the total radiation is simply a function of temperature and time, or whether it depends primarily on the amount of heat evolved by combustion. Fortunately the heat flow question does not depend on the solution of these problems. The total heat received in all ways by the various elements of the cylinder walls can be measured approximately by various more or less direct means. These measurements indicate that the jacket loss expressed as a percentage of the total heat supplied is nearly independent of the absolute size.

(3) The intensity of radiation is proportional to some power (3 to 4) of the absolute temperature. This fact helps to account for the fact that indicated efficiencies tend to fall off after a certain critical maximum temperature is exceeded. In what follows we shall only be concerned with the Diesel Cycle (four stroke or two stroke) employing a normal full load M.I.P. of about 85 to 105 lb. per inch (unsupercharged).

The water-jacketed parts of a Diesel Engine comprise some or all of the following:—

(1) Cylinder liner.
(2) Cylinder cover.
(3) Exhaust valves and cages.
(4) Piston (frequently oil cooled).
(5) Air compressor.
(6) Intercoolers.
(7) Exhaust manifold.
(8) Crosshead guides.
(9) Oil coolers.

Here we are only concerned with items (1) to (4) inclusive, which, on the average, may be taken to account for about 25% of the total heat supplied, reckoned on the lower calorific value of the fuel consumed, at about full load. This will be referred to below as the "total jacket heat."

Whilst water-cooled valves and cages absorb quite an appreciable percentage of the jacket heat, it is doubtful if the cooling of these parts materially affects the total jacket heat, since the exhaust gases are cooled thereby, and are consequently in a condition to give less heat to the exhaust port in the cylinder head or cover. The same argument applies to the piston. Heat received by an uncooled piston is mainly conducted to the cylinder liner, which therefore receives more heat than it would otherwise if the piston were cooled. Before considering the distribution of the jacket heat it may be remarked that the latter under approximately full load conditions, when expressed as a percentage of the total heat supplied, shews remarkably little variation with such variables as piston speed, mean indicated pressure, stroke bore ratio, absolute size, etc., within the limits of ordinary Diesel Engine practice.

Cylinder Cover.—The heat received by the cover of a four stroke engine consists of two parts:—

(a) That which passes through the bottom plate.
(b) That which passes through the exhaust valve port.

For simplicity we suppose that the cover is of the traditional flat-bottomed type. Gibson and Walker have attempted to measure "b" in the case of a gas engine, and found an exhaust valve and port loss of 8 to 11% of the total heat supplied, or rather less than one-third of the "total jacket loss" as here understood. From the conditions of the experiment these figures must be over-estimated. Dugald Clerk arrived at a similar estimate based on Burstall's gas engine trials, but this estimate is vitiated by the assumption that the heat passing through the cylinder cover face was equal to that received by the piston, an assumption which Hopkinson has shewn to be very improbable in a gas engine. Hopkinson's estimate of the exhaust port heat is about 6% of the total heat supplied in the case of a gas engine. For Diesel Engines the writer is inclined to adopt the figure of 4½% of the total heat supplied or about 18% of the total jacket loss, for reasons which appear below.

In the first place items "a" and "b" together only account for about 35% of the jacket loss in the case of a four stroke Diesel Engine having a stroke bore ratio of about 1·5. Furthermore, recent temperature gradient measurements in the cover plate of a four stroke Diesel Engine having a stroke bore ratio of 1, indicate that the bottom plate in this case receives about 28% of the total jacket heat. If the stroke bore ratio were increased to 1·5 the cover plate loss expressed as a percentage of the jacket loss would certainly be reduced and would probably be about 17% (see Fig. 155), leaving 35—17=18% for the port loss. As a first approximation we therefore adopt the following figures :—

PERCENTAGE OF TOTAL JACKET HEAT

Stroke bore ratio	1·0	1·5
Cover plate	28	17
Exhaust port	18	18
Total for cover	46	35

It will be noticed that it is here assumed that the exhaust port loss is independent of the stroke bore ratio.

In the case of two stroke engines where there is no exhaust port in the cover we are only concerned with the bottom plate.

Pistons.—Even with uncooled pistons the maximum crown temperature is low compared with that of the gases during the combustion and exhaust periods, and it has been shewn that the heat taken up from the piston by the charge is small. The net heat received by the piston crown is not therefore greatly influenced by the cooling means adopted. Hopkinson has shewn that the heat dissipated from the underside of the crown of an uncooled gas engine piston has very little effect on the temperature of the latter. He has also shewn how to calculate the amount of heat received by the piston from two temperature measurements at two different distances from the centre of the crown. On these assumptions the piston of his 11½″×21″ gas engine received about 12% of the total jacket heat. In Burstall's trials of a 16″×24″ gas engine the water-cooled piston received about 16½% of the total jacket heat.

With the four stroke Diesel Engine with " square " cylinder already referred to, the measured piston heat amounted to 12% of the total jacket heat, but in this case it is probable that, owing to the relatively mild character of the oil cooling, a certain amount of heat received by the piston was conducted to the liner. The water-cooled piston of a 15·4″×15·75″ two stroke Diesel Engine absorbed about 22% of the total jacket heat.

The water-cooled piston of a 16·1″×19·9″ two stroke Diesel Engine received on the average about 36% of the total jacket heat. This abnormally high figure is probably due to :—

(1) The use of a convex piston crown.
(2) The liberal extent of the piston jacket which extended half-way down the skirt.
(3) About a quarter of the total quantity of cooling water was passed through the pistons.

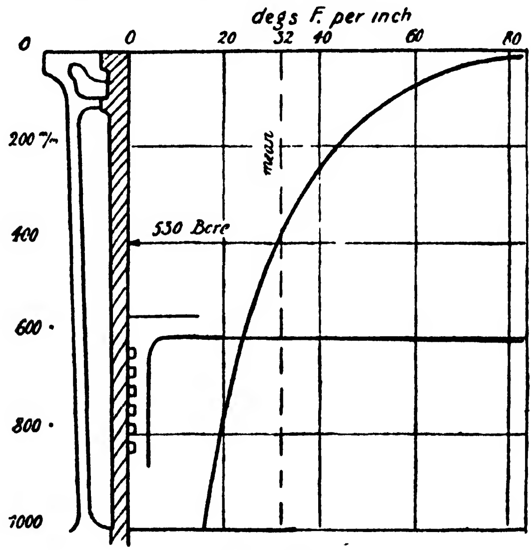


Fig. 154.

In this case the piston was probably extracting heat from the liner.

In a modern two stroke engine with concave piston crown and stroke bore ratio of 1·6, the cylinder jacket received about 10% and the piston and cylinder cover about 6% each of the total heat supplied.

The percentage of jacket heat absorbed by the pistons of two stroke engines may reasonably be expected to exceed that of four stroke engines on account of the exhaust taking place at the bottom of the cylinder with great velocity.

Comparing these figures and confining attention to the heat actually passing through the piston crown it would appear that in four stroke engines the piston heat is rather less than the cover plate heat, and that in two stroke engines those quantities are about equal, assuming concave piston crowns in both cases.

Cylinder Liners.—In four stroke engines with stroke bore ratios of about 1·5 with uncooled pistons about 65% of the

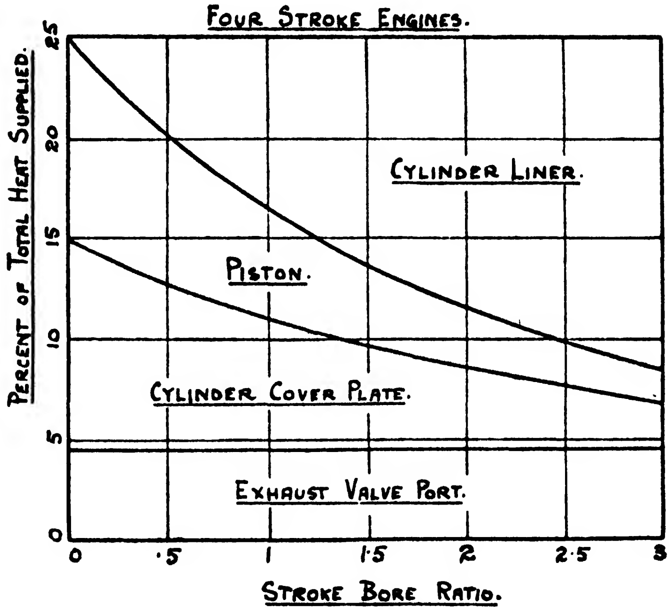


FIG. 155.

jacket heat is found in the cylinder jacket. Deducting 16% for the piston this leaves 49% for the liner heat proper. The thermal data given by Chaloner (*Motor-ship*, July, 1920) in connection with a four stroke cylinder, 20·85″×20·85″, enable the jacket loss to be estimated. The temperature gradients are shewn in Fig. 154, from which it appears that the mean temperature gradient is about 32° F. per inch, and the mean flux is therefore :—

$$32 \times 12 \times 26 = 10{,}000 \text{ B.T.U. per ft.}^2/\text{hr.}$$

Since the conductivity of cast iron is about 26 in ft. deg. F.

hour units. The total area of the liner is about 19·2 ft.^{2}, so the total heat flow is :—

$$10{,}000 \times 19{\cdot}2 = 192{,}000 \text{ B.T.U. per hour.}$$

The power of the cylinder is 285 B.H.P., and reckoning on a fuel consumption of 0·42 lb. per B.H.P. hr. the total heat supplied works out at :—

$$0{\cdot}42 \times 18{,}000 \times 285 = 2{,}150{,}000 \text{ B.T.U./hr.}$$

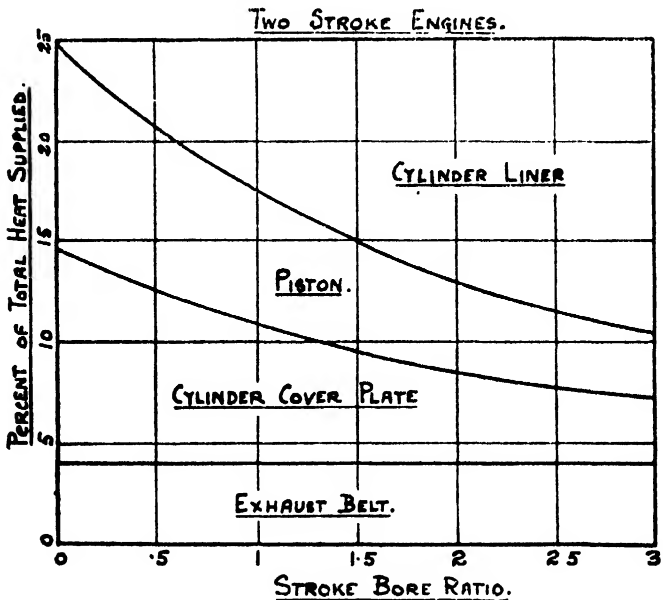


Fig. 156.

The liner heat is therefore 8·9% of the total heat supplied, or about 35% of the total jacket heat.

Heat Flow Diagrams.—On the basis of the foregoing figures and the consideration of limiting cases, Figs. 155 and 156 have been drawn up for four and two stroke engines respectively.[1] In each diagram the percentage of heat passing to the liner, piston, etc., is plotted on a base of stroke bore ratio. A few notes of explanation are desirable.

(1) In each case the total jacket heat is assumed to be 25% of the total heat supplied.

[1] Fig. 155 would appear to be applicable to 2 cycle engines with exhaust valves in the cover.

(2) The exhaust port loss is regarded as being independent of the stroke bore ratio.
(3) The liner loss approaches zero as the stroke bore ratio approaches zero.
(4) The cover plate loss and the piston loss are assumed to approach equality as the stroke bore ratio approaches zero.
(5) The cover plate loss and the piston loss tend to 0 as the stroke bore ratio tends to ∞.
(6) For finite values of the stroke bore ratio the trends of the curves have been guided by the data given above, combined with a "fairing" up process.

No great accuracy can be claimed for these diagrams, but they have been checked from time to time against published data relating mainly to marine engines, and the order of accuracy for individual items is probably round about $\pm 20\%$.

The Heat Flux through the Walls.—The foregoing data enable the heat flux through the walls of the liner, cover plate and piston to be calculated approximately in any given case. The mean flux through the liner is found by dividing the liner heat in B.T.U. per hour by the area of the water-cooled surface in sq. ft. In the cylinder, for which Fig. 154 has been drawn, the cooled length of the liner is equal to the stroke plus about 80% of the bore, and the maximum flux is 2·5 times the mean. In other cases the proportion of cooled length may be different, but this is hardly likely to have any appreciable effect on the maximum flux. Accordingly we shall assume that in all cases the maximum flux is 2·5 times the mean flux when the latter is calculated on an area equal to the product of the cylinder circumference $\times$ the stroke plus 0·8 times the bore.

Let I = Indicated power per sq. ft. of piston area.
B = Bore in feet.
S = Stroke in feet.

Consumption of fuel (18,000 B.T.U. per lb.) = 0·3 lb. per I.H.P. hr.

l = Fraction of total heat supplied which passes through the liner (see Figs. 155 and 156).

F = Maximum flux in 1000 B.T.U./ft.^{2} hr.

Then :—

$$F = \frac{\dfrac{I\pi B^2}{4} \times \dfrac{0{\cdot}3 \times 18{,}000}{1000} \times 2{\cdot}5l}{\pi B(S + 0{\cdot}8\,B)} = \frac{3{\cdot}4Il}{\left(\dfrac{S}{B} + 0{\cdot}8\right)} \quad \ldots\ldots\ldots\ldots (1)$$

hour units. The total area of the liner is about 19·2 ft.^{2}, so the total heat flow is :—

$$10{,}000 \times 19{\cdot}2 = 192{,}000 \text{ B.T.U. per hour.}$$

The power of the cylinder is 285 B.H.P., and reckoning on a fuel consumption of 0·42 lb. per B.H.P. hr. the total heat supplied works out at :—

$$0{\cdot}42 \times 18{,}000 \times 285 = 2{,}150{,}000 \text{ B.T.U./hr.}$$

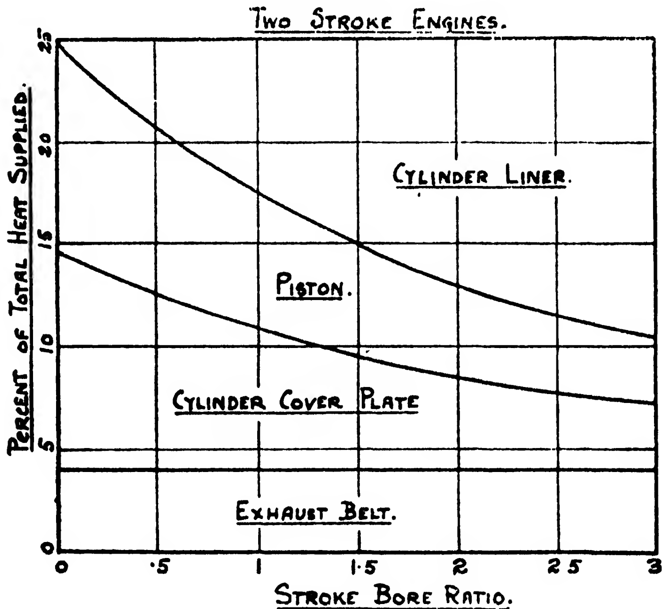


Fig. 156.

The liner heat is therefore 8·9% of the total heat supplied, or about 35% of the total jacket heat.

Heat Flow Diagrams.—On the basis of the foregoing figures and the consideration of limiting cases, Figs. 155 and 156 have been drawn up for four and two stroke engines respectively.[1] In each diagram the percentage of heat passing to the liner, piston, etc., is plotted on a base of stroke bore ratio. A few notes of explanation are desirable.

(1) In each case the total jacket heat is assumed to be 25% of the total heat supplied.

[1] Fig. 155 would appear to be applicable to 2 cycle engines with exhaust valves in the cover.

(2) The exhaust port loss is regarded as being independent of the stroke bore ratio.

(3) The liner loss approaches zero as the stroke bore ratio approaches zero.

(4) The cover plate loss and the piston loss are assumed to approach equality as the stroke bore ratio approaches zero.

(5) The cover plate loss and the piston loss tend to 0 as the stroke bore ratio tends to ∞.

(6) For finite values of the stroke bore ratio the trends of the curves have been guided by the data given above, combined with a " fairing " up process.

No great accuracy can be claimed for these diagrams, but they have been checked from time to time against published data relating mainly to marine engines, and the order of accuracy for individual items is probably round about $\pm 20\%$.

The Heat Flux through the Walls.—The foregoing data enable the heat flux through the walls of the liner, cover plate and piston to be calculated approximately in any given case. The mean flux through the liner is found by dividing the liner heat in B.T.U. per hour by the area of the water-cooled surface in sq. ft. In the cylinder, for which Fig. 154 has been drawn, the cooled length of the liner is equal to the stroke plus about 80% of the bore, and the maximum flux is 2·5 times the mean. In other cases the proportion of cooled length may be different, but this is hardly likely to have any appreciable effect on the maximum flux. Accordingly we shall assume that in all cases the maximum flux is 2·5 times the mean flux when the latter is calculated on an area equal to the product of the cylinder circumference $\times$ the stroke plus 0·8 times the bore.

Let I = Indicated power per sq. ft. of piston area.

B = Bore in feet.

S = Stroke in feet.

Consumption of fuel (18,000 B.T.U. per lb.) = 0·3 lb. per I.H.P. hr.

l = Fraction of total heat supplied which passes through the liner (see Figs. 155 and 156).

F = Maximum flux in 1000 B.T.U./ft.^{2} hr.

Then :—

$$F = \frac{\dfrac{I\pi B^2}{4} \times \dfrac{0{\cdot}3 \times 18{,}000}{1000} \times 2{\cdot}5l}{\pi B(S + 0{\cdot}8\,B)} = \frac{3{\cdot}4Il}{\left(\dfrac{S}{B} + 0{\cdot}8\right)} \qquad (1)$$

The heat received by the cylinder cover plate cannot be quite equally distributed, but must be somewhat more intense at the centre; thermocouple readings given by Chaloner shewed a difference of only about 4% between the flux at the centre of the cover, and that at a point situated at a distance of about three-quarters of the cylinder radius from the centre. Also some of the heat received by the underside of the plate is conducted away radially beyond the cylinder radius before reaching the water-jacket. The heat received by the plate divided by the area of the latter must, therefore, be an over-estimate of the mean flux, though it is probably a good approximation to the maximum flux. Similar considerations apply to the piston. We therefore estimate the maximum flux to the cover plate and to the piston respectively by dividing the heat received in each case by the piston area.

Let c = Fraction of total heat supplied which passes through the cover plate.
p = Fraction of total heat supplied which passes through the piston crown.

(See Figs. 155 and 156.)

Then :—

$$F = I \times \frac{0{\cdot}3 \times 18{,}000}{1000} c \text{ (or } p) = 5{\cdot}4\, I\, c \text{ (or } p) \qquad \ldots\ldots\ldots\ldots (2)$$

The values of c (or p) decrease as the stroke bore ratio increases. Examples :—

	I	$\frac{S}{B}$	$\frac{3{\cdot}4}{\frac{S}{B}+0{\cdot}8}$		p	c	F Liner.	F Piston.	F Cover.
4 Stroke Submarine Engine	170	1·0	1·88	·085	·055	·065	27	51	60
4 ,, Marine ,,	75	1·5	1·48	·115	·040	·050	13	16	20
4 ,, Marine ,,	85	2·0	1·21	·135	·030	·040	12	12	18
2 ,, Naval ,,	240	1·2	1·70	·085	·060	·060	35	78	78
2 ,, Marine ,,	140	1·5	1·48	·100	·055	·055	21	41	41
2 ,, Stationary ,,	160	2·0	1·21	·120	·045	·045	23	39	39

These figures give an approximate guide to the values of the heat flux to be reckoned with in typical cases.

Local Temperature Stresses.—A knowledge of the heat flux enables the local temperature difference, and consequently the local temperature stress, to be estimated.

Let t = Thickness of wall in inches.

(T_1-T_2) = Temperature drop through the wall in deg. F.

k = Conductivity of cast iron in ft. B.T.U. hr. units = about 25.

E = Modulus of elasticity of cast iron = about 12×10^6 lb./in.^{2}

λ = Coefficient of expansion of cast iron $=6\times10^{-6}$.

Then the maximum local difference of temperature from the mean $=\frac{1}{2}(T_1-T_2)$

$$=\frac{1000\ F}{2k}\times\frac{t}{12}=\frac{500\ Ft}{25\times12}=1{\cdot}67\ Ft$$

And the temperature stress f_t is given by :—

$$f_t=\tfrac{1}{2}(T_1-T_2).E.\lambda=1{\cdot}67\ Ft\times10^6\times12\times10^{-6}\times6=120\ Ft\ldots\ldots(3)$$

The important point is that for a given value of the heat flux F, the temperature stress is proportional to the wall thickness. Consider, for instance, the two stroke Naval engine referred to in the above table. For the cover and piston $F=78$, and the local temperature stress $=120\times78=9360$ lb./in.^{2} per inch of thickness. It is evident, therefore, that with ordinary materials the available range of safe thicknesses is limited, whatever the bore of the cylinder may be. The problem of the large highly rated cylinder is, therefore, reduced to finding methods of construction whereby comparatively thin walls may be used with safety. The question as to what temperature stress can be considered safe is naturally an important one and a very difficult one to decide. Most heat failures of covers and pistons are traceable to extensive temperature differences giving rise to stresses which are very difficult to estimate. Cylinder liners and plain ribless covers and pistons afford better criteria, and calculations of known designs on the lines indicated above indicate that the safe limit of temperature stress lies somewhere between 10,000 and 15,000 lb. per sq. inch. The ultimate tensile strength of good cylinder iron is about 40,000 lb. per sq. inch and a safe limit of $\frac{1}{3}$ this value, viz. 13,000 lb./in.^{2}, may be suggested.

The whole business is greatly complicated by the peculiar behaviour of cast iron at high temperatures. According to the theory outlined above, the hot side of the wall should be in a state of compression and the cold side in tension. Actually cracking, when it occurs, almost always takes place on the hot side, leaving when cold a gaping fissure which tends to close up during running of the engine. From this fact it appears that the temperature stresses are annealed out during prolonged running by rearrangement of the particles, and that cracking takes place on cooling by the reappearance of the temperature stresses with reversed sign. Nevertheless, it is to be expected (apart from the phenomenon of "growth") that the stresses induced during cooling will be proportional, if not actually equal, to those induced by the initial heating before annealing has taken place.

Local Temperature Stresses in Pistons and Covers.—It is interesting to see what results are yielded by formula (3) in representative cases.

(1) Four stroke engine, $12'' \times 18''$, $I=75$, $t=1\frac{1}{8}''$ for cover.
$f_t=120 \times 20 \times 1{\cdot}125=2700$ lb./in.^{2}
Local temperature stress obviously negligible.

(2) Four stroke engine, $30'' \times 45''$, $I=85$, $t=2''$ for cover.
$$f_t=120 \times 20 \times \frac{85}{75} \times 2=5500 \text{ lb./in.}^2$$

(3) Two stroke engine, $30'' \times 45''$, $I=140$, $t=1\frac{1}{2}''$ for cover.
$f_t=120 \times 41 \times 1{\cdot}5=7400$ lb./in.^{2}

(4) Two stroke engine, $20'' \times 24''$, $I=240$, $t=0{\cdot}875$ for cover.
$f_t=120 \times 78 \times 0{\cdot}875=8160$ lb./in.^{2}

(5) Four stroke engine, $30'' \times 45''$, $I=85$, $t=1\frac{3}{4}''$ for piston.
$$f_t=120 \times 16 \times \frac{85}{75} \times 1{\cdot}75=3800 \text{ lb./in.}^2$$

(6) Two stroke engine, $30'' \times 45''$, $I=140$, $t=1\frac{3}{4}''$ for piston.
$f_t=120 \times 41 \times 1{\cdot}75=8600$ lb./in.^{2}

In the above all pressure stresses and extensive temperature stresses are left out of consideration. These figures shew that if undue thicknesses of metal are avoided the local temperature stresses can be kept below a moderate limit.

Local Temperature Stresses in the Liner.—In the case of the liner the additional stress due to pressure is easily allowed for. Assuming a maximum working pressure of 500 lb./in.^{2} the pressure stress f_p is given by :—

$$f_p=250 \text{ B/t} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots (4)$$

Where B=cylinder bore and t=liner thickness in inches. The combined pressure and temperature stress f_c is the sum of f_p as given above and f_t as given by (3). For example, in a four stroke engine of 24″ bore × 36″ stroke with I=75 we have (since F=13, see table p. 300) :—

$$f_c = (250 \times 24/t) + (120 \times 13 \times t)$$
$$= 6000/t + 1560\ t \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (5)$$

Values of f_p, f_t and f_c are plotted on a basis of "t" in Fig. 157, and f_c has a minimum value of 6100 lb./in.² when t=2″, and

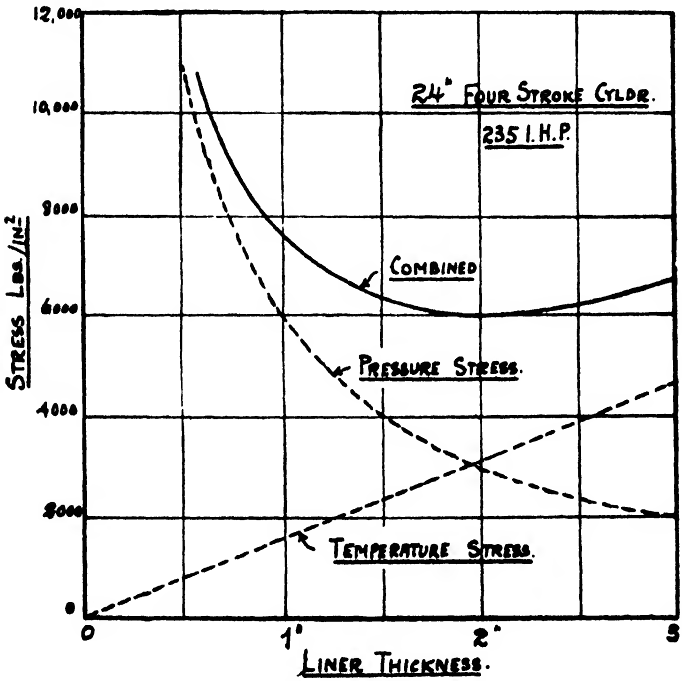


Fig. 157.

this value occurs when $f_t = f_p$. A similar diagram is shewn in Fig. 158 for a two stroke engine, 17·5″ bore, 26″ stroke with I=140 developing the same I.H.P. as the four stroke engine. In this case the minimum value of f_c is higher, viz. 6600 lb./in.², and the optimum thickness is about 1⅜″.

Both these values of the thickness are in good agreement with average practice with reference to the thickness under the top flange of the liner.

The following features of these diagrams may be noticed.

(a) There is a certain thickness of the liner for which the combined stress is a minimum.

(b) The minimum stress is slightly higher for the two stroke cylinder than for the four stroke cylinder of the same I.H.P. at the specified rating.

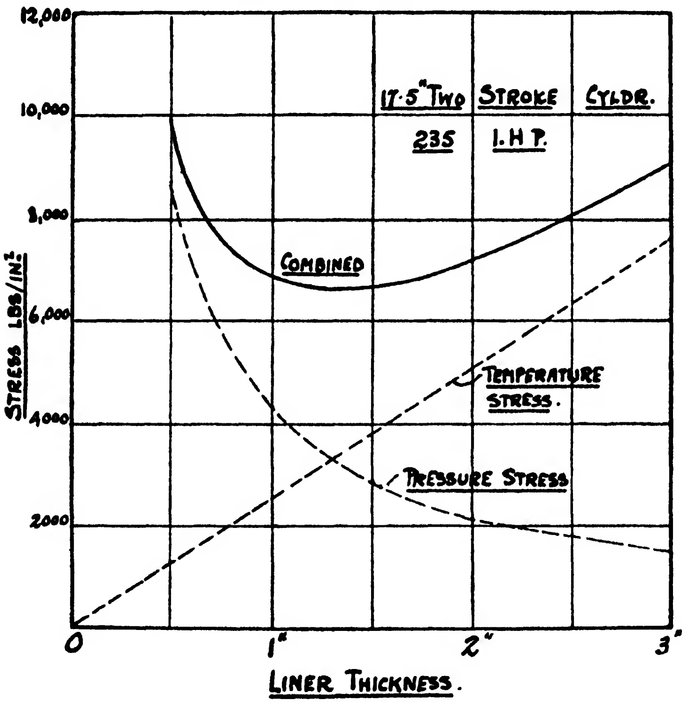


FIG. 158.

(c) The minimum stress increases as the cylinder size is increased.

(d) Considerable variations from the most favourable liner thickness are possible with small influence on the combined stress.

The last consideration is interesting in view of the thickening of the liner at the flange end which occurs in most designs. An increase of about 30% increases the stress according to the diagram by about 4 or 5%. If, as in some designs, the thickness at the flanged end is about double the normal thickness, the increase of stress may be very considerable in highly rated engines. This point is illustrated by Fig. 159, which has been drawn up for a 30″×45″ two stroke engine rated at I=166. In this case :—

$$f_c = (250 \times 30/t) + (120 \times 25{\cdot}5\ t)$$
$$= 7500/t + 3050\ t \ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (6)$$

The minimum combined stress is 9500 lb./in.^{2} with a thickness of about 1$\frac{9}{16}$″. The more usual thickness of about 2¼″

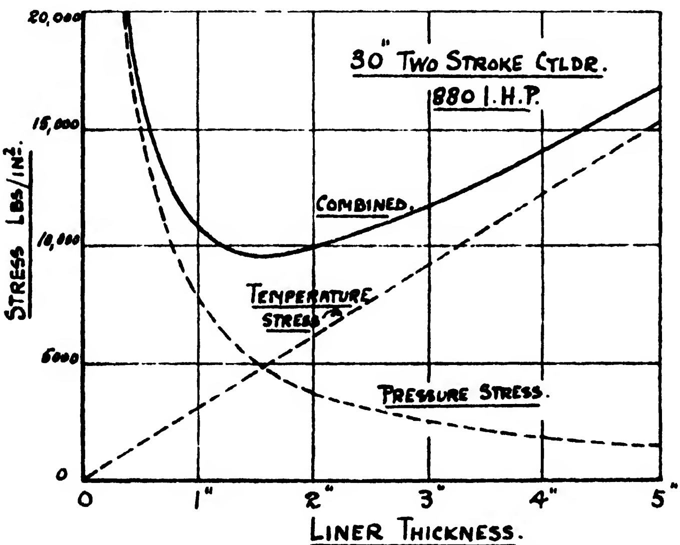


Fig. 159.

increases the stress to about 10,500 lb./in.^{2}, and a 4½″ flange would involve a stress of over 15,000 lb./in.^{2} This suggests the advisability of avoiding such heavy flanges in large cylinders subject to a high flux, and experience confirms this view.

Extensive Temperature Stresses.—Extensive temperature differences giving rise to high stress arise when the liner and jacket are cast in one piece. The jacket portion remains nearly stone cold, whilst the liner acquires a certain mean temperature which, under a constant load, slowly rises with time on account of deposits of scale until such time as the latter be removed, so that a construction which may be safe when new tends to become unsafe in service, unless deposits are prevented. Numerous failures of gas engine cylinders with integral jackets and liners have deterred most Diesel Engine builders from using this construction except in small sizes.

The traditional type of circular cylinder cover is liable to temperature stresses arising from the rigid connection of the hot bottom plate to the cool surrounding walls. Anything which tends to increase the mean temperature of the bottom plate tends to increase the temperature stress. Failures are, therefore, most frequent with highly rated engines using hard water. Tendency to failure is minimised by :—

(1) Conservative rating.
(2) Use of soft water.
(3) Introducing the water at high velocity at the centre of the cover.
(4) Using a low outlet temperature.

In this respect marine engines are at an advantage in working for the greater part of the time under conditions which admit of effective cooling with a liberal supply of clean water. The use of closed circuit cooling with heat exchangers is gaining in favour both for marine and stationary service.

With two stroke engines at normal ratings these extreme temperature differences are greatly accentuated and special constructions admitting of greater freedom for expansion become necessary. Some of these have already been referred to in Chapter XIV, together with some constructions adopted in the larger sizes of four stroke engines.

Uncooled Pistons.—With water or oil cooled pistons the temperature of the crown is probably fairly uniform over the greater part of the area, with a gradual falling off at the edge and down the sides. With uncooled pistons, on the other hand, there is a steep temperature gradient towards the edge of the crown due to the conduction of the heat to the liner. The simplest ideal case for calculation purposes is that of a flat crown of diameter B (feet) and small thickness "d" (feet), subject to a uniform flux F (1000 B.T.U./ft.^{2}hr.). If "r" is any radius less than B/2 the flow received inside this radius is

$$\pi r^2 F \times 1000 \text{ B.T.U. per hour,}$$

and this must be equal to the temperature gradient—dT/dr ×

conductivity "k" and the sectional area $2\pi rd$, we have therefore :—

$$\frac{dT}{dr} = \frac{-\pi r^2 F \times 1000}{2\pi r dk} = \frac{-rF \times 1000}{2dk}$$

$$\therefore T = Tc - \frac{r^2 F \times 1000}{4dk}$$

$$\text{and } Tc - Te = \frac{B^2 F \times 1000}{16dk} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots (7)$$

where Tc = Temperature at the centre.
and Te = „ „ edge.

Converting to inch units (7) becomes :—

$$Tc - Te = \frac{B^2 F \times 1000}{16 \times 12d \times 25} = 0{\cdot}208\ B^2F/d \ \ldots\ldots (8)$$

Under the simple conditions postulated the stress due to the uneven distribution of temperature must be proportional to $(Tc-Te)$, and temperature measurements made in cases where experience indicates that cracking is not to be anticipated indicate that a safe value of $Tc-Te$ is about 400° F.

Substituting this value in (8) we obtain the relation :—

$$d = \frac{0{\cdot}208\ B^2F}{400} = \frac{B^2F}{1920} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots (9)$$

The thicknesses of crown which result from this relation for two values of F, viz. 16 (four stroke engine rated at I=75) and 41 (two stroke engine rated at I=140) are tabulated under :—

Bore in ins.	5	10	15	20	25
d (F=16) . . .	·21	·84	1·87	3·35	5·20
d (F=41) . . .	·53	2·15	4·80	8·50	13·40

The small thickness (0·21″) required by the 5″ piston is characteristic of motor-car practice, and the other figures shew the hopelessness of trying to emulate the slender proportions of petrol engine practice, in the design of medium and large size uncooled pistons for internal combustion engines. The point to which attention is here drawn has sometimes been overlooked by critics of oil engine and gas engine designs.

The table also shews the limitations of uncooled pistons subject to the assumed values of the flux. It is clear, for example, that at the higher rating (F=41) uncooled pistons of cast iron are impracticable for cylinders above about 12″ diameter or so.

Assuming that the temperature difference is confined to some limit such as that suggested, the safety against cracking will depend on the form of the crown, whether concave or convex, the degree of curvature, the connection to the skirt, etc., and the physical and chemical properties of the material in relation to high temperatures. If the temperature difference is constant in different sizes the maximum temperature will be greatest (other things being equal) in the largest size, on account of the increased length of the paths of heat flow from the edge of the crown to the jacket. It can hardly be doubted that too high a maximum temperature tends towards cracking on account of the material becoming locally plastic when hot and developing high tensile stress when cooled.

The considerations advanced in this section do not enable the temperature stresses to be calculated, but they throw some light on necessary limitations and afford a basis of comparison on which proposed designs may be compared with known successful ones of similar form, but of different size and differently rated.

Literature.—Burstall, F. W., "Experiments on a Premier Gas Engine."—*Proc. Inst. Mech. E.*, 1908.

Callendar, H. L., British Association Report, 1910, on "Radiation from Gases."

Chaloner, J. L., "The Design of Marine Oil Engines."—*Motor-ship*, July, 1920.

Clerk, Sir D., *loc. cit.*, p. 35. Also B. A. Reports on Gaseous Explosions, 1912, *et seq.*

David, W. T., "The Pressures Developed in Gaseous Explosions."—*Engineering*, December 29th, 1922. Also "Some Properties of the Working Fluid of Gas Engines."—*Engineering*, March 10th, 1922.

Gibson, A. H., and Walker, W. J., "The Distribution of Heat in a Gas Engine Cylinder."—*Proc. Inst. Mech. E.*, 1915.

Hopkinson, B., "Collected Papers."—Camb. Univ. Press. Also B. A. Reports on Gaseous Explosions, 1912, *et seq.*

Purday, H. F. P., Articles in *Motor-ship*, June and October, 1920, and May and October, 1921. Also paper, "Marine Diesel Engines."—Diesel Engine Users' Assoc., June, 1922.

Sulzer, R., "Temperature Variation and Heat Stresses in Diesel Engines."—I.N.A., March 25th, 1926.

Letson, H. F. G., "Temperature Distribution and Thermal Stresses in a Diesel Engine Liner."—*I. Mech. E.*, 1925, No. 1.

CHAPTER XVI

TRUNK PISTONS

History.—The earliest Diesel Engines (about 1900) were of the crosshead type, but trunk pistons were very soon adopted in accordance with current gas engine practice and were almost universally used on Diesel Engines until about 1910, when Marine Diesel Engines were beginning to be considered seriously for ocean going vessels. For this service crossheads were considered practically essential and were adopted on most large Marine Diesel Engines (and many large stationary engines) built during the next decade. Prior to 1910 trunk pistons of the uncooled type of about 15″ to 25″ diameter had given an amount of trouble which could hardly be tolerated at sea. The chief of these troubles were piston seizure and cracking of crowns; both of these are reducible to negligible factors by oil or water cooling of the crown. As early as 1912 (or thereabouts) experimental single cylinder units developing 1000 B.H.P. were run with trunk pistons (cooled), but general confidence in large trunk pistons was not reached until many years later. The largest trunk pistons to-day are about 25″ dia., developing about 500 B.H.P. per cylinder and operate with entire reliability. There appears to be no necessary limitation of size or power.

Materials.—Most large and medium size Diesel trunk pistons are made of pearlitic cast iron, sometimes with the addition of small percentages of alloying metals, e.g. nickel and chromium. Many small high speed Diesel Engines for road traction, etc., as well as some larger ones for railway traction, submarine service or even stationary work, are fitted with pistons of light alloys of aluminium or magnesium. Some of the aluminium alloys give the best result in a forged condition; in other cases heat treated castings are used. A desirable feature in a light alloy is resistance to wear of the ring grooves; this has sometimes been excessive.

Piston sealing rings are of cast iron frequently centrifugal cast. Scraper rings are of cast iron or spring steel.

Gudgeon pins are commonly of mild or alloy steel case-hardened. Nitrided steel is also used.

For loose piston crowns forged or cast carbon and alloy steels (including stainless irons) have been used successfully.

Troubles to be avoided.—Before considering typical designs of trunk piston it is well to bear in mind possible troubles which can be minimised by suitable design.

(1) *Piston Seizure.*—Assuming correct clearances and lubrication piston seizure arises from distortion of piston body or cylinder liner. The surest safeguard against seizure is oil or water cooling of the piston crown; this minimises distortion of the piston body by heat from the combustion chamber and reduces the liability to hot gudgeon pin bearings. Quite apart from heat conducted from the piston crown, the frictional heat generated at the small end can in unfavourable instances cause piston seizure. The preventives are adequate lubrication of the gudgeon pin and relief of the piston surface around the gudgeon pin bosses where distortion is likely to be a maximum. Floating gudgeon pins are less liable to cause piston seizure than fixed ones, so long as they remain free in the bosses. This freedom may be lost if the top end bearing overheats for any reason.

Whether oil cooling of the crown is or is not necessary is mainly a matter of size and rating. Four stroke pistons about 16″ diameter developing 125 B.H.P. (per cylinder) can operate with uncooled crowns, but this is near the limit at which cooling is desirable unless there is some mitigating feature. For example, engines in which the shape of the combustion chamber is such that the piston crown is partially screened from the flame can safely use uncooled pistons of somewhat larger diameter than those with a more open combustion chamber.

With 2 cycle engines working at mean pressures about the same as four stroke engines, the limit of diameter for uncooled pistons is less by about one-third part. Generally, a 15″ four stroke piston and a 10″ 2 cycle piston working at M.I.P. of say 90 and 80 lb./in.^{2} respectively are both the better for piston cooling.

(2) *Cracking of Piston Crowns.*—Blast air injection engines are susceptible to star cracks in the centre of the piston crown. These are probably promoted or intensified by the turbulent action of the air blast reducing the thickness of the protective

layer of stagnant gas adjacent to the surface of the piston crown. Engines with mechanical injection appear to be less liable to this trouble than air blast engines. The smaller the cylinder the less prone are the pistons to develop heat cracks. The above indications with regard to oil or water cooling take this factor into consideration. On this point see also Chapter XV.

(3) *Piston Slap.*—This is determined mainly by piston clearance and piston surface. Slap is decreased by adopting variable clearance tapering from a minimum at the bottom of the skirt to a maximum at the crown. Figures are given later. Generous bearing surface above the gudgeon pin, preferably continued into the lands between the sealing rings; helps to eliminate slap. A piston which slaps badly at slow speed may be relatively quiet at higher speeds. Other things being similar small pistons are less liable to slap than large ones and can run quietly with less proportional length of bearing surface. Relatively long skirts below the gudgeon pin probably assist quiet running but also increase frictional drag.

(4) *Hot Small End Bearing.*—Assuming adequate bearing surfaces, suitable materials, and suitable oil, this trouble if it arises may be due either to inadequate lubricating arrangements or an insufficient supply of oil to carry away heat.

(5) *Excessive Lubricating Oil Consumption.*—Oil reaches the surface of the cylinder liner in the form of mist or splash and the piston sealing rings have a pumping action which tends to propel a certain amount of this oil towards the combustion chamber. This pumping action is reduced by minimising the clearance behind the rings. Most of the oil is scraped off the liner by the piston skirt and the scraper rings are required to deal with the remainder. The inside of the piston should be arranged to drain away the oil from the scraper rings and the oil which runs down the inside of the piston in such a way that as little as possible reaches the cylinder walls.

Typical Heavy Piston.—Fig. 160 is typical of a relatively heavy cast iron piston up to say 25″ diameter. The distances A and C from the gudgeon pin centre to the top and bottom may each be about equal to the diameter B. The distance D from the top to the first sealing ring may be about 0·2 B in large four stroke pistons. For 2 cycle pistons D is usually less say 0·1 to 0·15 B in order to reduce leakage of the charge to the ports during the early part of the compression stroke. The sealing rings E are commonly 3 to 5 in number and one or two scraper rings F are usually provided below the gudgeon pin bosses. One or more sealing rings may also be provided at this point in the case of a 2 cycle engine with scavenge ports. The scraper ring grooves are provided with horizontal vent holes. The surface of the piston is relieved at G to allow for distortion of the surface in way of the gudgeon pin bosses. The maximum thickness of metal K occurs at the crown and the minimum H at the bottom of the skirt. Thickness K may be about 0·1 B for a cooled crown or a small uncooled crown. Thickness H may

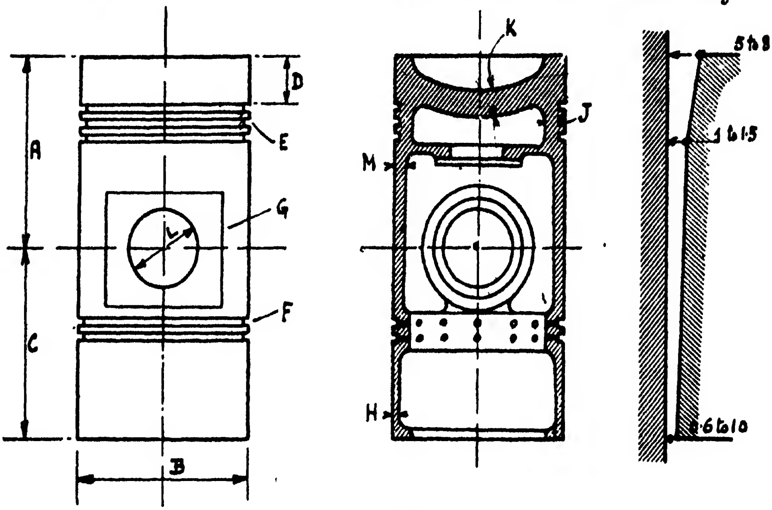


FIG. 160.

be about 0·03 B tapering to about M 0·06 B below the sealing rings. The thickness J may be about a mean between K and the wall thickness below the rings.

Below the crown there is a diaphragm closed by a studded door; this serves to prevent oil mist from carbonising on the under side of the crown if the piston is uncooled. If fluid cooling is provided the space between the diaphragm and the crown constitutes the cooling jacket.

The right-hand diagram shows the tapered surfaces much exaggerated and figures are given for diametrical (*not* radial) clearances, when cold. Sometimes three tapers are used to mitigate the abrupt change between the two gradients. The clearances shewn refer to four stroke engines. For two stroke engines the clearances may be considerably greater to the extent of 50% or more (clearances in mils.).

The mouth of the piston is shaped with a view to throwing oil drainage clear of the liner. The skirt is sometimes extended to prevent the cylinder liner being uncovered at the top dead point.

Typical Light Piston.—Fig. 161 shews the main features of a typical small aluminium alloy piston. Compared with the previous example the axial dimensions are relatively shorter

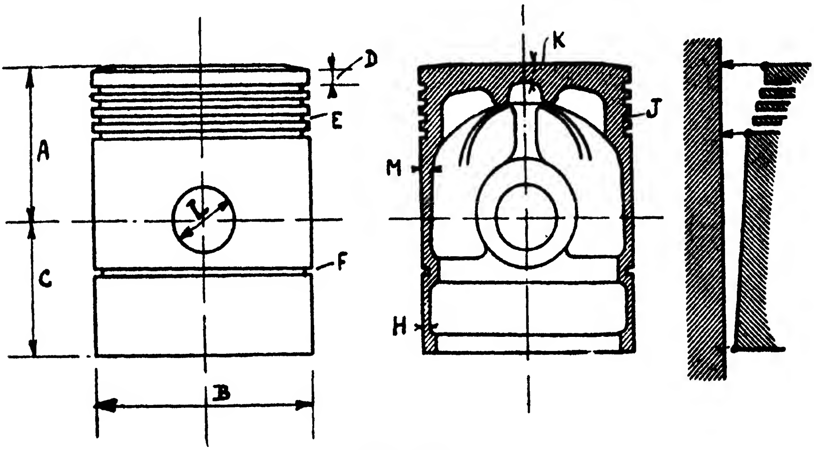


Fig. 161.

and the gudgeon pin smaller in diameter. The smaller size of the gudgeon pin is justified by,

(1) The use of high tensile alloy steel.
(2) The high pressure which can safely be sustained by bearings of small absolute size.

The dimensions of this particular example (aluminium alloy) are tabulated under :—

A=0·66 B	K=0·08 B
C=0·5 „	M=0·06 „
D=0·07 „	H=0·03 ,
L=0·27 „	

With magnesium alloy the thicknesses are usually greater. For aircraft purposes dimensions A and C as small as 0·5 B and 0·3 B respectively, have been used.

Piston Crowns.—Many different shapes of piston crown are in use and in each case the shape must be considered in relation to the combustion space as a whole (see Chapters XIV and XX). If the cylinder cover is flat or gently arched on

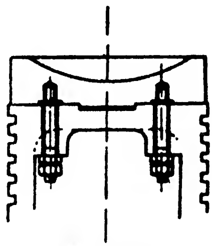


Fig. 162.

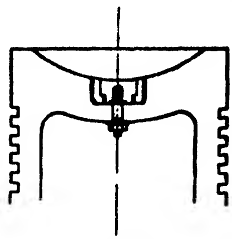


Fig. 163.

the under side, the usual shape of the crown is concave of lenticular or hemispheroidal shape. The vertical edge at the rim of the crown (Fig. 160) tends to prevent fuel reaching the cylinder walls.

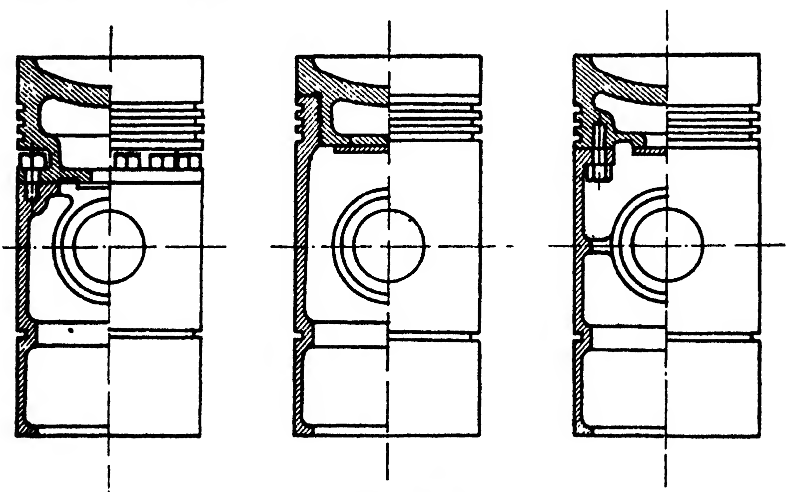


Fig. 164.

Sometimes the spheroidal shape is modified by a conical elevation at the centre.

If the combustion chamber takes the form of a cavity in the cylinder cover, the piston crown may be flat, curved outwards, or conical. Crowns curved inwards or outwards seem to be the most desirable from the point of view of the avoidance of heat cracking.

In some designs of quite small engines, the piston crown contains one or more cavities playing the parts either of combustion chambers or vessels from which gases issue on the expansion stroke and promote turbulence in the remainder of the charge.

To avoid replacing the whole piston in the event of a crown developing cracks, loose renewable crowns are sometimes fitted. Fig. 162 shews a design for an uncooled piston. Fig. 163 shews a system of plugs and cones patented by Mr. P. H. Smith and applied by him to pistons in which cracks had already appeared. Fig. 164 shews three other designs of loose tops, two of which are suitable for fluid cooling. Such loose tops are frequently made of special heat resisting material, for example, stainless iron.

Piston Cooling.—Water, or lubricating oil of the same quality as that used for the bearings may be used. Oil cooling is the simplest, as small leakages are not important. Fig. 165 shews a typical system for conveying the oil to the jacket; the outlet is arranged similarly. If water is used, arrangements are made to trap leakage and convey it outside the crank-case.

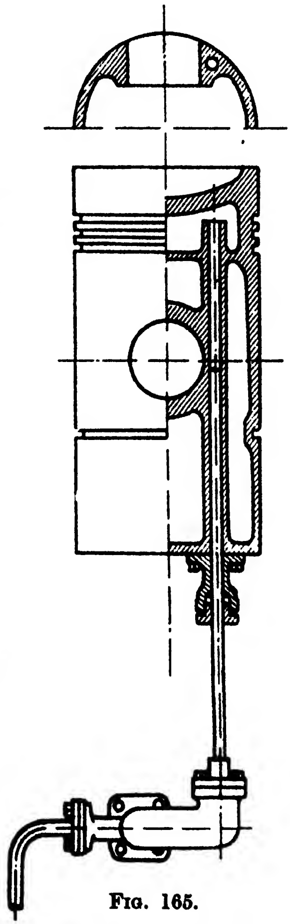


FIG. 165.

An alternative plan for conveying oil as a piston cooling medium consists of an arrangement of swinging links used also as an indicator gear. It is not easy to design such a gear to operate successfully with water.

If oil cooling is used the outlet temperature should preferably not exceed about 140° F. if local cooling conditions permit. With marine engines working under tropical conditions it is usually possible to secure an oil inlet temperature of 120° F., giving a temperature rise of 20° F. The heat absorbed by piston cooling oil is usually about 5% of the total heat supplied,

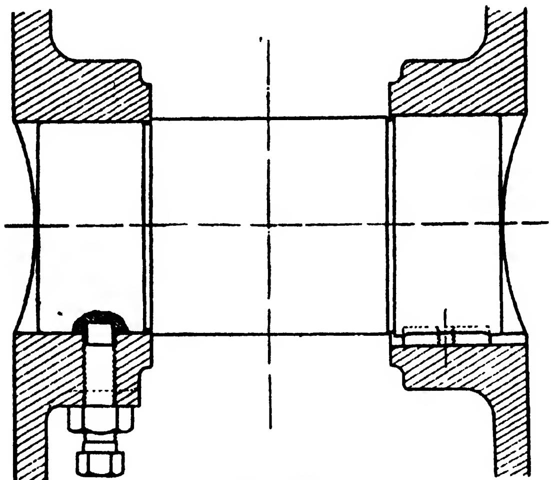


FIG. 166A.

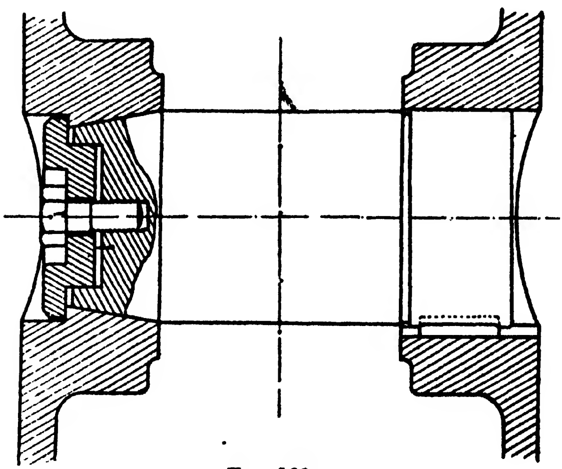


FIG. 166B.

reckoned on the nett calorific value of the oil. Consider a 1000 B.H.P. engine using 0·37 lb. of fuel per B.H.P. hr., nett cal. value=18,000 B.T.U. per lb.; specific heat of oil say 0·48.

The flow required in tons per hour for piston cooling is then :—

$$\frac{1000\times 0{\cdot}37\times 18{,}000\times 0{\cdot}05}{0{\cdot}48\times 2240\times 20}=16 \text{ tons/hr.}$$

The total discharge of the lubricating oil pump allowing for bearing oil, overload and margin would be about double this quantity.

Gudgeon Pins.—Two alternative forms of fixed gudgeon pins are shewn in Figs. 166A and 166B. The diameter and length of bearing surface are about 0·4B and 0·5B respectively. If the resultant maximum pressure (gas pressure minus inertia pressure due to piston) is 500 lb. per sq. in., the maximum bearing pressure works out at

$$\frac{0{\cdot}785\times 500}{0{\cdot}4\times 0{\cdot}5}=2000 \text{ lb./in.}^2$$

The pins are usually bored hollow to the extent of about five eighths of the external diameter to reduce reciprocating weight. The above proportions refer to relatively heavy pistons of say 12″ diameter and upwards. The gudgeon pins are of mild steel or alloy steel case-hardened or special steel hardened by the nitrogen or other process giving a surface of intense hardness (sceleroscope No. about 80 and upwards).

Suitable bearing metals are bronzes (including Delta metal), aluminium alloys and white metals.

Fixed gudgeon pins have certain disadvantages, viz. :—

(1) They cause distortion of the piston bearing surface on account of the force required to drive them in place and on account of expansion by heat.

(2) They tend to become slack in the bosses after repeated removals and as a consequence develop knock.

These circumstances have given rise to the widespread use of floating gudgeon pins referred to in the next section. Before considering these it may be remarked that the above-mentioned bearing pressure (2000 lb./in.^{2}) is greatly exceeded in small high speed engines. In such engines the gas pressure may be 1000 lb./in.^{2} or more and relatively small gudgeon pins of high tensile material are used, resulting in bearing pressures of 5000 lb./in.^{2} or more.

Floating Gudgeon Pins.—Three types of floating gudgeon pins are shewn in Fig. 167; in one example the sides of the small end bush and the piston bosses are flush in contrast to another example in which they are stepped with a view to giving greater surface in the direction of greater load. In the

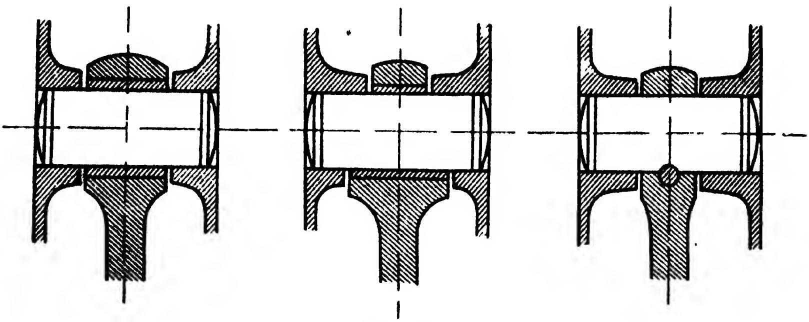


FIG. 167.

third example the gudgeon pin is fixed to the top end of the rod and is therefore not fully floating. There is great diversity of practice regarding the lubrication of the piston bosses. In

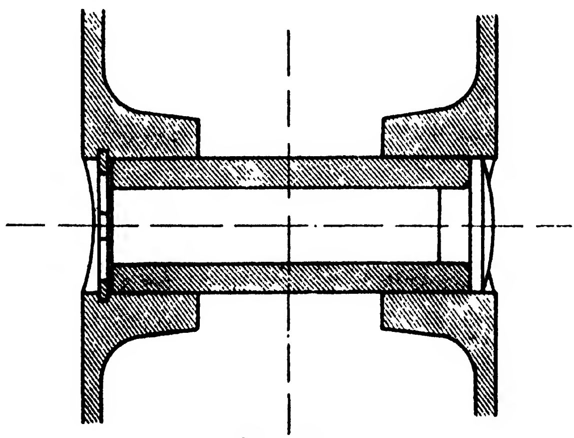


FIG. 168.

very small pistons lubrication by oil mist is sometimes relied on for lubrication of piston bosses and small end bearing, in which case holes may be drilled to facilitate access of oil to the rubbing surfaces. In this case the action seems to be one of alternate suction and discharge of oil from and to the mouth of the hole and the capillary film space respectively. In such cases there is usually no provision against oil leakage at the ends of the gudgeon pin. On larger pistons the small end is usually forced lubricated, in which case passages may be provided to feed oil to the piston bosses, via the hollow gudgeon pin.

Fig. 168 shews two ways of preventing excessive side travel of the gudgeon pin without attempting to stop oil flow, and Fig. 169 some more elaborate means by which oil flow is prevented.

Piston Rings, for Diesel Engines are usually of the Ramsbottom constant section type, the radial thickness being about 0·025 to 0·033B. With four stroke engines of all sizes and large two stroke engine the axial depth is commonly a little less than the radial thickness. With small two stroke engines the axial depth is not infrequently slightly greater than the radial thickness to give greater strength in view of running over ports. Increased gas tightness is obtainable by the use of composite rings with mutually sealed gaps. These are particularly useful in connection with worn liners.

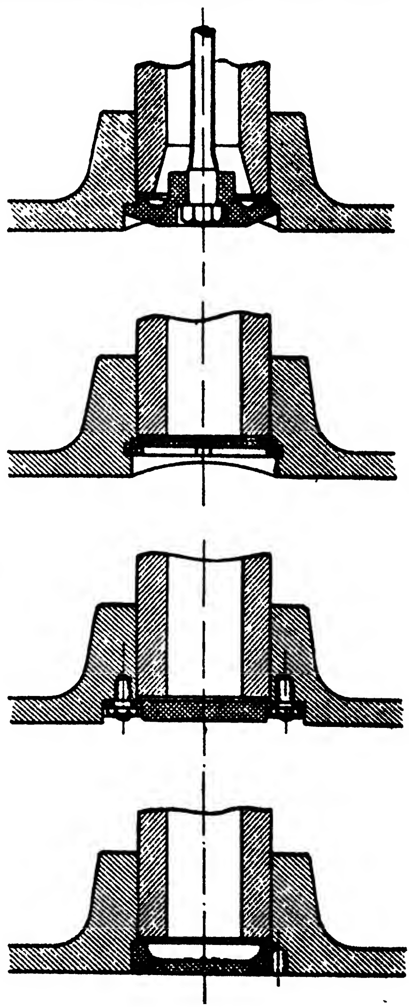


Fig. 169.

With two stroke engines the rings are usually pegged to prevent the joints riding over ports (air or exhaust). With four stroke engines the rings are seldom pegged on small engines and opinion is divided as to the advisability of pegging rings on large pistons.

Scraper rings are made in a variety of shapes; the slotted

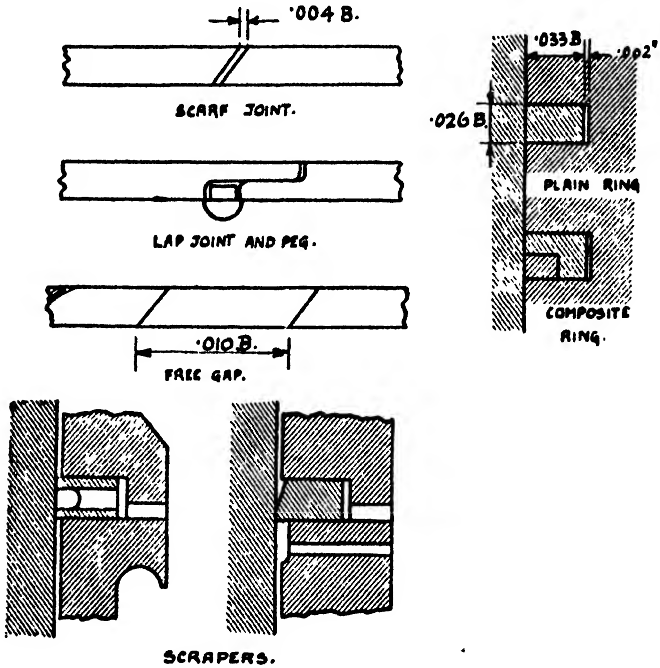


Fig. 170.

type is sometimes used as a sealing ring in conjunction with rings of the Ramsbottom or other type.

Fig. 170 shews a number of details and proportions of piston rings to supplement these brief notes.

CHAPTER XVII

CROSSHEAD PISTONS, RODS AND GUIDES

Pistons for Four Stroke Crosshead Engines.—These are generally made of not much greater length than is necessary to accommodate the rings, eight to ten in number. The provision of an extra number of rings above what is considered sufficient for a trunk piston may be attributed to :—

(1) The throttling effect lost by discarding the piston skirt.
(2) The lower speed of revolution usually associated with engines of the crosshead type.

Cooling by means of a blast of air has been used (apparently successfully) in cylinders of medium size, but liquid cooling is now almost universal. Fig. 171 exhibits different forms of

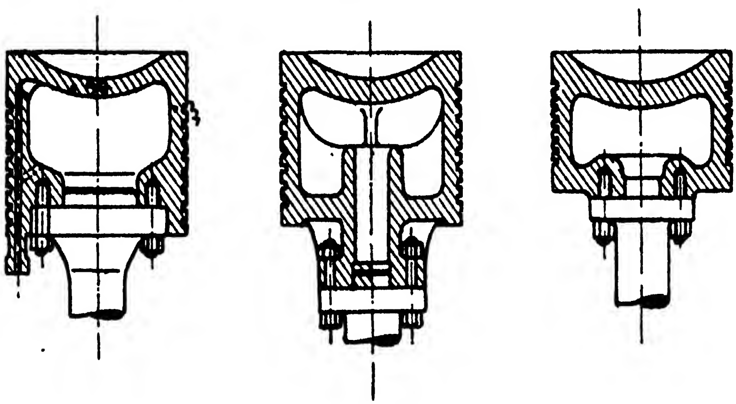


Fig. 171.

piston having one feature in common, viz. a self-supporting crown. There seems to be no doubt that internal ribs as used in the past are conducive to cracking both of cooled and uncooled pistons. They are therefore omitted in most modern designs.

Two systems of admitting the water to and leading it away from the piston are shewn in Fig. 172, which is practically self-explanatory. In each case there is a tray or diaphragm separating the crank-case from the cylinder so that any small leakage cannot reach the former.

With oil cooling a certain amount of leakage inside the crank-case does no harm. The usual arrangement consists of a pair of vertical tubes bolted to flanges on the piston body as shewn to the left of Fig. 171. Each tube passes through a gland in the top of the crank-case into a surrounding tubular vessel in the crank-case. An air vessel is usually provided to prevent oil hammer.

Pistons for Two Stroke Crosshead Engines.—The existence

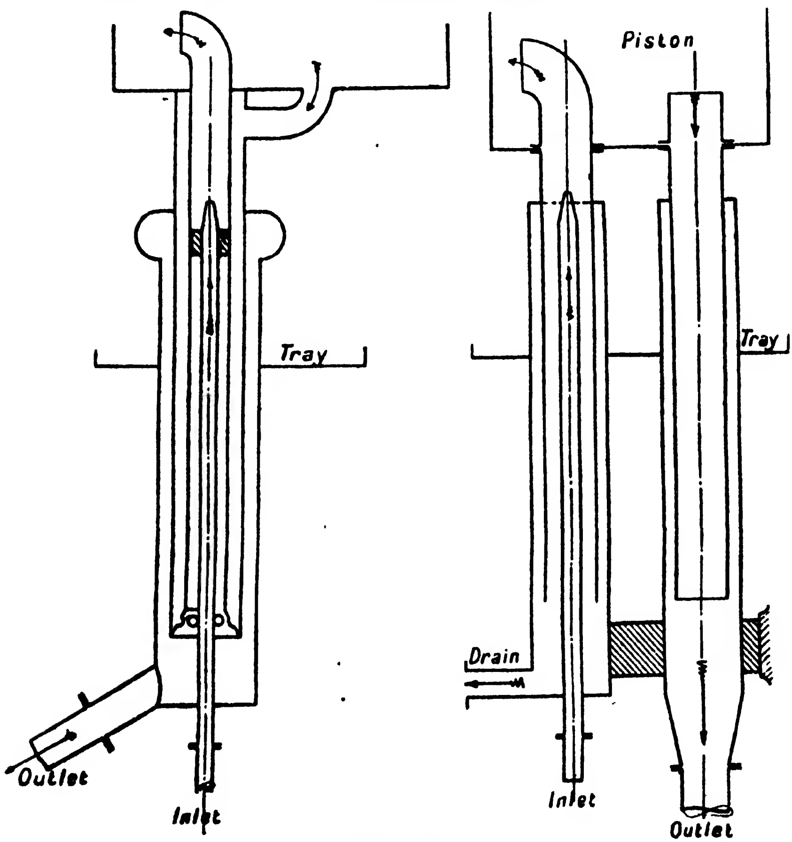


Fig. 172.

of ports leading to an air pipe or exhaust pipe at the lower end of a two stroke cylinder necessitates the provision of a skirt or extension of the piston to prevent the uncovering of these ports when the piston is at the top dead centre. The skirt usually takes the form of a light drum secured to the piston by a number of well-locked studs. It is common practice to arrange one or two inwardly expanding rings at

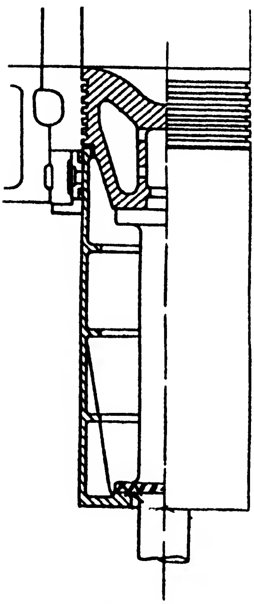


Fig. 173.

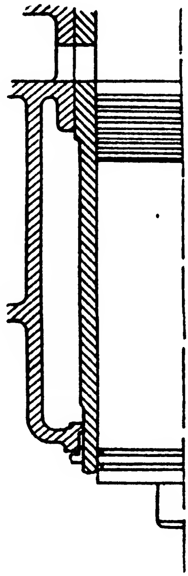


Fig. 174.

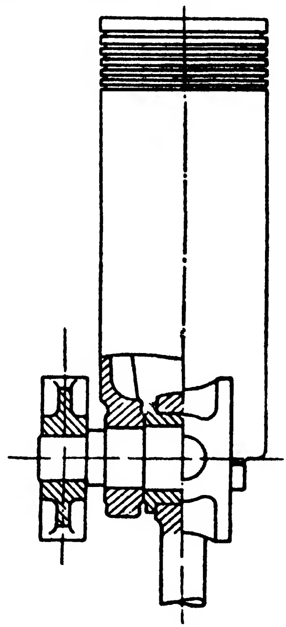


Fig. 175.

the lower end of the cylinder to prevent leakage past the skirt (see Fig. 173).

If the cylinder liner is of sufficient length these exhaust rings may be located in the skirt itself, as in Fig. 174. Such an arrangement involves a higher engine than that of Fig. 173, but facilitates conduction of heat from the piston and incidentally secures a lower mean temperature for the cylinder liner. The construction of the piston proper is generally similar to that of a 4 cycle engine, but it must be borne in mind that the conditions as to temperature are more severe than in a four stroke engine of the same size working at the same mean indicated pressure, so that the remarks of the preceding article in reference to ribs under the crown apply with still greater force to two stroke engines.

Fig. 175 shews a piston the walls of which are extended

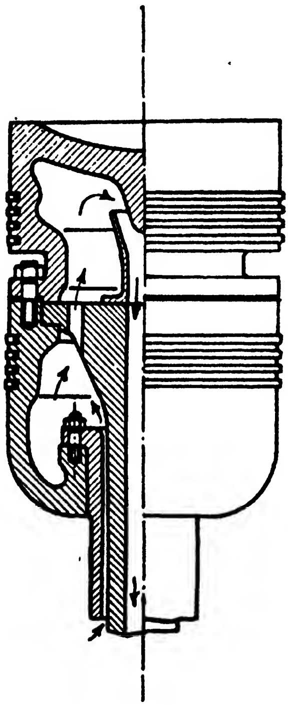


FIG. 176.

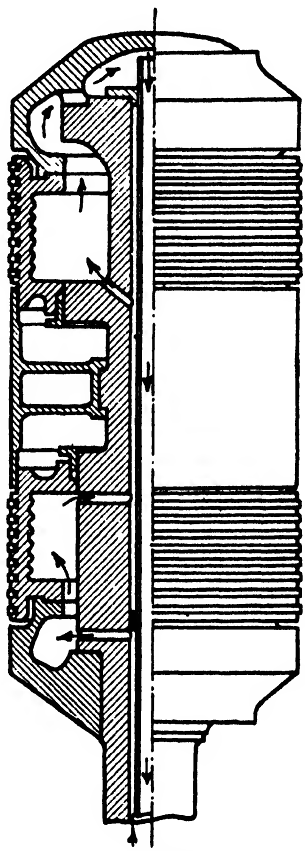


FIG. 177.

downwards and attached to the crosshead pin, thus dispensing with a piston rod.

Double-acting Pistons.—Two designs are shewn in Figs. 176 and 177 for four stroke and two stroke engines respectively. In the first (B. & W. Type) the piston is about as short as other features of the engine will allow. The piston body is in two parts, secured together by a row of studs which nip a flange forged on the upper end of the piston rod. The rings are divided between the top and bottom half. The piston rod is surrounded by a cast iron sleeve with an annular space between the rod and the sleeve. The course of the cooling oil is indicated. In the second example (Fig. 177, Worthington) the length of the piston is regulated by the stroke of the engine as the top end of the piston controls the compression of the top of the cylinder while the bottom end of the piston controls the scavenge and exhaust ports for the bottom end of the cylinder and vice versa. Two complete sets of rings are provided. The ends of the piston are shaped with regard to the exhaust and scavenge streams.

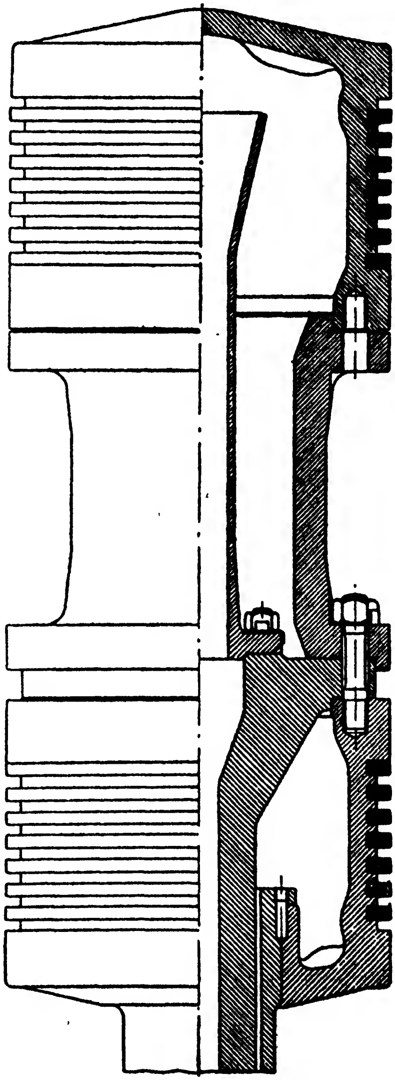


Fig. 178.

Fig. 178 shews a piston for a double-acting two stroke engine of Burmeister & Wain design. The combustion ends are of heat resisting alloy cast steel fitted with cast iron ring bands inserted in halves. Oil cooling is used. The cooling oil enters through the annular space between the piston rod and a cast iron sleeve which is free to expand at the lower end; it leaves by a hole bored through the middle of the piston rod.

Fig. 179 shews a piston of Sulzer design also for a double acting two stroke engine. In this piston water is used for cooling and the surfaces of the rod are sheathed to protect them against corrosion.

Piston Rods.—These are commonly made of forged Siemen Martin steel 28 to 32 tons/in.^{2} tensile strength, the diameter being about 0·3 B. The loading consists of gas pressure, the

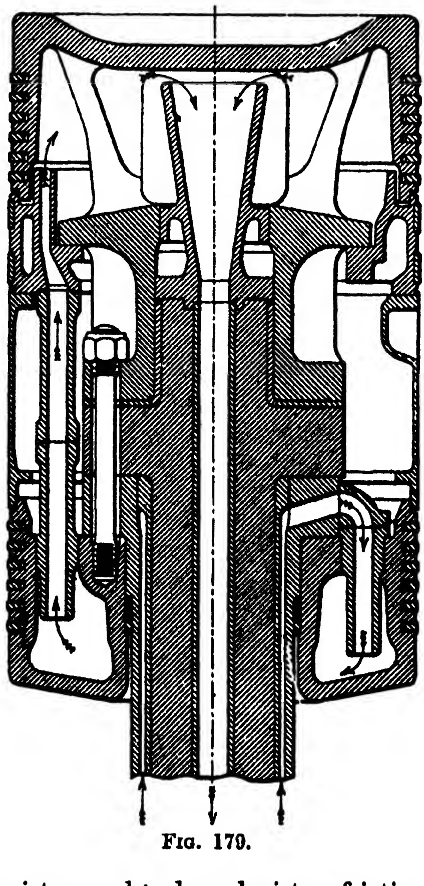


FIG. 179.

inertia of the piston and rod, and piston friction. The latter is small and not worth consideration in stress calculations. Assuming a maximum cylinder pressure of 700 lb./in.^{2} the stress in the rod at slow speed is about,

$$\frac{700}{(0{\cdot}3)^2}=\text{about } 8000 \text{ lb./in.}^2$$

If the engine is of the single acting four stroke type, the above stress is compressive and the effect of speed is to reduce the maximum compressive load to the extent of the inertia of the piston at firing dead centre. On the exhaust dead centre inertia contributes the only load of importance and the stress in the rod is tensile. At full speed, therefore, a typical stress variation will be from say 6500 lb./in.^{2} compressive to 1500 lb./in.^{2} tensile, the range being the same as at dead slow speed, assuming the maximum pressure unchanged. A range of stress of 8000 lb./in.^{2} is about one-quarter of the fatigue range of the above mentioned material.

If the engine is of the single acting two stroke type the range is somewhat less, since inertia is cushioned by compression.

With double acting engines the inertia load is relatively greater on account of the increased weight of the piston as compared with that of a single acting engine. The gas load on the underside of the piston is about 10% less than that on the top (assuming the pressures the same). At dead slow speed the stress range in a double acting two stroke piston rod of diameter$=0{\cdot}3$ B, may be :

8000 lb./in.^{2} compression to 7000 lb./in.^{2} tension giving a stress range of 15,000 lb./in.^{2}

At full speed the range may be reduced by inertia to about $6000+5500=11{,}500$ lb./in.^{2}. These figures refer to the body of the rod.

At the threaded lower end of the rod the area at the bottom of the thread is about 20% less, giving a stress range of about 14,000 lb./in.^{2}, which is equal to the fatigue stress range divided by about 2·4.

Even from these rough figures it is easy to appreciate the wisdom of precautions against corrosion fatigue, the desirability of avoiding " stress raisers," such as radial holes, particularly near the weakest section, and the importance of using fatigue resisting material.

High tensile alloy steel is not necessarily precluded for such rods, but caution is advisable. Some alloy steels, admirable for other purposes, have failed in this duty. Absolute size of the forging may be an important factor in relation to stresses induced by heat treatment and the quality of the material in large masses.

In addition to the stresses already mentioned stresses may arise through radial temperature gradients (see Dorey's paper, *loc. cit.*, p. 331).

Since the length of a piston rod rarely exceeds about fifteen times the diameter, strength to resist buckling hardly arises as a serious consideration.

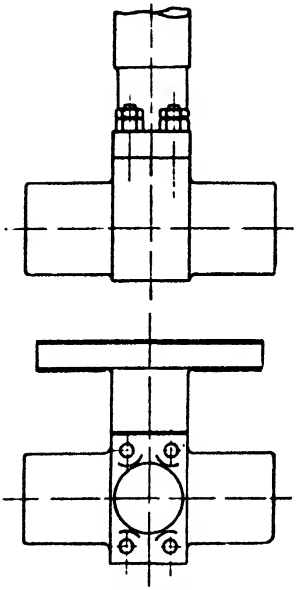


Fig. 180.

The top end of the piston rod is usually flanged and secured to the piston by a circle of bolts or studs. It is a safe rule to proportion these bolts or studs to the gas pressure load in spite of the fact that gas pressure tends to close the joint. Inadequate bolting at this point has resulted in trouble.

The lower end of the piston rod of a single acting engine may be reduced to form a bolt proportionate to the inertia load, the compression load being taken on the angular area of the shoulder. The nut being relatively small, the jaws of the connecting rod are relatively close and the forging relatively light as compared with that of a double acting engine. It is advantageous to run the bolt into the body of the rod with a taper to avoid too sudden change of section. (Fig. 181.)

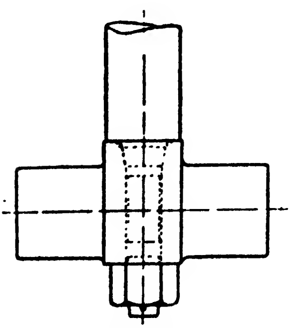


Fig. 181.

In double acting engines the telescopical tubes for conveying water or oil to and from the pistons are attached to the crosshead and connected by means of pipes or passages to the piston rod; in other respects the systems are similar to those used on single acting engines.

Crossheads and Guides.—For marine engines the slipper guide, shewn in Fig. 182, is the favourite. The bearing surface is usually made equal to the piston area,* and the maximum bearing pressure with a connecting rod 4·5 cranks long then has a value of about 55 lb. per sq. in. The slipper itself

* *i.e.* the sectional area of the cylinder.

is of cast steel, white-metalled on ahead and astern faces. The studs securing the slipper to the gudgeon block must be adequate to carry the maximum guide pressure when running astern. The area of the gudgeon bearing is based on a bearing pressure of about 1500 lb. per sq. in. The ahead guide face is of cast iron sometimes water cooled. The astern bars are frequently of forged steel, secured by fitting bolts. The stress in the latter is usually very moderate, as stiffness is the chief consideration. The proportions given in Fig. 182 are of course

Width of crosshead face 0·75 B.
Depth „ „ 1·13 B.
Thickness „ plate 0·10 B.

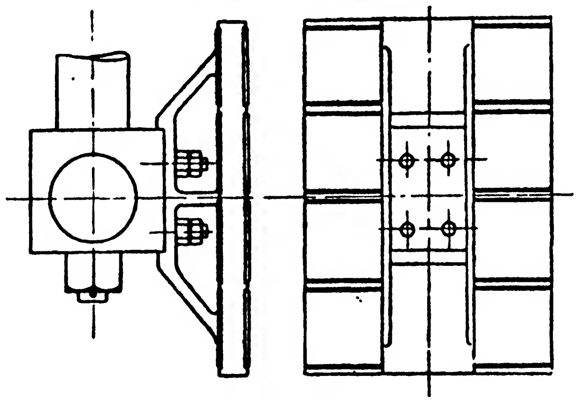


FIG. 182.

approximate only and subject to modification to suit different conditions.

The type of guide block indicated on Fig. 175 is well known in connection with paddle-steamers and locomotives, and needs no further description here. For land engines double semicircular guides are sometimes used, particularly when the cylinder and frame are cast in one piece. In general, the crossheads and guides used in Diesel Engine construction differ but little from those commonly fitted to steam engines and large gas engines, the most important point of difference being the gudgeon pin and its bearing, which require to be liberally dimensioned to withstand the high maximum pressures to which they are subject. It is also desirable to provide means to prevent carbonised oil from the cylinder from reaching the guide surface.

Guide Pressure Diagrams.—A diagram shewing approximately the guide pressure at any crank angle is very simply obtained from the twisting moment curve in the manner described below, with reference to Fig. 183.

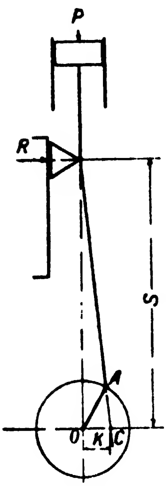


FIG. 183.

P = Piston load in lb.
R = Guide reaction in lb.
T = Twisting moment in in. lb.
S = Height of gudgeon pin centre above the centre of the crank-shaft.

$$\text{Now } R = \frac{P.k}{S}$$

$$\text{But } T = P.k$$

$$\text{Therefore } R = \frac{T}{S}$$

The rule is therefore: Divide the turning moment at any instant by the distance from the gudgeon pin centre to the crank-shaft centre, and the result is the guide reaction at the same instant. It would appear that the guide reaction and the twisting moment should change sign simultaneously. This is not quite the case, for the following reason :—

The twisting moment curve contains an inertia element in which an approximation is obtained by dividing the mass of the connecting rod in a certain proportion between the revolving and reciprocating parts. This approximation, though good so far as vertical forces are concerned, gives very inaccurate values for horizontal forces. Also the centrifugal effect of the revolving parts of the rod influences the guide reaction but not the twisting moment. These discrepancies are of very small importance with the piston speeds at present obtaining. For a full discussion of the influence of the connecting rod inertia forces on the guide reaction, the reader is referred to Dalby's "Balancing of Engines." A table of values of $\frac{S}{l}$ where "l" = the length of the connecting rod, is given below for various crank angles, assuming a rod 4·5 cranks long.

Crank angle	0°	20°	40°	60°	80°	100°	120°	140°	160°	180°
Values of $\frac{S}{l}$	1·22	1·21	1·16	1·09	1·02	0·94	0·87	0·82	0·79	0·78

Literature.—Dorey, S. F., "The Problem of the Water Cooled Piston Rod in Two Stroke Cycle Double Acting Oil Engines."—I.N.A., 1933.

CHAPTER XVIII

CONNECTING RODS

Types of Connecting Rod.—Apart from details the connecting rods of Diesel Engines can be roughly classified as follows, viz. those suitable for :—

(*a*) Moderate speed single acting engines.
(*b*) Double acting engines.
(*c*) High piston speed engines.

Class A is intended to include most stationary and marine trunk and crosshead engines for ordinary commercial purposes running at speeds from 100 to 400 R.P.M. The material is usually Siemens Martin forged steel of the same quality as that used for crank-shafts. Since inertia forces are not excessive there is no object in cutting weight to a minimum.

Class B is characterised by about equal duty in tension and compression, the need of strong bolts and caps, top and bottom.

Typical instances of Class C are rods for bus and locomotive and aircraft engines. Lightness is obtained by the use of alloy steels capable of sustaining high stresses with safety..

The following sections (until further notice) are concerned with Class A.

Connecting Rod Bodies.—The section of the body or shaft of the rod is generally circular, or part circular, with flattened sides. The latter section is slightly lighter for a given strength, but involves an extra machining operation.

The body is sometimes tapered gently from the big to the small end and this can be defended on logical grounds as will appear later. The amount of taper which can be justified is small and more usually the body is made parallel.

A hole is usually drilled up the rod for forced lubrication of the small end ; the diameter of the hole may be anything up to about five-eighths of the diameter of the rod.

Big Ends.—Fig. 184 shews two forms of big end, type A being the cheapest and that most commonly used. Type B is the strongest, but suffers from the disadvantage of providing no facility for adjusting the compression by means of liners.

Returning to type A, the "brasses" are usually of cast steel lined with white metal. With a stronger section, as shewn in Fig. 185, cast iron may be used instead of cast steel, with satisfactory results, but the practice is uncommon. The bolts are usually reduced to the core diameter between fitting lengths, as shewn in Fig. 184, A; but it is doubtful if full diameter

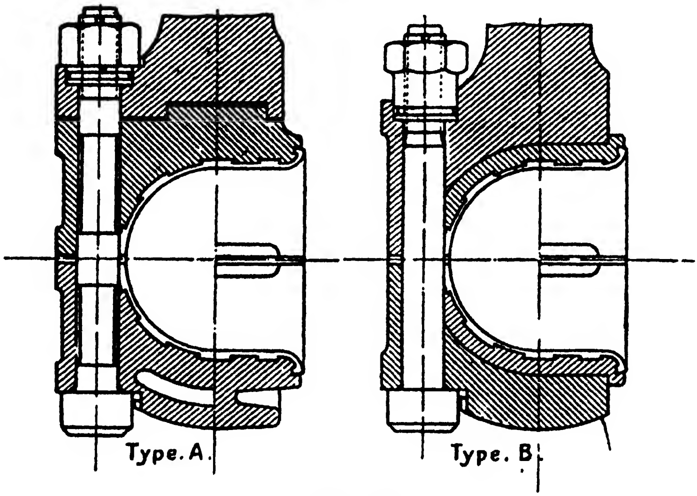


FIG. 184.

bolts are weaker under the conditions to which they are subject in trunk piston engines. In addition to tensile stresses the big end bolts have to resist shearing forces between the two brasses and also between the crown brass and the palm end of the rod. Partial relief of this duty is afforded by the following means, one or more of which are generally used in good designs :—

(1) Spigoting the two brasses into each other, as in Fig. 186. This is very rarely done.
(2) Spigoting the crown brass into the palm of the rod, as in Fig. 184, type A.
(3) Providing fitting rings, half in the palm and half in the crown brass at the bolt holes (Fig. 187).

Small Ends.—Various types of small end for trunk engine rods are shewn in Fig. 188.

With type A the chief difficulty is to find room for bolts of adequate strength. If a big end of type B, Fig. 184, be fitted, it is possible to make provision at the small end for adjusting the compression, as in Fig. 188, types B and D. Type C combines strength and adjustability of the bearing itself, but makes no provision for altering the compression. Type E contains a solid bush, which must be replaced when worn

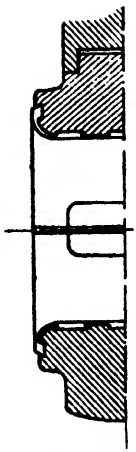


Fig. 185.

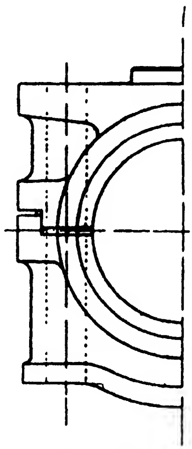


Fig. 186.

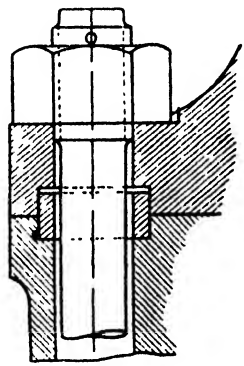


Fig. 187.

(the life is usually several years with forced lubrication), and adequate section of metal round the bush must be provided to prevent the eye becoming enlarged (see approximate proportions on Fig. 188). The brasses are usually of phosphor bronze.

The forked end of a marine connecting rod is shewn in Fig. 189, on which approximate proportions are noted in terms of the cylinder bore. It differs from the similar member of a marine steam engine chiefly in the following points :—

(1) The cap brass is not provided with a steel keep.
(2) The fork gap is relatively narrower, owing to the piston-rod nut having to deal with inertia only.
(3) The brasses are of cast steel, instead of gun-metal.

Points of Detail.—The lubrication of the big and small ends has been referred to elsewhere (see Chapters VIII and XXIV). In very small engines the lubrication of the small end is sometimes allowed to depend on oil mist from crank-case.

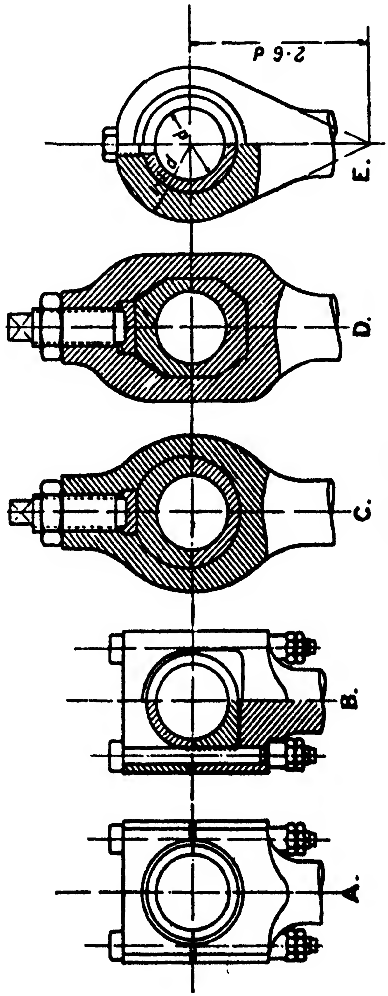


FIG. 188.

An alternative method consists of a steel pipe clipped or welded to the rod and leading oil from the big to the small end.

When air compressors, oil pumps, or other gear are worked by links from the connecting rod the connection to the latter should be made near the top end, as in Fig. 190, so that the strength of the rod to resist buckling is not impaired.

Strength of Connecting Rods.—The forces acting on the connecting rod are :—

(a) The joint effect of
- (1) The pressure load on the piston.
- (2) The inertia of the piston and crosshead.
- (3) The piston-ring friction.
- (4) The lubricated friction of piston and crosshead, all divided by the cosine of the angle of obliquity of the rod.

(b) The longitudinal component of the inertia of the rod itself.

(c) The transverse component of the inertia of the rod itself.

(d) The friction of the top and bottom end bearings.

When considering the compressive stress of the rod on the expansion stroke one is on the safe side in neglecting item (4). The tensile forces attain their maximum at the top dead centre following the exhaust stroke, and the reciprocating parts being then at their position of minimum speed, item (4) may probably be neglected with safety.

Item (b) is estimated with sufficient accuracy by the usual

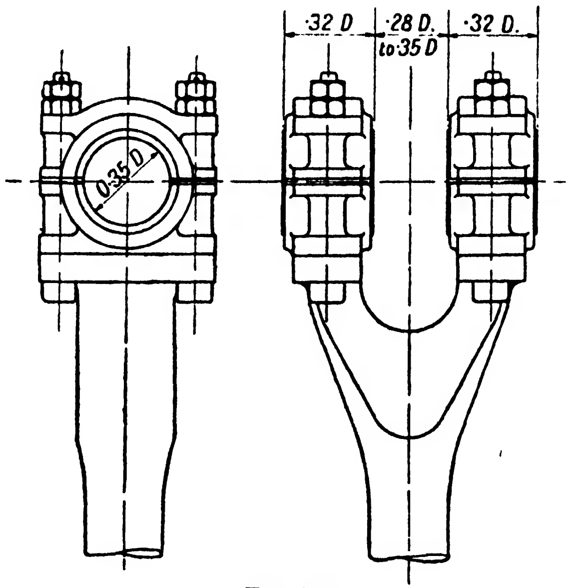


FIG. 189.

procedure of dividing the mass of the rod between the reciprocating and revolving parts in that ratio in which the centre of gravity of the rod divides its line of centres.

Item (c) gives rise to a bending moment, the maximum value of which is given approximately by the formula :—

$$f=\frac{\left(\frac{n}{100}\right)^2 .R.L^2}{26d} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots (1)$$

Where f = Bending stress.
n = Revolutions per minute.
R = Crank radius in inches.
L = Length of connecting rod in inches.
d = Diameter of rod in inches (mean).

The sign of the bending moment is such that the latter always tends to bend the rod outwards to the side on which the crank stands. The variation in the magnitude of the stress over the length of the rod for various positions of the crank relative to the top dead centre is shewn in Fig. 191 for a rod five cranks long. The stress varies as sin $(\theta+\phi)$, where θ=the crank angle relative to top dead centre and ϕ=the angle of obliquity of the rod, and as $Lx-\frac{x^3}{L}$, where L is the length of the rod and x is the distance from the small end of the section under consideration.

The assumptions made use of in equation (1) are that the

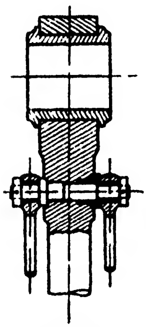


FIG. 190.

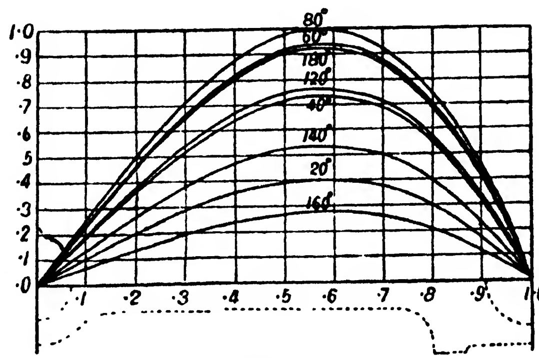


FIG. 191.

rod is of uniform section, and, as is usually assumed in books on applied mechanics and machine design, that the influence of the rod ends is small.

Item (d) may be estimated on the assumption that the coefficient of friction attains Morin's value of 0·15 for slightly greasy metal at the top or bottom end or at both ends simultaneously. The effect of journal friction is to divert the line of thrust from the centre line of the rod, and the amount of this deviation is found by the well-known graphical construction shewn in Fig. 192, in which the line of thrust is shewn to be tangential to two very small circles whose radii are equal respectively to the radii of the crank-pin and gudgeon pin multiplied by the coefficient of friction. The deviation at any point of the rod of the line of thrust from the line of centres will be denoted " d."

The effect of this deviation is to bend the rod into an " S " shape, as shewn much exaggerated in Fig. 193. This form of failure is one consistent with the Eulerian theory of pure buckling, but usually regarded as an improbable solution. The author has actually seen one instance of a Diesel Engine rod failing in this way, and the effect was probably due to the

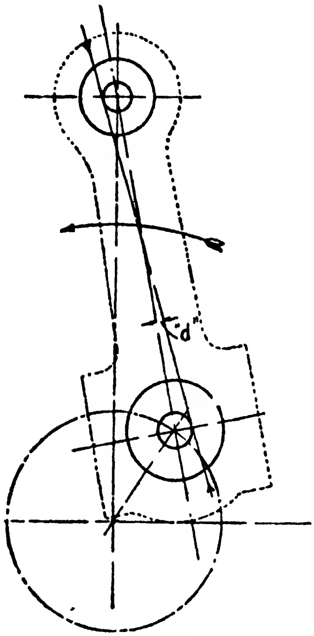


FIG. 192.

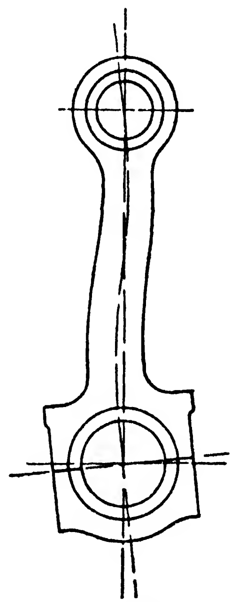


FIG. 193.

causes indicated above, arising in acute form. To be on the safe side, it seems advisable to consider two cases :—

(1) Coefficient of friction negligible at the gudgeon-pin and equal to 0·15 at the crank-pin.

(2) Coefficient of friction negligible at the crank-pin and equal to 0·15 at the gudgeon-pin.

The weak sections are clearly those (cf. Fig. 196, KK and SS) where the big and small ends merge into the shaft with a radius, and incidentally those selected by the draftsman when giving dimensions for the diameters of the rod. The reasonableness of making the rod tapered will now be apparent, as the bending effect is clearly greater at the large end of the rod on account of the crank-pin being of greater diameter than the gudgeon-pin.

Returning to Fig. 192, it is evident that the deflection at any point of the rod should, to be strictly accurate, be added to the deviation of the line of thrust at that point, in order to find the bending moment, and further, this new bending moment involves the construction of a revised deflection curve, and so on. This evidently calls for some form of mathematical treatment, which with certain approximations can readily be applied. It will be found, however, that the deflections

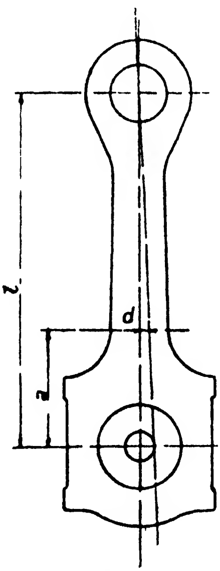


Fig. 194.

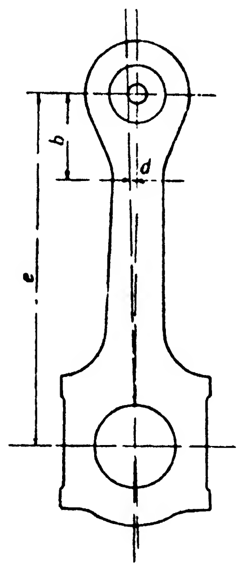


Fig. 195.

involved are small compared with the deviation of the line of thrust, and whatever error may be incurred can be considered to be covered by the factor of safety.

On these assumptions, " d " for the weak sections KK and SS is given by the following :—

Case (1), Fig. 194 . . $d=0{\cdot}15\,R_c.\,\dfrac{l-a}{l}$ ....(2)

Case (2), Fig. 195 . . $d=0{\cdot}15\,R_g.\,\dfrac{l-b}{l}$ ....(3)

And $$f = P\left(\frac{1}{A} + \frac{d}{Z}\right) \quad \ldots\ldots\ldots\ldots (4)$$

Where P = The thrust in the rod in lb.
A = Sectional area of rod in sq. in.
Z = Sectional modulus of rod in in.^{3}
f = Maximum compression stress in lb. per sq. in.
R_c = Radius of crank-pin.
R_g = ,, ,, gudgeon-pin.

It will be noticed that the strut formulæ of Euler, Gordon and Rankine and others have not been utilised above. It appears to the writer that these formulæ are irrelevant to the case of Diesel Engine connecting rods, for the following reasons :—

(1) Euler's formula is based on the calculation of the load required to produce elastic instability, and with short rods the stress commonly works out at a higher value than the ultimate strength.

(2) The Gordon and Rankine formulæ are based on experimental values of the buckling stress under static conditions, and give no indication of the strength under repetitions of stresses, which are generally only a fraction of the buckling load.

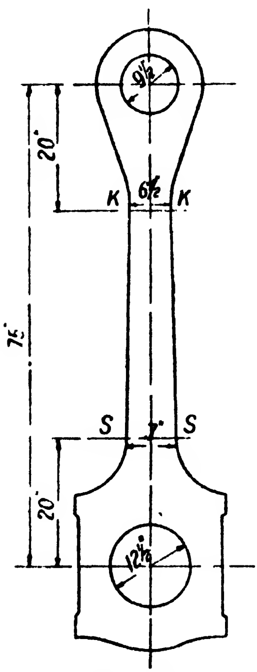


Fig. 196.

It seems more rational, therefore, to calculate the maximum direct stresses as closely as possible and to apply to the approximately known fatigue stress of steel a factor of safety of 2·5 to 3, which is known to be satisfactory in other cases.

In view of occasional abnormal pressures of about 1000 lb. per sq. in., it is interesting to see what factor of safety a given rod has for meeting such contingencies, and the table of buckling stresses given on page 461 may be used for this purpose.

Example of Stress Calculation for Connecting Rod.—

FOUR STROKE CYCLE.

Bore of cylinder	24 in.
Stroke	30 in.
Revolutions per minute	200
Weight of piston (trunk)	2200 lb.
,, connecting rod complete	2500 lb.

Main dimensions of rod. as in Fig. 196 :—

(1) Calculation of stress due to thrust 30° after firing centre.

Piston pressure load $=0{\cdot}785\times 24^2\times 500=226{,}000$ lb.

Inertia load 30° after dead centre

$$=\left(\frac{2\pi\times 200}{60}\right)^2\times 15\times\frac{2200+0{\cdot}35+2500}{386}\times(\cos 30^\circ+\tfrac{1}{5}\cos 60^\circ)$$
$$=51{,}000 \text{ lb.}$$

Resultant vertical force $=226{,}000-51{,}000=175{,}000$ lb.

At 30° after dead centre the obliquity of the rod is 6°.

$\therefore$ Connecting rod thrust $=175{,}000\div\cos 6^\circ=176{,}000$ lb. $=P$.

At section KK :— Area $=33{\cdot}2$ in.$^2=A$

Section modulus $=27{\cdot}0$ in.$^3=Z$

$$\text{Deviation of line of thrust}=\frac{0{\cdot}15\times 4{\cdot}75\times 55}{75}=0{\cdot}52=d$$

$$\text{and the stress } f=176{,}000\left(\frac{1}{33{\cdot}2}+\frac{0{\cdot}52}{27{\cdot}0}\right)=8970 \text{ lb./sq. in.}$$

At section SS :— Area $=38{\cdot}5$ in.$^2=A$

Section modulus $=33{\cdot}7$ in.$^3=Z$

$$\text{Deviation of line of thrust}=\frac{0{\cdot}15\times 6{\cdot}25\times 55}{75}=0{\cdot}69 \text{ in.}$$

$$\text{and the stress } f=176{,}000\left(\frac{1}{38{\cdot}5}+\frac{0{\cdot}69}{33{\cdot}7}\right)=176{,}000\times 0{\cdot}0464$$
$$=8160 \text{ lb./in.}^2$$

(2) Calculation of stress due to inertia bending at 30° after dead centre.

Maximum inertia stress in rod, from equation (1),

$$=\frac{2^2\times 15\times 75^2}{26\times 6{\cdot}75}=1920 \text{ lb./in.}^2$$

From Fig. 191 the fraction of this maximum applying to position KK at 30° after dead centre is 0·37.

$\therefore$ Inertia bending stress at section KK $=0 \cdot 37 \times 1920$
$=710$ lb./in.^{2}

The fraction applying to section SS at 30° after dead centre is 0·49.

$\therefore$ Inertia bending stress at section SS $=0 \cdot 49 \times 1920$
$=940$ lb./in.^{2}

(3) Resultant stress at KK $=8970-710=8260$ lb./in.^{2}

Since the bending actions due to inertia and eccentricity of thrust are of opposite sign,

Resultant stress at SS $=8160+940=9100$ lb./in.^{2}

Since the two bending actions are of the same sign.

(4) Tensile stress at SS at beginning of suction stroke.

$$\text{Inertia force}=\left(\frac{2\pi \times 200}{60}\right)^2 \times 15 \times \frac{2200+0 \cdot 35 \times 2500}{386} \times 1 \cdot 2$$
$$=63{,}200 \text{ lb.}$$

$$\text{Stress at SS}=63{,}200\left(\frac{1}{38 \cdot 5}+\frac{0 \cdot 69}{33 \cdot 8}\right)=2930 \text{ lb./in.}^2$$

The total range of stress is therefore

$$9100+2930=12{,}030 \text{ lb./in.}^2$$

The range of stress required to produce fracture of mild steel by fatigue appears to be about 35,000 lb./sq. in., so the factor of safety is about 3.

Calculation on the above lines might with advantage be made for several different positions of the crank.

It is evident that the results of the calculation depend very largely on the assumed conditions of journal friction, but it should be borne in mind that almost any possible combination of unfavourable conditions is a probable contingency in the combined lives of a number of similar engines.

Proportions Found in Practice.—In the preceding example the mean diameter of the rod is approximately 0·28 of the diameter of the cylinder, a very favourite ratio in practice. In different designs this ratio varies from about 0·26 to 0·33.

The maximum and minimum diameters are usually about 5% more and less than the mean, if the rod is tapered.

Connecting Rod Bolts.—In four stroke engines these are usually proportioned to the maximum inertia load with a nominal stress of 4000 to 6000 lb./in.^{2} based on the inertia and centrifugal loads divided by the area of two bolts at the bottom of the threads. With trunk piston engines, failure when it occurs is generally due to piston seizure, to which it would be difficult to apply definite rules of calculation. Danger of seizure is largely eliminated by the use of a crosshead. The strength of connecting rod bolts for four stroke Diesel Engines forms the subject-matter of a paper by Mr. P. H. Smith, read by him before the Diesel Users' Association and containing the results of several years' experience. For the big end it appears that the bolts seldom fail if made of a diameter 12 to 13% of the cylinder bore. Mr. Smith also points out that the bolts, for both big and small ends, are not equally stressed, as may easily be seen by reference to Fig. 197.

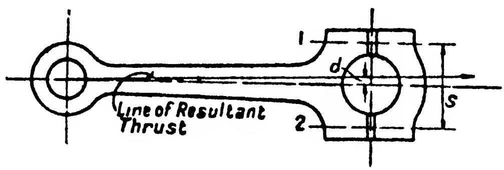


FIG. 197.

Owing to the deviation of the line of pull from the centre line of the rod, that bolt (No. 1 in the Fig.) which first passes the top dead centre at the beginning of the su tion stroke is nearer the line of pull than the other bolt, and consequently more highly stressed.

If P = Resultant pull in lb.
S = Centres of bolts in in.
d = 0·15 radius of crank-pin.

Then

$$\text{Pull in bolt No. 1} = \frac{P \times \left(\frac{S}{2} + d\right)}{S}$$

$$\text{Pull in bolt No. 2} = \frac{P \times \left(\frac{S}{2} - d\right)}{S}$$

With crosshead engines the small end bolts have to carry the inertia load due to piston, piston rod and crosshead, and also any frictional forces acting on the piston and crosshead. The latter being more or less indeterminate, it is customary to allow a nominal stress on these bolts about 30% less than that allowed for the big end bolts. The bolts or studs connecting the crosshead to the piston rod and the latter to the piston are given a large margin of strength for the same reason. Connecting rods for two stroke engines are not as a rule distinguishable from those for four stroke engines, as the possibility of compression being lost has always to be kept in view.

Material for Connecting Rod Bolts.—Large bolts say 4″ diameter and upwards are usually made of forged Siemens Martin steel of about 28 to 32 tons per sq. in. tensile strength. Small bolts for high speed engines are usually made of alloy steel for example heat treated nickel chrome steel having a tensile strength of about 50 to 60 tons per sq. in. and Izod value about 30 to 40 ft./lb.

Bolts of medium size may be turned from heat treated bars of 3% nickel steel, 40 to 50 tons tensile.

So long as bolts remain tight the fluctuation of stress is not great but there is always the possibility of overstrain on tightening and the development of undetected slackness in service. Experience seems to shew that high tensile steels with good Izod value give the best service in small and moderate sizes.

Large mild steel bolts of 3″ or 4″ and over seldom give any trouble.

Double-acting Connecting Rods.—Double-acting pistons are very much heavier than single-acting ones of the same diameter, so that other things being equal the inertia loads are correspondingly heavier in double-acting engines. At slow speeds a double-acting piston rod is subject to alternate compressive and tensile loads of nearly equal amount. At full speed both these loads are reduced by inertia effects, the reduction being more at the top dead centre than the bottom on account of obliquity of the rod. Under normal running conditions, therefore, the total range of load may be little if any more than that of a single-acting cylinder of the same size. In practice the diameters of piston rods and connecting rods expressed as fractions of the cylinder diameter are nearly the same for single- and double-acting engines. The connecting rod bolts of the latter are necessarily made of much greater section to withstand the pull due to ignitions at the bottom of the cylinder. This involves rather heavy connecting rod ends (both top and bottom) still further augmenting the inertia and centrifugal forces. Double-acting Diesel Engine connecting rods, even of long stroke engines, have, therefore, a characteristically stumpy appearance which seems to be almost unavoidable so long as moderate stresses are adopted. The connection of the crossheads to the piston-rods may be either by a pair of nuts allowing adjustment of the vertical position of the piston, or alternatively by a conical shoulder and a single nut. In either case adequate locking of the nut or nuts is necessary.

The top and bottom connecting rods bearing caps are usually designed for strength by the conventional beam formulæ adopting the conservative stresses of marine practice. Massive proportions result from this procedure and it is an interesting

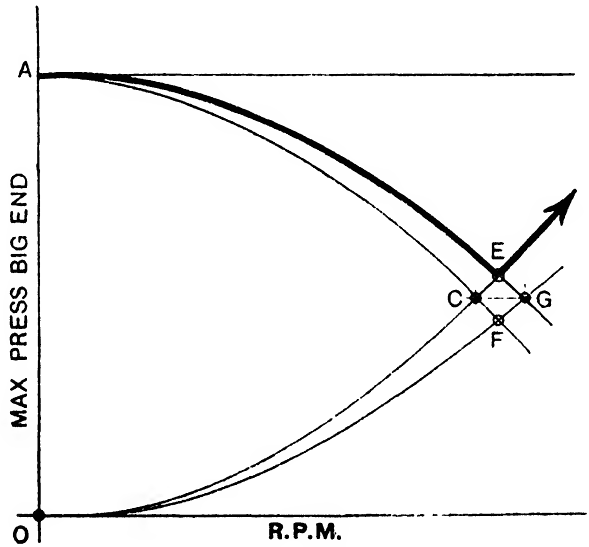


FIG. 198.

question whether more accurate analysis would indicate the possibility of using lighter proportions with adequate margin of safety. It has to be remembered that stiffness, to resist distortion of the bearing surface is necessary as well as strength.

Inertia and centrifugal effects have great influence on connecting rod bearing pressures. At slow speeds the maximum bearing pressures occur at the ignitions. In four stroke double-acting engines the maximum bearing pressure at the big end at full speed may occur at the end of the top exhaust stroke. When this occurs the maximum bearing pressure is about half its value at dead slow speed. In this respect high inertia and centrifugal forces may be actually advantageous. According to some authorities maximum bearing pressures are of little account, and what really matters is the product of mean bearing pressure and rubbing speed. This is probably a fair approximation to the truth in regard to engines running at say 2000 R.P.M. and over. So far as slow running engines are concerned there are indications that maximum bearing pressures cannot be disregarded. The influence of inertia and centrifugal force on

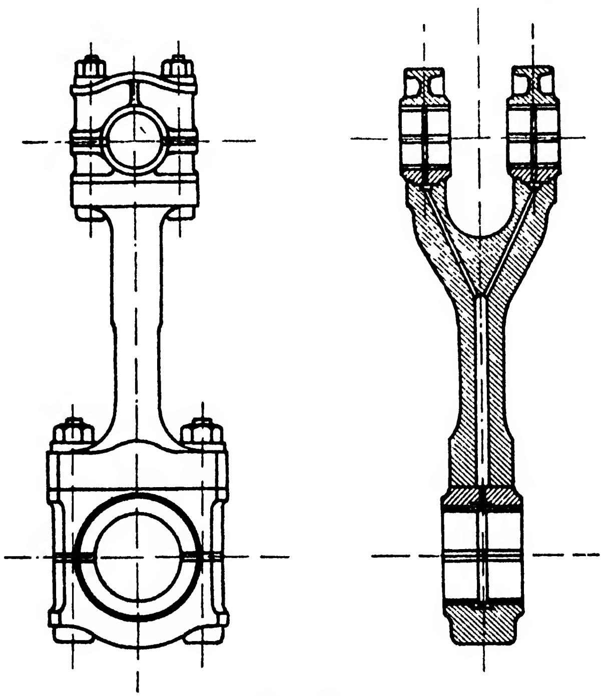


Fig. 199.

the bearing pressures may be explained with reference to Fig. 198. Line A E→ refers to a double-acting four stroke engine. At point A (R.P.M.=O) the bearing pressure (neglecting dead weight) is due to gas pressure only and approximately equal top and bottom. Line O C E (parabola) represents the bearing pressure due to inertia and centrifugal force at top dead point and line O F G at the bottom dead point. At E the inertia and centrifugal force at bottom dead point have reduced the pressure to a little over one half its value at O A, and at higher R.P.M. the maximum bearing pressure is determined by inertia and centrifugal force at the top dead point (line C E). In a two stroke double-acting engine each dead centre is a firing centre in opposition to inertia and centrifugal forces, so that the maximum pressure (at dead centre) continues to fall beyond E.

Provided bearing clearances are maintained at a low figure (about 1/2000 of the diameter), particularly in the early life of the engine, wear of bearings, journals and crank-pins is exceedingly small in large slow-running Diesel Engines of single or double acting type, both two cycle and four cycle. When remetalling is required, it is usually on account of cracked whitemetal. This also is minimised by fine running clearances.

Fig. 199 shews a typical connecting rod for a slow speed double acting Diesel Engine.

Connecting Rods for High Speed Engines.—For piston speeds much in excess of, say, 1300 ft. per min. it becomes necessary (particularly with four stroke engines) to provide connecting rods which are light in comparison with those which are usually fitted to slower speed engines, for the following reasons, viz. :—

(1) To reduce inertia bending.

(2) To moderate the stresses on the big end bolts.

(3) To moderate the mean pressure on the bottom end bearing.

(4) To assist general minimisation of weight for particular purposes, for example, air service, road traction or railway traction.

The means of effecting a reduction of weight are as follows :—

(a) The use of an H section instead of a circular section for the body of the rod ; this is the most effective section for minimising stresses due to inertia bending.

(b) The use of high tensile alloy steels which can safely carry higher stresses than mild steel.

(c) The use of a big end of the split eye type in place of the so-called marine type of big end.

Fig. 200 shews a high speed rod of this type. The small end is fitted with a phosphor-bronze bush pressed into place, and the big end is fitted with steel bushes lined about $\frac{1}{32}''$ thick with whitemetal cast centrifugally. Adhesion of the whitemetal is purely superficial, no dovetailing being used.

In other cases further weight reduction is attained by omitting the big end bushes and running the whitemetal into the rod and cap.

Another variation consists of the use of lead bronze instead of whitemetal, with the object of obtaining an increased length of service before cracks appear on the bearing surface.

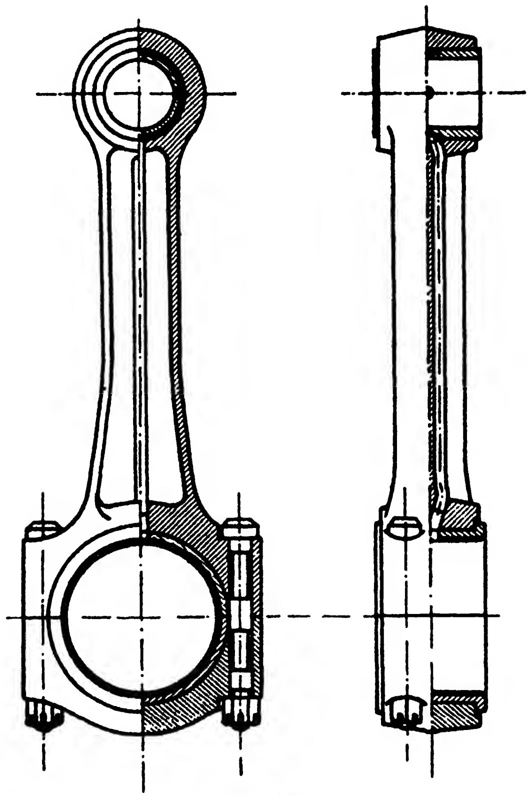
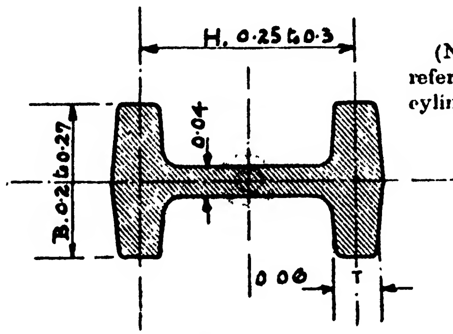


Fig. 200.

For very small engines connecting rods of aluminium alloy have also been used.

Strength of High Speed Connecting Rods.—The proportions of the mean section of H section rods, as used in high speed Diesel Engines (such as motor coach engines), are given in Fig. 201. The sectional area is about $\frac{1}{20}$ to $\frac{1}{30}$ of the piston area and since, in such engines, the maximum pressure is commonly 800 lb./in.^{2} the maximum stress during slow running may be 24,000 lb./in.^{2} or over. In such cases heat treated alloy steels are used.

The formula given on p. 336 for the inertia bending of circular rods can be adapted for use with H section rods by substituting



(NOTE.—Figures refer to fractions of cylinder bore.)

FIG. 201.

78H in place of 26d, where H is the distance in inches between the flange centres of the H section. This can be shewn as follows :—

Let H and O refer to the H section and circular section respectively, then :—

$$\frac{\text{Wt. per unit length H}}{\text{,, ,, ,, ,, O}} = \frac{2BT}{\pi d^2/4} = \frac{8}{\pi} \cdot \frac{BT}{d^2}$$

$$\frac{\text{Modulus of section H}}{\text{,, ,, ,, O}} = \frac{BTH}{\pi d^3/32} = \frac{32}{\pi}\,\frac{BTH}{d^3}$$

$$\frac{\text{Stress H}}{\text{,, O}} = \frac{8}{\pi} \cdot \frac{BT}{d^2} \times \frac{\pi}{32} \cdot \frac{d^3}{BTH} = \frac{d}{4H}$$

The above neglects the weight of the web and slightly underestimates the moment of inertia of the H section. An approximate correction gives stress H/stress O$=d/3H$.

The formula for inertia bending of an H rod is therefore :—

$$f = \frac{\left(\frac{n}{100}\right)^2 . R . L^2}{78H} \qquad \ldots\ldots\ldots\ldots\ (2)$$

As an example consider the following data :—

B=dia. of cylinder	4·5″
S=stroke	6·0″
R=crank radius	3·0″
L=length of rod	12·0″
H=width between centres of flanges . .	1·35″
n=revolutions per minute	3000
Wt. of piston (aluminium alloy) . .	4·5 lb.
Wt. of connecting rod	9·0 ,,
Reciprocating Wt.	7·5 ,,
Piston area	15·9 in.^{2}
Sectional area of rod	0·53 ,,

The acceleration in the crank circle $\div$ g

$$=\left(\frac{2\pi\times 3000}{60}\right)^2\times\frac{3}{12}\times\frac{1}{32\cdot 2}=770$$

Inertia stress at top dead point,

$$=\frac{770\times 7\cdot 5}{0\cdot 53}\times 1\tfrac{1}{4}=13{,}700 \text{ lb./in.}^2$$

Ditto at bottom dead point,

$$=\frac{770\times 7\cdot 5}{0\cdot 53}\times \tfrac{3}{4}=8200 \text{ lb./in.}^2$$

Maximum stress due to inertia bending,

$$=\frac{(30)^2\cdot 3\times 12^2}{78\times 1\cdot 35}=3700 \text{ lb./in.}^2$$

Fig. 202 shews the combined stresses throughout a four stroke cycle. For a two stroke cycle omit the two "idle strokes."

Curve A refers to gas pressure, B inertia of reciprocating parts, C and D inertia bending stress at the following and leading flanges of the H section, respectively. Curve E represents the combined effect of gas pressure and inertia, omitting inertia bending. Curves F and G include inertia bending for the following and leading flanges.

The following conclusion can be drawn for this particular case.

I. Four Stroke Cycle.

(a) The range of stress is not altered by inertia since the maximum compression is reduced to the extent of the maximum tension.

(b) Inertia bending does not increase the range of stress; with a rod of circular section there would be a serious increase of the range of stress.

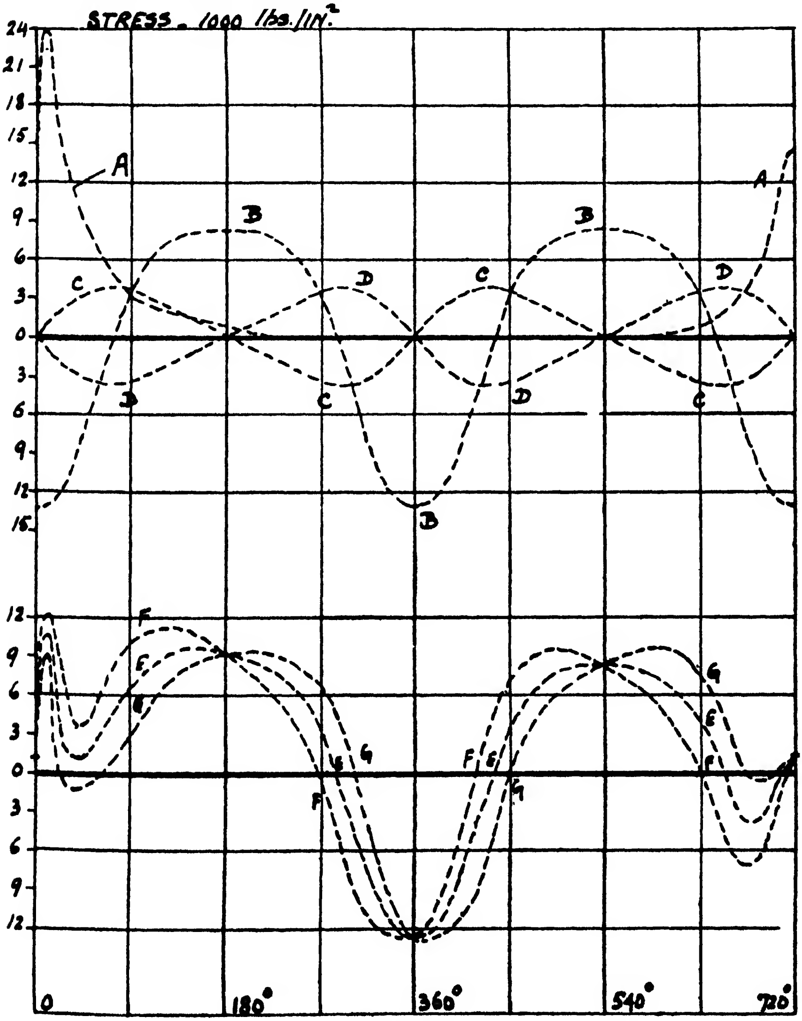


FIG. 202.

As an example consider the following data :—

B=dia. of cylinder	4·5″
S=stroke	6·0″
R=crank radius	3·0″
L=length of rod	12·0″
H=width between centres of flanges	1·35″
n=revolutions per minute	3000
Wt. of piston (aluminium alloy)	4·5 lb.
Wt. of connecting rod	9·0 ,,
Reciprocating Wt.	7·5 ,,
Piston area	15·9 in.²
Sectional area of rod	0·53 ,,

The acceleration in the crank circle $\div$ g

$$=\left(\frac{2\pi\times 3000}{60}\right)^2\times\frac{3}{12}\times\frac{1}{32\cdot 2}=770$$

Inertia stress at top dead point,

$$=\frac{770\times 7\cdot 5}{0\cdot 53}\times 1\tfrac{1}{4}=13{,}700 \text{ lb./in.}^2$$

Ditto at bottom dead point,

$$=\frac{770\times 7\cdot 5}{0\cdot 53}\times\tfrac{3}{4}=8200 \text{ lb./in.}^2$$

Maximum stress due to inertia bending,

$$=\frac{(30)^2\cdot 3\times 12^2}{78\times 1\cdot 35}=3700 \text{ lb./in.}^2$$

Fig. 202 shews the combined stresses throughout a four stroke cycle. For a two stroke cycle omit the two "idle strokes."

Curve A refers to gas pressure, B inertia of reciprocating parts, C and D inertia bending stress at the following and leading flanges of the H section, respectively. Curve E represents the combined effect of gas pressure and inertia, omitting inertia bending. Curves F and G include inertia bending for the following and leading flanges.

The following conclusion can be drawn for this particular case.

I. Four Stroke Cycle.

(a) The range of stress is not altered by inertia since the maximum compression is reduced to the extent of the maximum tension.

(b) Inertia bending does not increase the range of stress; with a rod of circular section there would be a serious increase of the range of stress.

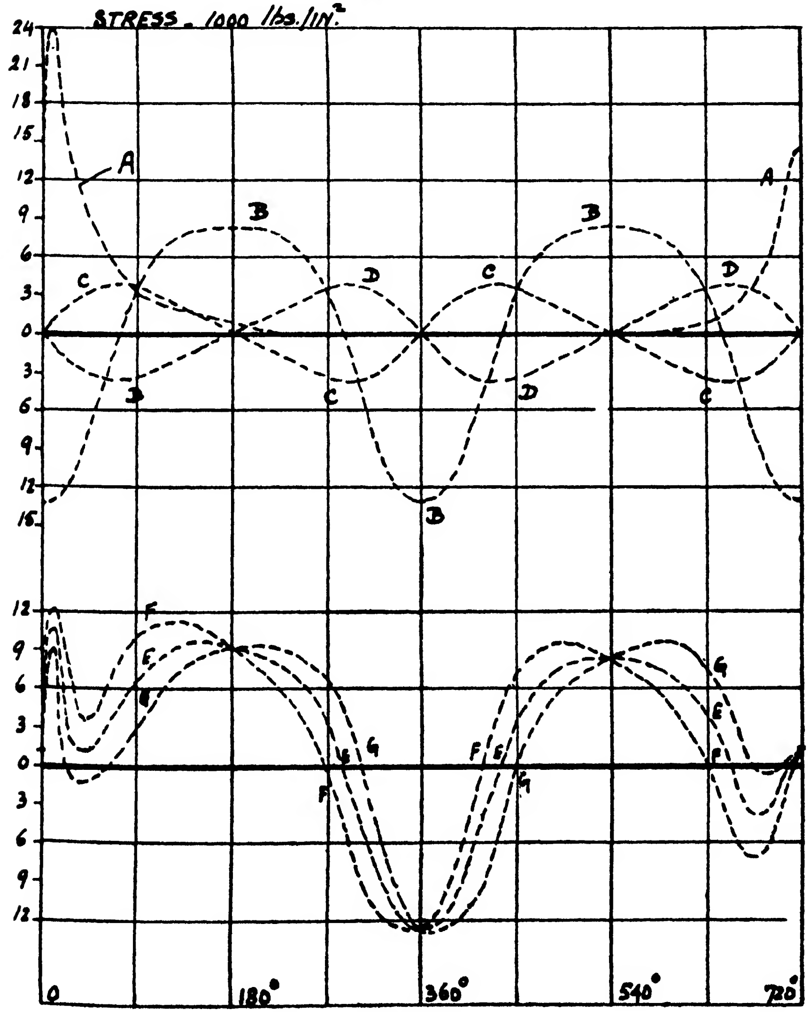


Fig. 202.

II. Two Stroke Cycle.

(a) The range of stress is substantially reduced by inertia.

(b) The range of stress is increased by the inertia bending. With a rod of circular section the increase would be much greater.

Literature.—Low, B. B., " Stresses in Connecting Rods."—*Engineering*, August 14th, 1925.

Kearton, W. J., " The Strength of Forked Connecting Rods." —Br. Assoc. September 17th, 1923 ; see *Engineering*, October 5th, 1923.

CHAPTER XIX

AIR INJECTION

For purposes of description, the complete fuel oil system is conveniently divided into two parts, the first consisting of those elements, such as tanks, etc., which are external to the engine, and the second of those organs of the engine itself which are directly concerned with the delivery of the fuel to the working cylinder.

External Fuel Oil System.—Fig. 203 represents diagrammatically a fuel system for a small Diesel power station and

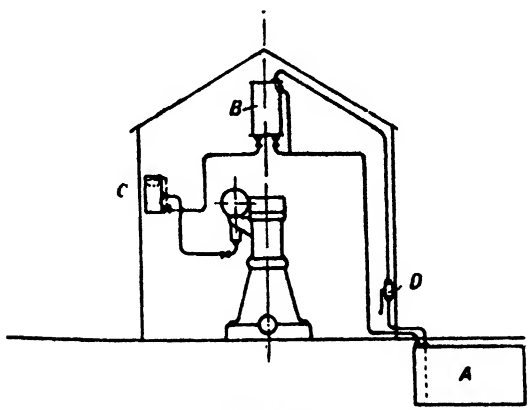


Fig. 203.

consists essentially of a main storage tank A, a ready-use tank B, a filter C, and a pump D, for raising the oil from the storage tank.

The storage tank is preferably arranged underground, as close as possible to the railway siding, so that oil can be run from the railway tank wagon to the storage tank by gravity, through a hose pipe. Some form of level indicator or a plugged hole for a sounding rod should be provided. The capacity of the tank will depend on the size of the station and the local conditions of supply.

The pump D, by means of which the oil is pumped to the ready-use tank, may be of the semi-rotary type, capable of being worked by one man in the case of small stations; but where the daily demand is greater, a motor-driven rotary or reciprocating pump is generally fitted.

The ready-use tank may have a capacity of say half a day's run, so that the routine of replenishing it will occur twice daily. Some form of float indicator should be fitted, so that the level of oil may be conveniently ascertained from the engine room floor. Other necessary fittings are an overflow pipe leading to the storage, or a special drain tank, and a drain valve communicating with the overflow pipe. The tank must be closed at the top to exclude dirt. The valves in connection with the fuel system are preferably of the gate or sluice type, as cocks are liable to leakage and globe valves tend to choke by accumulation of sediment. The tank only requires to be located a few feet above the level of the filter as the discharge is very small.

The filter usually consists of a cylindrical tank of about 40 gallons capacity, located about two feet above the level of the cylinder cover and provided with a filter diaphragm at about a third of the height of the tank from the bottom. The diaphragm consists of a sheet of felt sandwiched between two sheets of wire gauze and reinforced by an angle iron ring.

The fuel enters the filter at the bottom, passes through the diaphragm by virtue of its static head, and is drawn off by the engine fuel pump at a point a few inches above the diaphragm. The filter vessel is prevented from being overflooded by a ball float mechanism which closes the inlet cock when the oil reaches a predetermined level. The plug of this cock is kept fairly tight by means of a spring acting on the plug, but slight leakage is almost inevitable, so it is desirable to mount the filter on an oil-tight tray provided with a drain. It is very usual to provide a small reservoir of the same capacity as the filter arranged alongside the latter for the reception of paraffin, by means of which the piping leading to the engine, and also the fuel pumps and fuel valves, etc., may be cleansed from time to time by running the engine for a few minutes on this fuel before stopping the engine.

Marine installations follow on similar lines with a few complications. The double bottom is used as a storage tank, and the fuel is raised to the ready-use tank by motor-driven pumps, when electric power from auxiliary engines is continuously available or by means of pumps driven off the main engine in cases where the main engine drives its own auxiliaries. In either case it is usual to install duplicate pumps to guard against breakdown. The motion of ships being unfavourable to the successful operation of float devices, the level of the oil in the ready-use tank has to be inferred from gauge glasses, test cocks and the like. For similar reasons, the filters must be totally enclosed and provided in duplicate with change-over cocks, so that they may be overhauled at any time. In addition, special requirements of the Board of Trade and Lloyd's have usually to be complied with.

Fuel System on the Engine itself.—The commoner arrangements fall into one of two broad classes :—

(1) Those in which each cylinder has a separate fuel pump or separate plunger and set of valves to itself. In this case the oil is delivered direct from the pump to the injection valve by the most direct route possible.

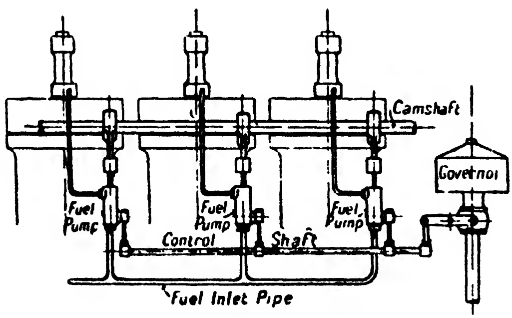


FIG. 204.

(2) Those in which one fuel pump plunger supplies the oil for a plurality of cylinders, usually a maximum of four. In this case the pump delivers to a fuel main provided with a branch and distributing valve separate to each cylinder, whereby the amount of oil delivered to each cylinder may be equalised while the engine is running.

Figs. 204 and 205 illustrate the two systems diagrammatically. It will be noticed that in Fig. 204 the governor operates on all the pumps by means of a shaft extending nearly the whole length of the engine, and as the quality of the governing is dependent on the freedom from friction of the governing mechanism it is desirable to mount this shaft on ball bearings. The expense of providing separate pump bodies and drives is sometimes reduced by grouping the pumps in the neighbourhood of the governor, even to the extent of driving all the plungers by a common eccentric.

With the arrangement of Fig. 205 there is only one pump to regulate, and this renders possible the use of a type of governor which is probably unrivalled for sensibility and which will be described later. The distributors indicated in Fig. 205 are a special feature of this system and are illustrated to a larger

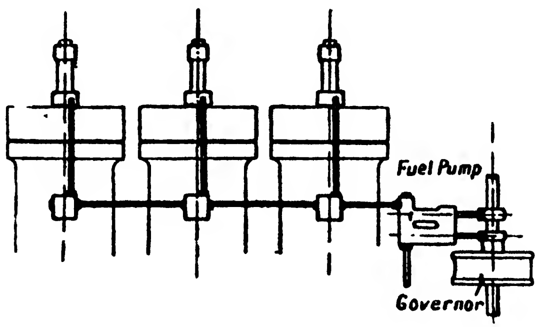


Fig. 205.

scale in Fig. 206. A particularly neat arrangement of piping is obtained by combining the fuel distributor and blast air T-piece in one fitting.

The inclusion of a non-return valve prevents in a great measure the oil being forced back through the pump by the blast air pressure in the interval elapsing between the turning on of the blast air and the attainment by the engine of full working speed. A non-return valve is sometimes fitted to the fuel valve itself for the same reason. Vent cocks are provided on the distributors, and sometimes on the fuel valves, to enable the pipes to be primed before starting the engine.

The priming may be effected in various ways :—

(1) By gravity, means being provided for holding the fuel pump valves off their seats during the process.

(2) By means of an auxiliary hand-operated and spring-returned plunger on the fuel pump.

(3) By means of the fuel pump plunger itself, where provision has been made for disconnecting the latter from its operating eccentric in order to enable it to be operated by a hand lever provided for the purpose.

The piping in connection with the high pressure fuel system deserves special attention, on account of the high pressures used, and the type of union shewn in Fig. 207 is probably the

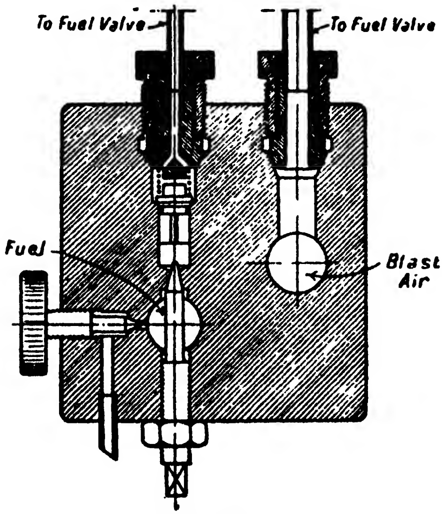


FIG. 206.

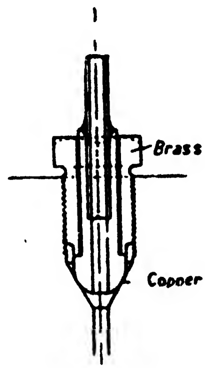


FIG. 207.

most satisfactory that has yet been devised both for oil and high pressure air.

Fuel Pumps.—A simple fuel pump for a large marine engine is shewn in Fig. 208, and is representative of a large class of pumps for both marine and stationary purposes.

The operation of the pump is almost obvious from the figure, but may be described briefly as follows :—

The eccentric A works the plunger B, which is guided at C. E and F are the delivery and suction valves respectively, and the latter communicates with the suction chamber D, to which the fuel is led by means of a pipe not shewn in the figure. M is an auxiliary plunger operated from the crosshead C by links I, H, K, etc., and whose function is to keep the suction valve off its seat for a fraction of the delivery stroke, depending upon how much oil is required per stroke. The duration of this inoperative portion of the stroke is altered as required by raising or depressing the point K by means of an eccentric keyed to the shaft L, according as less or more oil is required. In the case of a governed engine the shaft L is controlled by the governor. On marine engines the shaft L is operated by hand gear, consisting of levers, rods, etc. Neglecting the obliquity of the eccentric rod the main plunger describes simple harmonic motion of amplitude equal to half the stroke of the eccentric, and it will be clear from the drawing that the auxiliary plunger M will describe a similar motion exactly in phase with the first but of amplitude equal to stroke of main plunger $\times \frac{KH}{KI}$.

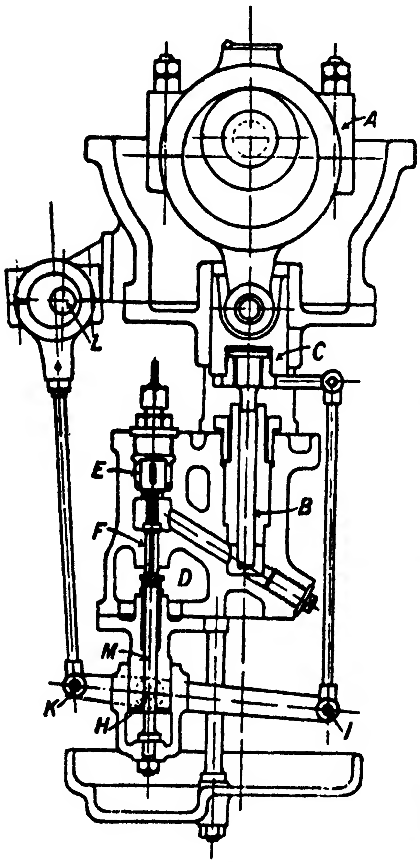


Fig. 208.

When the main plunger is at the bottom of its stroke the auxiliary plunger is also at the lowest point of its travel, and the clearance between the top of the auxiliary plunger and the suction valve multiplied by the ratio $\frac{KI}{KH}$ is equal to the effective stroke of the pump, that is that portion of the stroke during which the suction valve is on its seat, as of course it must be (apart from viscosity effects) for delivery to take place. The quantity of oil delivered per stroke therefore depends on a certain clearance between the auxiliary plunger and the suction valve, which clearance is readily adjusted by shortening or lengthening the rod LK when assembling or adjusting the engine and in the ordinary course of running by the eccentric at L.

Constructional Details.—The pump body, plunger sleeve and guide are of cast iron. The main and auxiliary plungers, the crosshead pin and joints in the linkwork are of case-hardened steel. The valves may be either of steel or cast iron. If the latter, then the suction valve should be fitted with a hardened steel thimble where it makes contact with the auxiliary plunger. The main eccentric and strap may be of cast iron, the lower half of strap being white-metalled in some cases. It will be noticed that no packing is provided for the main plunger, but reliance has been placed on the fit of the plunger. With tar oil this arrangement has given trouble due to scoring of plungers. This was overcome by fitting a gland. A cast tray is provided to catch drips during working and the overflow at priming. A light sleeve encircling the auxiliary plunger is arranged for operation by external gear so that the suction valve may be lifted by an emergency governor in cases of excessive speed, and also by hand in case it is desired to cut any individual cylinder out of operation.

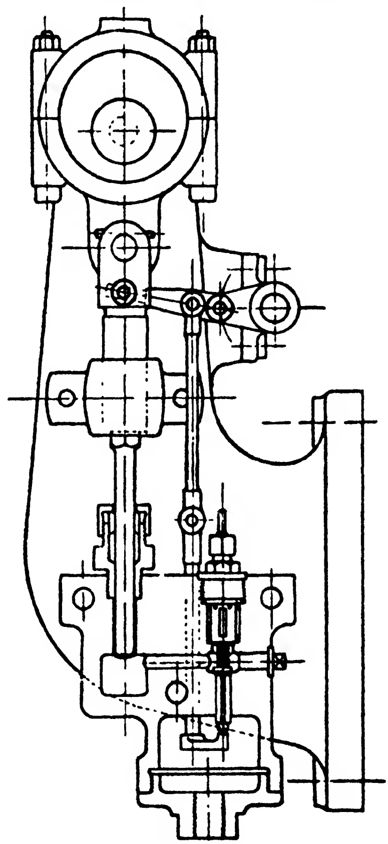


FIG. 209.

Variations of this system, embodying the same principle, are shewn in Figs. 209 and 210. The front view of the latter shews three pumps grouped together, but each worked by its own eccentric. Fig. 211 shews four plungers being operated by eccentrics in common. It is evident that with this arrangement the oil delivered to the pulverisers of the various cylinders will have different allowances of time in which to settle before injection into the cylinder. This appears to have no effect on the efficiency, but it is usual to space the eccentrics so that oil is not in process of delivery whilst a fuel valve is open. The pumps so far illustrated have been driven off the cam-shaft. That shewn in Fig. 212 is arranged with horizontal plungers for driving off a vertical shaft. The auxiliary plunger is driven by a separate eccentric which on account of the intermediate lever L requires to be at 180 degrees or thereabouts to the main eccentric. The suction valve control may in this case be effected in one of two ways.

(1) By an eccentric movement of the lever L.
(2) By advancing or retarding the auxiliary eccentric.

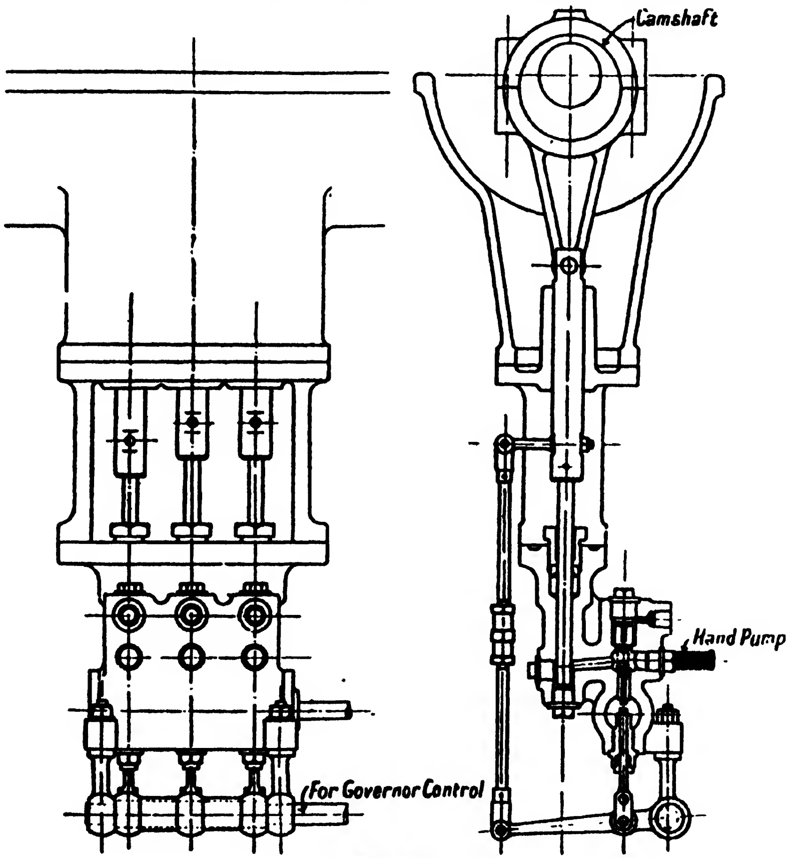


FIG. 210.

The latter leads to a very neat and efficient arrangement of governor and fuel pump, to which reference has already been made. It will be immediately obvious that with a given maximum clearance between the suction valve and the auxiliary plunger, an angular movement of the auxiliary eccentric will have the effect of advancing or retarding the instant at which the suction valve comes on its seat, and consequently increasing or decreasing the amount of oil delivered

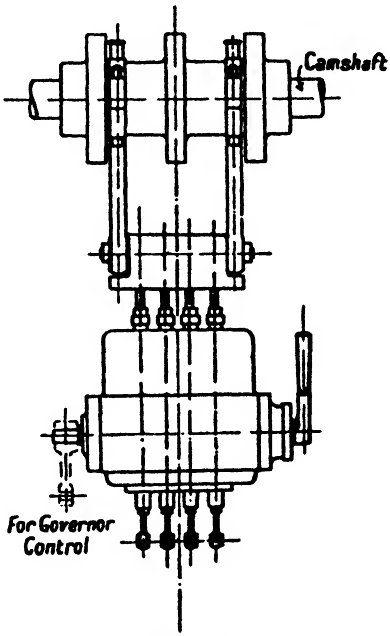


FIG. 211.

per stroke. This angular movement is effected very simply by a type of governor which has been well known for a long time, in steam practice, and which is illustrated in Fig. 221.

Returning to Fig. 212, the use of this type of fuel pump is almost entirely confined to land engines. The provision of an eccentric mounting for lever L enables the pump to be set in three different positions, apart from its normal running position, viz. :—

(1) "Starting." In this position the lever is moved so that the maximum clearance under the suction valve is

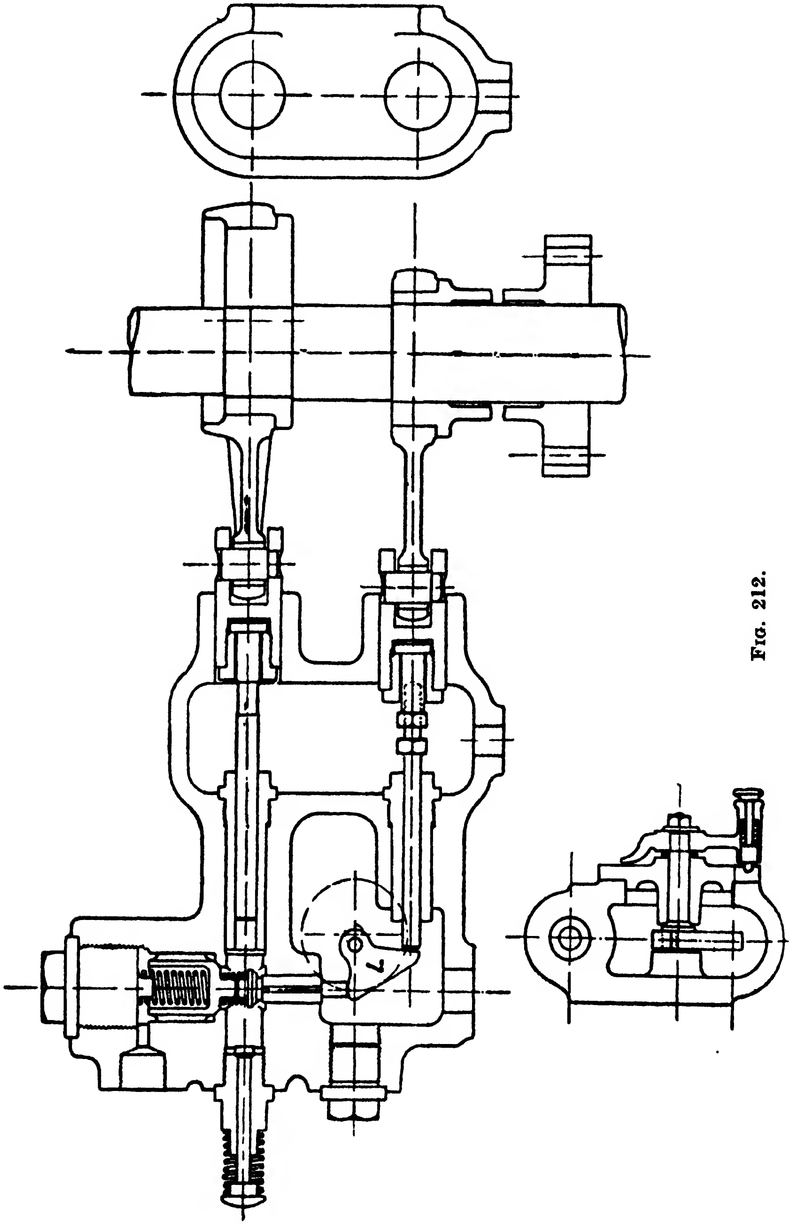


Fig. 212.

increased about 50% so that the delivery of oil per stroke is increased correspondingly.

(2) "Stop." In this position the suction valve is held continuously off its seat and no oil is delivered.

(3) "Priming." In this position both suction and delivery valves are held off their seats and the oil has a clear passage through the pump.

Figs. 214 and 215 will make this matter clear without further explanation.

Details of Fuel Pumps.—The bodies are usually of cast iron, but solid slabs of steel are sometimes used. In designing the body three considerations should be kept in view :—

(1) The shape to be favourable to sound casting.

(2) As few machining operations as possible to be necessary apart from those which can be done on a drilling machine.

(3) The pump chamber and passages to be free from air-locks.

Owing to the costly precautions necessary to ensure the plunger and guide being concentric and in line it is convenient to allow some side play at the point where they join, as in Fig. 216. Some different forms of plunger packing are shewn in Fig. 217 and a selection of suction and delivery valves in Fig. 218. Fig. 219 shews a hand-operated plunger for priming purposes.

Calculations for Fuel Pumps.—The process of computing the capacity of a fuel pump for a proposed engine is most easily illustrated by an example, as follows :—

B.H.P. of one cylinder (four stroke), 250.

One plunger to each cylinder.

Estimated fuel consumption, 0·4 lb. per B.H.P. hour.

Revolutions per minute, 120.

$$\text{Estimated quantity of fuel per cycle} = \frac{0{\cdot}4 \times 250}{60 \times 60} = 0{\cdot}0278 \text{ lb.}$$

Volume occupied by 1 lb. of fuel = about 31 cub. in.

Therefore volume of fuel per cycle $= 0{\cdot}0278 \times 31 = 0{\cdot}86$ in.^{3}

Adding 50% to allow for overload, possible increase of fuel consumption, leakage of plunger, etc. :—

Stroke volume of plunger $= 1{\cdot}5 \times 0{\cdot}86 = 1{\cdot}29$ in.^{3}

Which is satisfied by a plunger diameter of ¾ in. $\times$ 3 in. stroke.

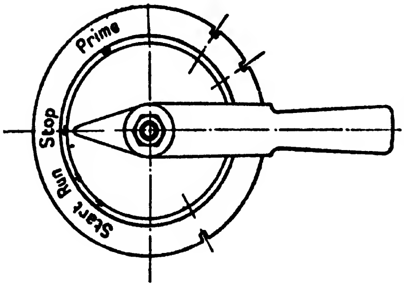


Fig. 214.

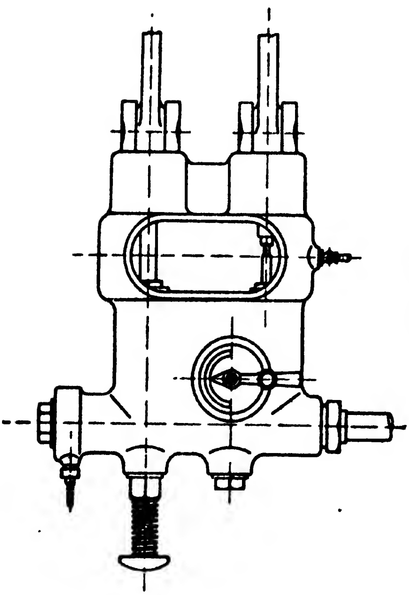


Fig. 213.

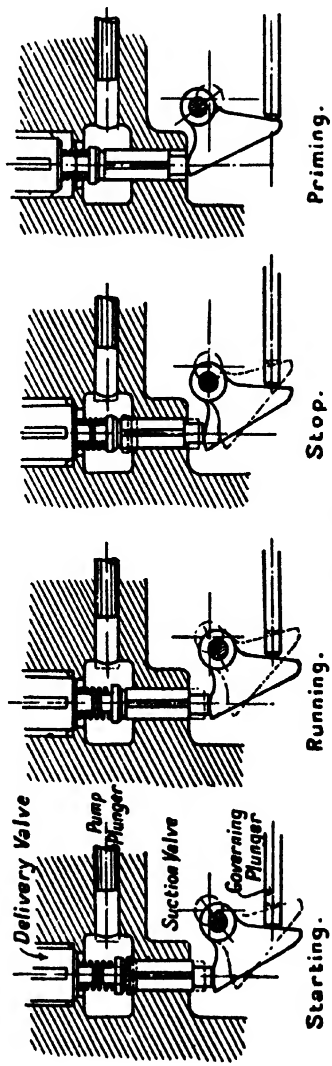


Fig. 215.

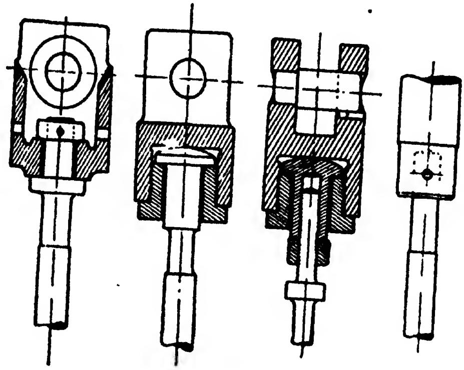


FIG. 216.

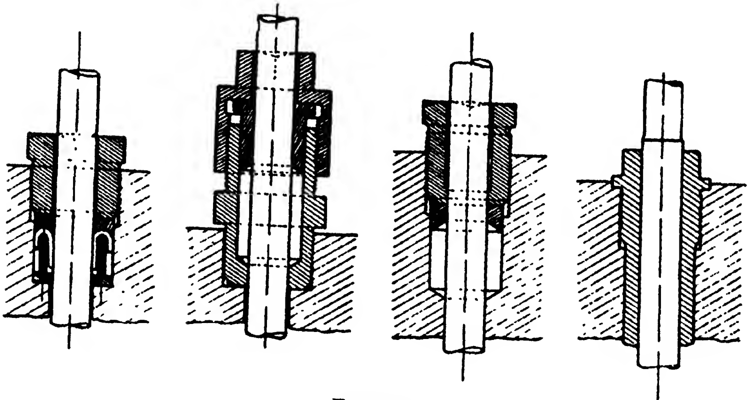


FIG. 217.

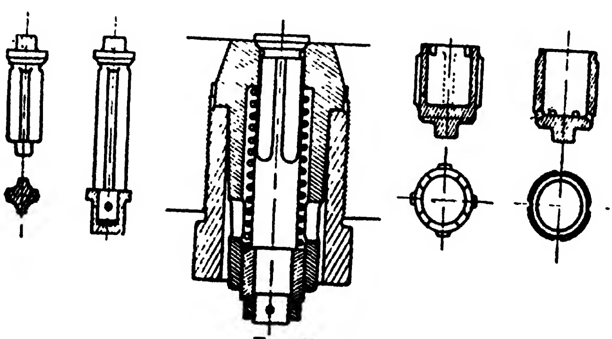


FIG. 218.

This size of plunger would only be permissible on a marine engine. If the cylinder belonged to a governed engine the stroke volume of the fuel pump plunger would need to be about four times the above figure, as it is found that good governing at all loads is only to be obtained by using about the last quarter of the stroke. This is probably due to the fact

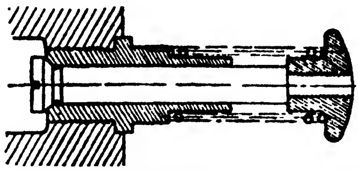


FIG. 219.

that the quantity of fuel consumed by the engine in a given time is not proportional to the load but more nearly proportional to the load plus a constant representing the engine friction. The actual position taken up by the governor and the effective stroke of the pump plunger at any specified proportion of full load are not easy to determine experimentally with great accuracy, but the angular positions indicated in Fig. 220, with reference to the fuel pump eccentric circle, are those generally used as the basis of calculation.

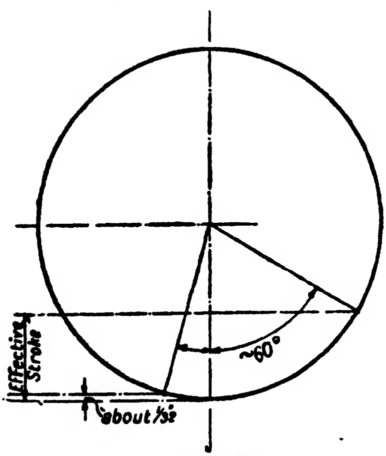


FIG. 220.

When one plunger is used to supply several cylinders the length of effective discharge period is limited by the condition that the latter should not overlap the injection periods. In estimating the capacity of a fuel pump driven off a vertical shaft the speed of the latter must be kept in mind, being usually the same speed as the engine, and in some cases 50% more.

The valves, hand plungers, etc., are suitable subjects for distributive standardisation. For example, a suction valve $\frac{5}{8}$ in. in diameter would be quite suitable for all sizes of cylinder (assuming one plunger per cylinder) up to about 20 in. bore provided that the use of fuels of exceptional viscosity were not contemplated. For oils like crude Mexican, of the consistency (when cold) of tar, larger valves are probably advisable. With the valve arrangements in common use the diameter of the delivery valve is determined by that of the head of the suction valve plus adequate clearance for the flow of the oil round the latter.

The general thickness of metal of the pump body is usually kept as uniform as possible to facilitate casting, and the nominal stress in the neighbourhood of the pump chamber based on a blast pressure of 1000 lb. per sq. in. is about 2500 lb. per sq. in. The design of a fuel pump affords ample scope for a draftsman's skill in many directions, in which numerical calculation plays a very small part, and the following suggestions are offered :—

(1) The arrangement generally to be neat and substantial and presenting an external appearance in keeping with its surroundings.

(2) The flanges and brackets by which it is secured to the framework of the engine to be unobtrusive and to have the appearance of growing as naturally as possible out of the main body of the casing, so as to convey an impression of rigidity and equilibrium.

(3) Every detail to be carefully studied, both with regard to its special function and also to economy in manufacture, efficiency always having precedence over economy. In particular, case-hardening and bushing of parts subject to wear must not be stinted, and provision should be made for lubrication of all moving parts.

(4) Valves and other internal mechanism to be easily accessible for inspection and overhaul.

(5) Arrangements to be made to catch all drips, both of fuel and lubricating oil, avoiding the use of trumpery sheet-iron guards and the like.

Many of the above principles apply of course to the design of any part of any high-grade machine, and they are mentioned here because the matter on hand provides an excellent opportunity of emphasising them in a particular case, in which the subject is singularly free from the complications arising from

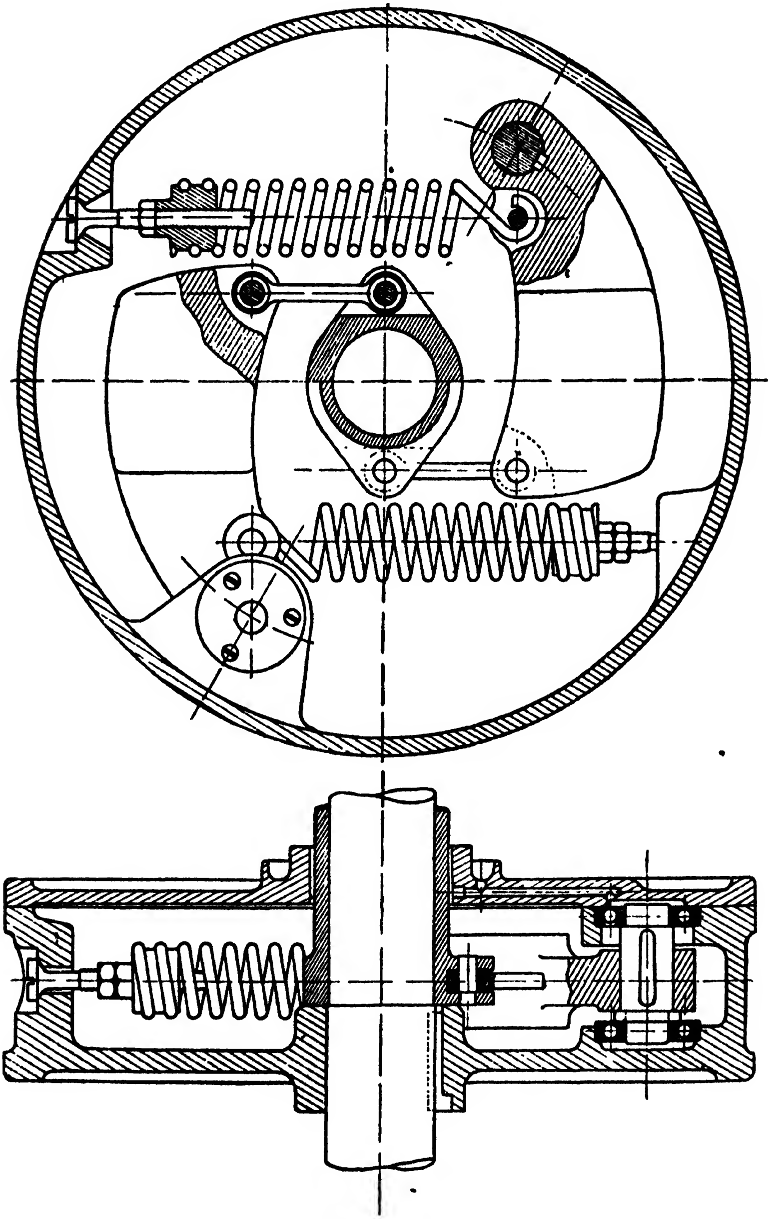


FIG. 221.

calculations. When the discussion is transferred to some large part of a machine, in which the stresses are approximately determinate and the scope of the design appears to be limited by adjacent parts, it becomes increasingly difficult to reconcile the ideals of high-class design with the requirements of efficiency and economy and the skill of a designer may be gauged by the extent to which this difficulty is overcome. From this point of view no part of a design can be said to be finally determined until the whole design is complete, as there is always the possibility that a design for a certain part, perfect in itself, may require to be modified subsequently on account of its relationship, perhaps remote, to some other part as yet undetermined.

Governors.—It is not proposed to deal here with governor design generally, as that is a subject for a specialist in this particular department of mechanical design, but only to illustrate the application of governors to stationary Diesel Engines by means of a few examples, and to give the main lines of calculation for the type of governor shewn in Fig. 221, which is a type not usually standardised by governor specialists. The action of the weights in causing rotation of the central sleeve will be immediately obvious from the figure. The amount of this rotation between the limits of no load and full load should correspond with the angle $\sim 60°$ of Fig. 220, but as a safety precaution it is advisable to give the governor sufficient range to give a complete cut-out, and the sleeve should therefore be free to describe an angle of about 70°. The first stage in the design of the governor is to rough out a drawing similar to Fig. 221, fulfilling all the requirements as to space, accessibility, etc., and in which the above angular rotation is secured. As regards the size of the governor, it is generally wise to avail oneself of all the space obtainable. The next step is to find the mass and centre of gravity of the weights and the positions of the latter in the extreme in-and-out positions. A diagram similar to Fig. 222 should now be constructed, in which the abscissæ are distances of the weight from the centre of the shaft in inches and the ordinates are the centrifugal forces at these distances at no load speed and full load speed respectively. Point " P " corresponds to " no load " speed and " no load " distance from centre, and point " Q " the same quantities for full load. The line PQ then determines the properties which the controlling spring would have to possess if it were connected to the weight at its centre of gravity. These properties are as follows :—

(1) The initial tension, when the weights are full in, is equal to the centrifugal force corresponding to the point "Q."

(2) The weight of the spring per inch extension is equal to the slope of the line PQ, that is the amount in lb. by which the ordinate increases as the abscissa increases by one inch.

Actually the spring is attached to the weight at a point nearer to the fulcrum than the centre of gravity, and both the initial tension and the rate as found thus require to be increased in the ratio $\frac{k}{l}$, where :—

k=Distance of the weight fulcrum from the line joining the C.G. of the weight to the centre of the governor.

l=Distance of the weight fulcrum from the line of action of the spring.

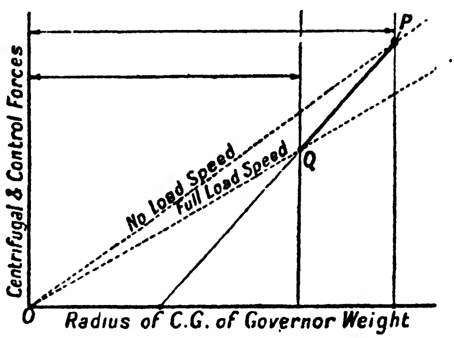


FIG. 222.

This very simple construction, repeated as often as may be necessary in the process of trial and error, contains all the dynamical calculation required to ensure sensibility and stability, but it is advisable to provide adjustments for spring tension, in the manner shewn in the figure, to allow for unavoidable errors and routine adjustment on the test-bed. Strictly speaking, the diagram shewn in Fig. 222 should be corrected to allow for the versed sine of the arcs described by the points of suspension, and so on ; but these are practically negligible. Other types of governor are designed on similar lines, but are usually complicated by link mechanism, of which the variations of configuration are not negligible.

Variation of speed during the running of the engine is readily secured by transferring a part of the controlling force to an auxiliary spring, the tension of which can be varied by mechanism provided for the purpose, as shewn in Fig. 223,

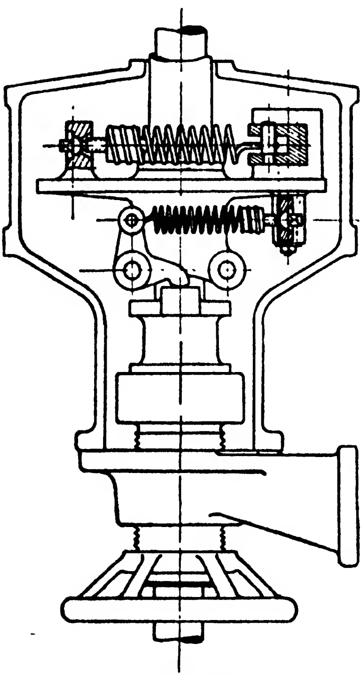


Fig. 223.

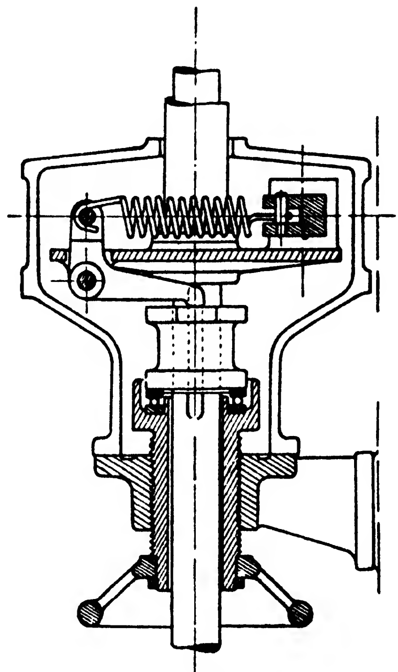


Fig. 224.

or by varying the tension of the main spring itself, as in Fig. 224.

Some points to be observed in governor design are :—

(1) Weights as heavy as possible, to give power and consequently render the effects of friction negligible.
(2) Springs to be readily adjustable.
(3) Small pin-links, etc., to be as substantial as conveniently possible.
(4) Friction to be reduced to a minimum by ball-bearings.

(5) Joints other than ball-bearings to be bushed and provided with well-hardened pins.
(6) Lubrication, both as regards supply of lubricant to the working parts and systematic disposal of the surplus, to be considered carefully.

The general disposition of the governor and fuel pump with respect to the framework of the engine is shewn in Figs. 225, 226 and 227, in three cases. In Fig. 225 the governor is of the angular movement type described above and illustrated in Fig. 221, driven off the vertical shaft from which the cam-shaft

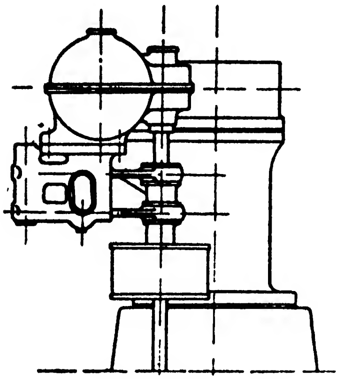


Fig. 225.

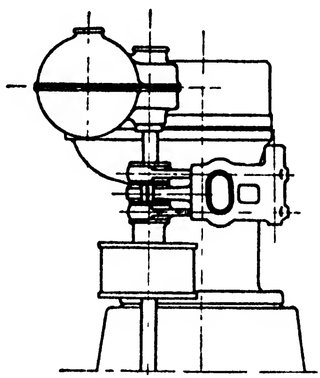


Fig. 226.

receives its motion. The fuel pump is of the horizontal plunger type, receiving its motion from eccentrics mounted on the same vertical shaft. The fuel pump body is supported by a facing on the lower side of the case containing the upper spiral gears.

In the arrangement shewn in Fig. 226 the fuel pump is attached to the cylinder jacket, but in other respects the details are similar to those of Fig. 225.

The governor shewn in Fig. 227 is of the more usual type characterised by a sleeve which is mounted on a feather and which rises as the engine's speed increases. The pump is of the vertical multi-plunger type, and regulation is effected by rotation of an eccentric shaft, on which are hinged the levers which operate the auxiliary plungers.

The arrangements described briefly cover the bulk of the fuel pump and governor mechanisms found in practice, but mention must be made of some later refinements which have sometimes been used.

(1) Control of fuel valve opening. At light loads the duration of opening of the fuel valve is greater than necessary if uncontrolled and the instant of opening

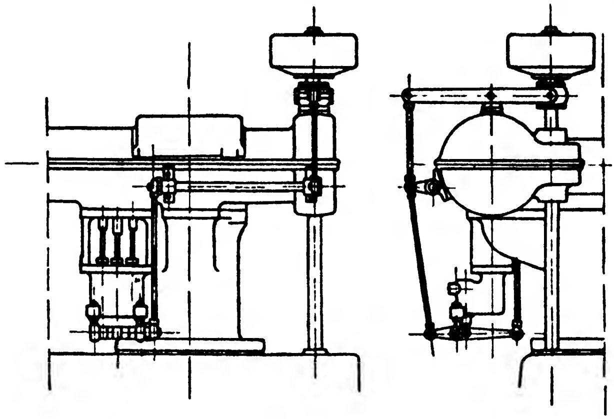


FIG. 227.

which is most favourable for full load running is inclined to be late for light load running. At least one firm has attacked this problem of governor control of the fuel valve operating mechanism.

(2) Blast pressure control. This question is closely allied with (1), as a shortened opening period would lead to increase of the blast pressure if the latter were not corrected. Apart from this the blast pressure has in any case to be altered in accordance with the load (unless cylinders are cut out of operation) if good combustion is to be secured at all loads, including no load. The blast pressure is placed under the control of the governor by means of a throttle slide on the compressor suction.

(3) Pilot ignition. This refers to cases where exceptionally refractory oils are being used which require for their combustion a preliminary charge of a lighter oil, such as Texas oil, which is deposited in the pulveriser in advance of the main charge by a small auxiliary pump provided for the purpose. The necessity for this device appears likely to be obviated by improvements in fuel valves and combustion chambers.

Fuel Injection Valves.—It now remains to deal with the valve by means of which the fuel is injected into the combustion space and to which oil is delivered by the fuel pump for this purpose. These may be broadly classified as the open and closed types respectively, and as the former form a relatively small class at present it is convenient to dispose of them first.

Open Type Fuel Valves.—Fig. 228 is a diagrammatic view of such a valve, omitting all detail not required to illustrate the bare principle. Oil is delivered to the space A past the non-return valve B by means of the fuel pump, and this type of fuel valve derives its name from the fact that the space A is in

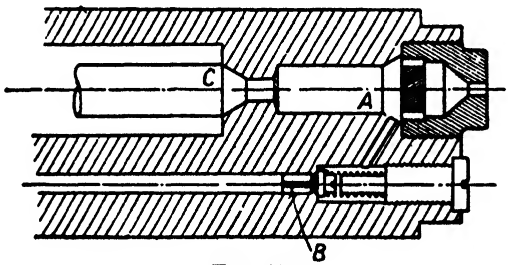


Fig. 228.

constant communication with the interior of the cylinder. It is to be noticed that the fuel pump is not required to deliver against the pressure of the blast air as the latter is restrained by valve C. The latter is opened by appropriate gear at the predetermined instant for injection and carries with it the fuel oil contained in the space A. The action appears to be highly efficient in pulverising effect and excellent fuel consumptions have been reported for engines in which these valves have been fitted. This type of fuel valve appears to have been devised in the first instance for use in horizontal engines in which it was anticipated that the more usual type of fuel valve would be at a disadvantage. A valve working on a somewhat similar principle has been tried, from all accounts successfully, on vertical engines, but has not yet, to the author's knowledge, become a standardised fitting.

Closed Type of Fuel Valves.—In this type communication between the combustion space and the interior of the fuel valve only exists during the injection period, when the flow is always in the same direction, apart from such derangements as stuck valves or failure of the blast pressure.

Fig. 229 shews what may not improperly be called the Augsburg type of fuel valve. Apart from the cast-iron body, the construction of which is sufficiently illustrated by the drawing, the principal parts are :—

(1) Needle valve A.
(2) Spring B.
(3) Stuffing box C.
(4) Pulveriser D.
(5) Flame plate E.

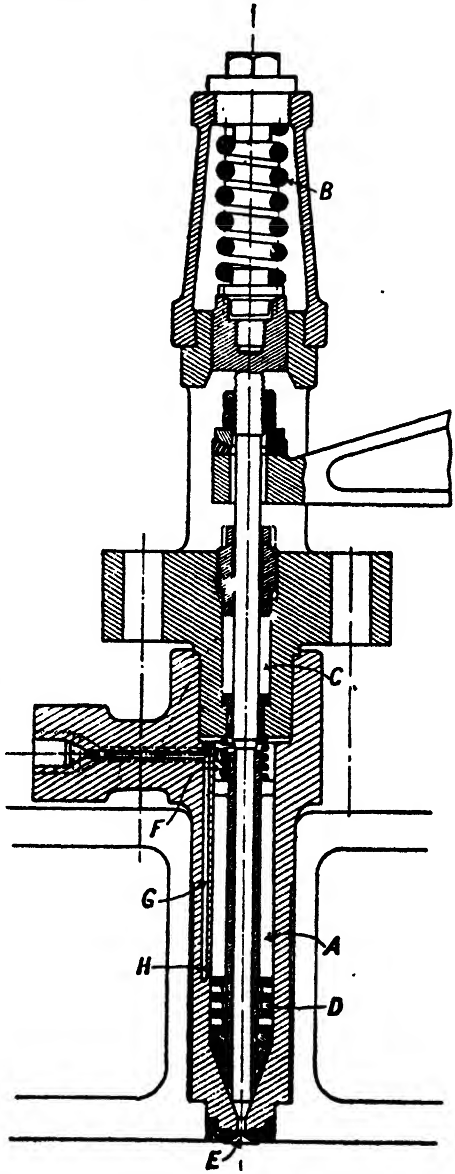


Fig. 229.

The needle valve is usually made of special steel, case-hardened in way of the stuffiing box to prevent cutting by the packing. Accurate alignment of all parts of the needle is essential and readily secured by grinding between fixed centres. The lower part of the needle is preferably reduced in diameter by a few thousandths, as a certain temperature gradient exists between the needle and the pulveriser tube which may lead to seizure if sufficient clearance is not allowed. The tip generally has an angle of about 40 degrees. The needle spring, in addition to returning the needle to its seat against the pressure of the blast air, has to deal with the friction of the stuffing box, and may be figured out on the basis of a pressure of 1500 lb. per sq. in. over the sectional area of the needle at the stuffing box. The latter is usually provided with a screwed gland.

The pulveriser tube is held on its seat by a stiff spring, and serves the double purpose of affording some support to the needle and retaining in their relative positions the rings and the cone which play an important part in pulverising the fuel. It will be clear from the figure that the pulveriser is

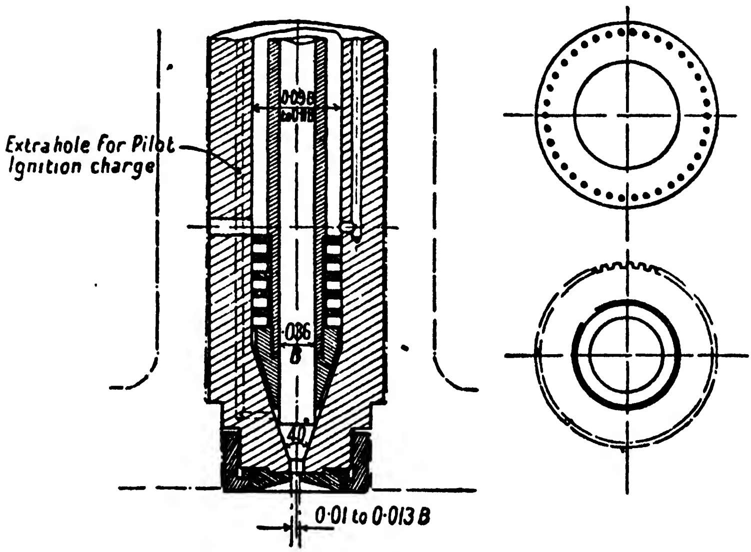


Fig. 230.

surrounded by blast air which enters at F. The fuel is introduced by means of a narrow hole G, at a point H immediately above the top ring. If the point H is located too high the oil fails to distribute itself evenly round the pulveriser rings and inefficient combustion results.

Fig. 230 shews the injection end of the pulveriser, together with the flame-plate and nut, to a larger scale. The details shewn are those in most common use, but are subject to variation in the practices of different manufacturers. The proportions shewn are roughly indicative of good practice, but it must be admitted that the rule of linear proportionality does not appear to be rational in this case. Experience in this matter discloses two facts :—

(1) That for a given engine there is a certain minimum diameter of pulveriser ring, below which results are not satisfactory (about 9% of the cylinder bore).

(2) That as cylinders are increased in size it becomes increasingly difficult to obtain a high M.I.P.

These suggest the following hypothesis :—

That the best results are to be obtained when the depth of oil in the pulveriser before injection is a certain amount, and the same for cylinders of all sizes. If this is true, then the area of the pulveriser ring should be in proportion to the cylinder volume. This would lead to the diameter of pulveriser rings being made proportional to the cylinder bore raised to the power of 1·5. Such a rule has not been adopted, and would probably lead to inconveniently large fuel valves in the larger sizes of engines. A very large number of different types of pulveriser are in use, and have been described in the technical press; but it still remains to be proved that they are more efficient than the common variety shewn in Fig. 230. A neat form of pulveriser tube, which dispenses with the long narrow hole drilled in the fuel valve casting, is shewn in Fig. 231, from which it will be seen that the oil is led to an angular space A at the top of the tube, whence it flows downwards to the pulveriser rings via a number of grooves in the surface of the pulveriser tube. Holes B are provided to give passage for the blast air.

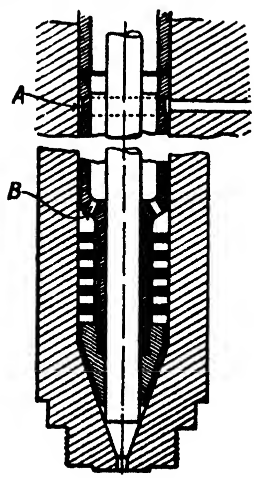


Fig. 231.

Swedish Type of Fuel Valve.—Fig. 232 shews the construction of this type of valve, which has also been widely adopted and which is characterised by the fact that the needle is completely enclosed within the casing and is subject on all sides except the extreme tip to the pressure of the blast air. On this account the spring A does not require to be as strong as that of an Augsburg type of valve of the same size. The needle is lifted in working by the lever B attached to a cross-shaft C, the end of which penetrates the casing through a stuffing box D. The

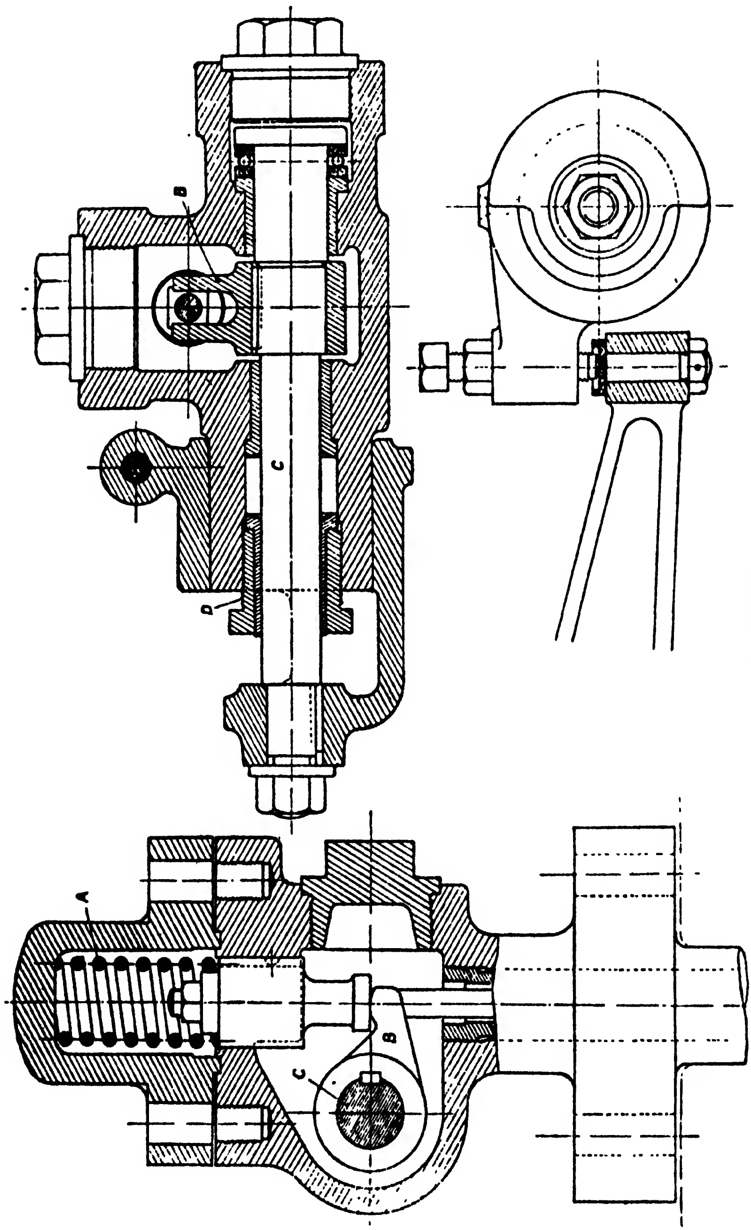


FIG. 232.

mechanical means by which end thrust on the cross-shaft and bending actions on the overhung end, due to the pressure on the external lever, are dealt with, will be clear from the figure without further explanation. The use of this type of valve appears to be limited at present to those designs in which the requirements of other parts of the valve gear necessitate the fuel valve operating lever being arranged off the centre line of the cylinder cover.

Burmeister Fuel Valve.—The construction of this valve is shewn diagrammatically in Fig. 233, and its outstanding features are the use of a mushroom valve, the extreme simplicity of the whole arrangement, and the fact that the valve is opened by a downward movement. The latter is a particularly valuable feature as it secures uniformity of valve gear and ease of withdrawal.

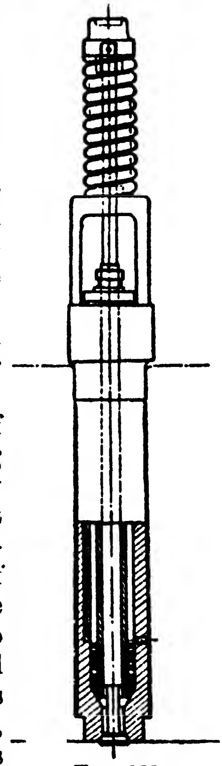


FIG. 233.

The four classes of fuel valve described above include as members most of the blast air fuel valves used on Diesel Engines at present. Each type has its advantages, but no one of them can be said to hold the field. Something similar might be said for the enormous variety of pulverisers patented and in actual use. It seems doubtful if any of these can claim outstanding efficiency. When pilot injection of a less refractory oil is used to facilitate the use of tar oil as fuel an additional hole has to be drilled in the fuel valve, as shewn dotted in Fig. 230. The question of burning tar oil is still in the experimental stage in this country, but the results so far obtained hold out hopes that it may be possible to dispense with pilot ignition in favour of special arrangements of a simpler character in connection with the fuel valve details.

Some of the arrangements by means of which fuel valves are operated are shewn in Figs. 234, 235, 236, and 237. The long lever, which is a feature of all these schemes, is usually of cast steel, and should be of stiff construction. The fulcrum on which the lever hinges is common to the levers which operate the other valves, viz. air and exhaust valves in the case of four stroke engines and scavenge valves in the case of two stroke engines, and starting valves in both cases. With land engines, and many marine engines, it is usual to mount the fuel and starting valve levers on eccentric bushes mounted on the fulcrum shaft at such angles that the operation of putting the starting valve into gear automatically puts the fuel valve out

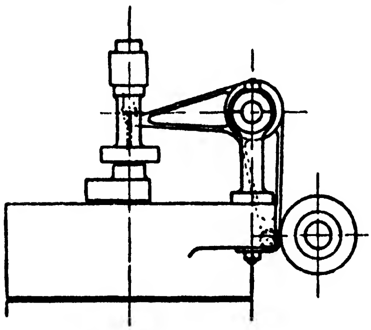


FIG. 234.

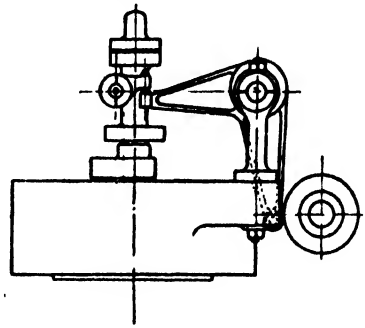


FIG. 235.

of gear and vice versa. This is considered in detail in Chapter XXII.

The use of the needle type of fuel valve in conjunction with a single lever necessitates the latter being so disposed that its roller is rendered more or less inaccessible by the cam-shaft,

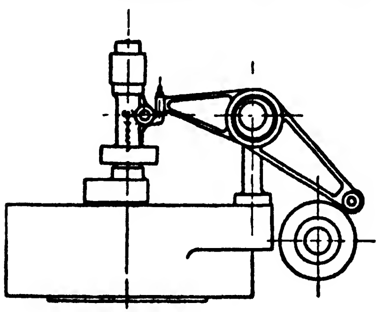


FIG. 236.

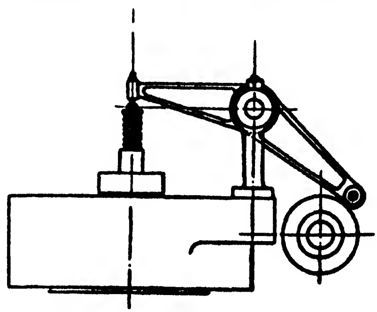


FIG. 237.

particularly if the latter runs in a trough (see Fig. 235). The difficulty may be got over by providing a small intermediate lever, as shewn in Fig. 236, to reverse the direction of motion. In spite of the objections which have been raised against this arrangement it appears to be satisfactory in practice.

Design of Fuel Valves.—An approximate rule for the internal diameter of the body has already been given, being the same as the diameter of the pulveriser rings. The thickness of the walls (cast iron) may be from a third in large valves to a half in the case of small valves of the internal diameter. If the valve is of the Swedish type this thickness will be approximately constant throughout the body of the valve, except in the neighbourhood of flanges, etc. If of the Augsburg type, those parts of the body not subject to pressure may be a little thinner. In all cases a good rigid job should be aimed at, as lack of alignment leads to sticking of the valve. The pulveriser tube is made of steel and the details, such as rings and cones, of steel or cast iron. The Swedish type of valve requires special care to be devoted to the design of the cross-shaft and its fittings, in order to obtain freedom under load, adequate bearing surface and accessibility of the stuffing box. As regards the valve as a whole, the designer should aim at shapely solidity and avoid flimsiness of detail.

With the Augsburg type of valve (Fig. 229) the load necessary to lift the needle is the spring load less the product of the blast pressure and the area of the needle at the stuffing box (approx.) plus the gland friction. With the Swedish type (Fig. 232) the load may be taken as approximately equal to the spring pressure plus the product of the blast pressure and the area of the needle at its seat. This load evidently induces bending and twisting actions, which the cross-shaft should be proportioned to carry with a low stress. The weakest section is generally at the reduced diameter to which the external lever is keyed. The key itself should be amply proportioned, and is preferably made of tool steel. The ball thrust must be proportioned to the load obtained by the product of the maximum blast pressure into the sectional area of the cross-shaft at the stuffing box. The flame plate is of nickel steel and the diameter of the hole is usually about 1% of the cylinder bore, but the best size for any particular case must be found by experiment. The flame plate nut may be of steel or bronze secured to the fuel valve body by a fine thread and provided with flats to accommodate a spanner.

The main points in the design of fuel valves may be summarised as follows :—

(1) Rigidity and alignment of casing.
(2) Alignment of the needle and its guide.
(3) Freedom of all working parts.
(4) Sturdy proportions for all small details.

Literature.—Chaloner, J. L., "Tar Oils as Fuel for Diesel Engines."—*Internal Combustion Engineering*, July 15th, 1914, *et seq.*

Chaloner, J. L., "Oil Engine Progress."—Diesel Engine Users' Assoc., Feb. 24th, 1921.

Day, C., "Tars and Tar Oils as Fuel for Diesel Engines."—Diesel Engine Users' Assoc., Jan. 19th, 1916.

Hawkes, C. J., "Injection and Combustion of Oil in Diesel Engines."—N.E. Coast Inst. of Engineers and Shipbuilders.

Hawkes, C. J., "Some Experimental Work in connection with Diesel Engines."—Inst. Nav. Arch., July 8th, 1920.

Hawkes, C. J., "Fuel Oil in Diesel Engines."—N.E. Coast Inst. of Engs. and Shipbuilders, Nov. 26th, 1920.

Porter, G., "Tar Oil Fuel and Diesel Engines."—Diesel Eng. Users' Assoc., May 24th, 1917.

Smith, P. H., "Two Essential Conditions for burning Tar Oil in Diesel Engines."—Diesel Eng. Users' Assoc., May 16th, 1918.

CHAPTER XX

AIRLESS FUEL INJECTION

Mechanical Injection.—The various kinds of Diesel type engines, known as solid injection, airless injection, or cold starting heavy oil engines, which share the characteristic Diesel feature of spontaneous ignition but differ from true Diesel Engines inasmuch as the fuel is injected by mechanical means instead of by an air blast, owe their origin to two distinct lines of development.

In one line of development firms already experienced in the manufacture of the true Diesel have sought to eliminate the air compressor for injection purposes with a view to simplification and reduction of cost. In these cases the traditional arrangement of valves and a concave piston top is usually retained, but the compression is usually reduced to 400 to 500 lb. per square inch, and the indicator card reveals a certain amount of combustion at constant volume so that the maximum pressure attains a value of 600 lb./in.^{2} and upwards. The use of compressed air for starting purposes is retained.

In the second line of development the evolution can be traced from the earlier hot bulb engines of the four and two stroke types. In the search for economy and ability to use heavier oils, the compression has been gradually raised until at last the use of externally heated surfaces can be dispensed with. At this stage the compression pressure and the indicator diagram generally correspond with those attained as indicated above. Such engines were usually of the horizontal type in the case of four stroke engines with the inlet valve arranged over the exhaust valve, and the combustion chamber in the form of a compact pocket (Fig. 238).

There are now a number of makes of such engines both of horizontal and vertical types on the market. The characteristic arrangement of combustion chamber and valves is retained in the vertical engines also in some cases (Fig. 239).

The question whether the adoption of mechanical injection actually does achieve simplification and reduction of cost as compared with air blast injection was for a long time doubtful. Given equal sizes of cylinders in the two cases it seemed clear that the mechanical injection engine must be cheaper to build, even after allowance is made for higher maximum cylinder pressures. On the other hand, experience showed that the

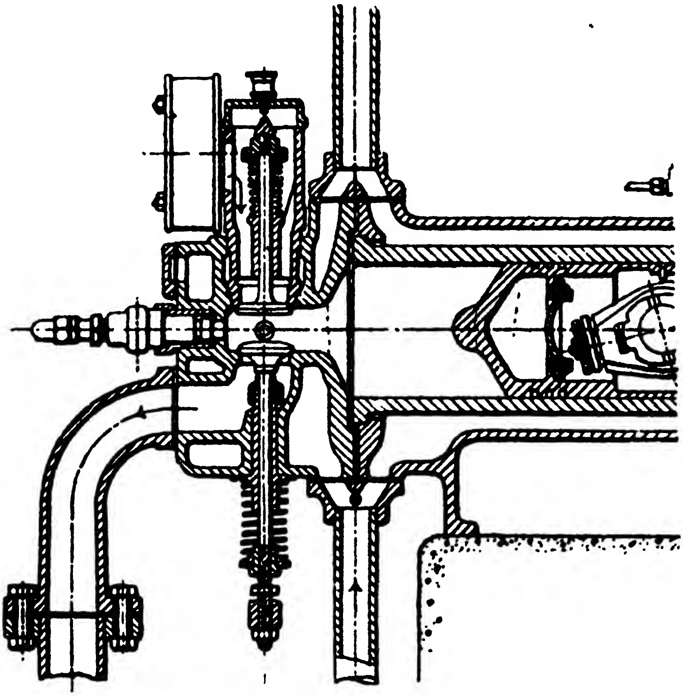


FIG. 238.

air blast engine would achieve the higher brake mean pressure without smoke in spite of its mechanical efficiency being lower by about 5 or 6%. It then became a question which engine could safely maintain the higher brake mean pressure under service conditions and this experience alone could decide.

Technical improvements in airless injection apparatus and adjustment have shewn the airless injection engine to be definitely superior to the air blast injection engine when using most of the Diesel fuels available to-day, i.e. fuels of rather low viscosity. The difference in favour of the airless injection engine is particularly marked at light loads.

Airless injection is now universally used for small engines, is rapidly displacing blast air injection for engines of medium and large power, and is in fact used on the largest Diesel Engine in existence to-day (22,000 B.H.P. in 8 cylinders).

The Combustion Chamber.—The ability of the air blast engine to carry a higher brake mean pressure (and, therefore,

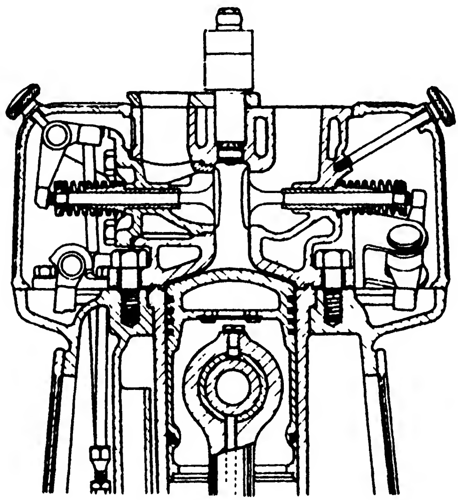


FIG. 239.

a fortiori, a higher M.I.P.) than the mechanical injection, is traceable to two causes :—

(1) The supercharging effect of the blast air which increases the available oxygen by about 5 to 10%.

(2) The action of the air blast in promoting rapid combustion (turbulence).

In the mechanical injection engine the turbulence is dependent upon the air speed through the inlet valve (about 150 ft./sec.), and the effect produced by the piston displacing the charge into a pocket shaped combustion space when this construction is adopted.

Any attempt to increase the air inlet speed unduly would react unfavourably on the pumping loss and reduce the charge weight. In order to obtain satisfactory combustion with the limited turbulence available it is necessary to get the fuel in quickly and early in order to gain time for combustion and to attain a high temperature as quickly as possible. This means a period of combustion at constant volume with consequent rise of pressure at the dead centre. As very high maximum pressures are undesirable the compression pressure is accordingly lowered to the lowest point which will secure certain ignition under the circumstances contemplated. A compression pressure of about 400 lb./in.^{2} seems to be about the lower limit in practice unless preheating of the jackets can be

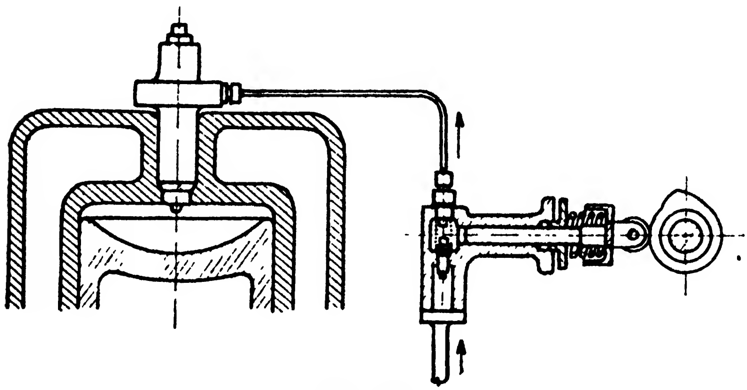


FIG. 240.

resorted to. When starting has to be effected with cold jackets on heavy oils 450 lb./in.^{2} appears to be desirable.

Mechanical Injection Systems.—The existing systems can be broadly divided into two classes. In the first class a fuel pump is provided for each cylinder and the injection period corresponds to the delivery period of the pump, which is frequently cam operated (Fig. 240). The fuel valve is entirely automatic in action and consists either of a simple capillary orifice or more usually of a spring loaded needle valve which is lifted by the pressure of the fuel acting on an unbalanced area.

In the second class the fuel valves are mechanically operated in much the same manner as with the air blast system. Fuel is supplied to them from a common main or reservoir maintained at a sensibly constant pressure by a pump of one or more plungers (Fig. 241). This system is sometimes referred to as the rail system. The pressure in the rail or reservoir is maintained practically constant at a value of 2000 to 8000 lb./in.^{2} by providing sufficient volume capacity to absorb the fluctuations in demand and supply by the elasticity of the fuel itself. In earlier designs special arrangements of accumulators or resilient members were provided, but it is easy to shew that these are unnecessary.

Suppose that the permissible pressure fluctuation is 4000 to 4500 lb./in.^{2}, whilst a charge of 1 in.^{3} of fuel is being pumped

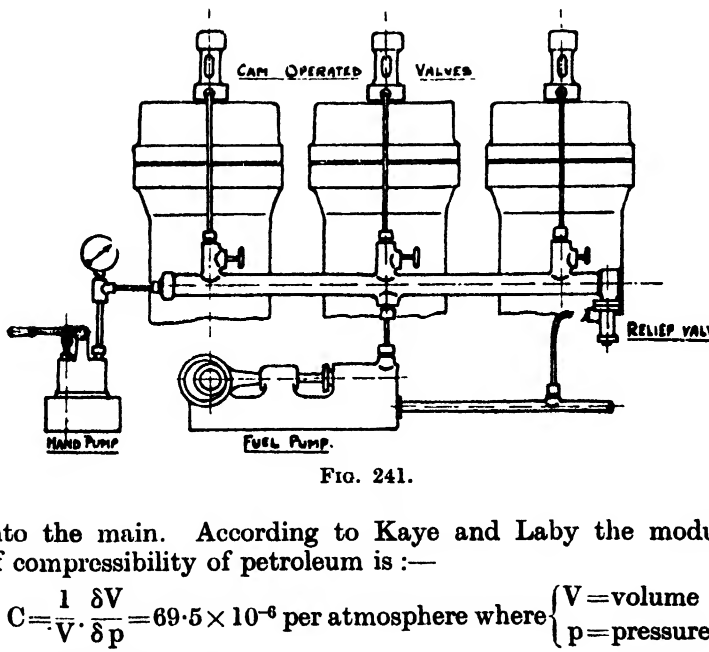


FIG. 241.

into the main. According to Kaye and Laby the modulus of compressibility of petroleum is :—

$$C = \frac{1}{V} \cdot \frac{\delta V}{\delta p} = 69 \cdot 5 \times 10^{-6} \text{ per atmosphere where } \begin{cases} V = \text{volume} \\ p = \text{pressure} \end{cases}$$

So the required capacity is :—

$$1 \text{ cub. in.} \times \frac{10^6}{69 \cdot 5} \times \frac{14 \cdot 7}{500} = 423 \text{ cub. in.}$$

apart from the yielding of the tube wall, which is in general small compared with the yielding of the fluid itself.

Fuel Pumps for Mechanical Injection.—With the rail system the fuel pumps are most conveniently driven by eccentrics, and the construction of the pump may follow very closely the designs already referred to in connection with the air blast system with the following differences :—

(1) The high pressure used, viz. 2000 to 8000 lb./in.^{2}, demands a stronger construction of eccentric and plunger connections and more rigid connection between the pump body and the eccentric shaft bearings.
(2) The pump body is preferably made of forged steel or bronze as cast iron is liable to be too porous.
(3) A relief valve and by-pass are provided in order to maintain a constant pressure in the delivery main.
(4) The full stroke of the plunger can be utilised for delivery, allowance being made for overload and slip.
(5) The clearance volume of the pump should be small to reduce slip by re-expansion of the compressed oil as in an air compressor.
(6) The slightest air lock must be avoided.
(7) The delivery fittings must be very substantial to stand strenuous tightening up. The same applied to the plunger gland, unless a lapped plunger working in a plain sleeve is adopted.

In those systems in which an individual pump controls the injection to each cylinder the delivery stroke must only occupy about 20 crank-shaft degrees at full load and less at light loads. This is usually arranged by one of two alternative methods. In the first, the plunger is driven outwards on the delivery stroke by means of a cam with a fairly sharp lift of the desired duration and returned on the suction stroke by a spring, the returning side of the cam being fairly gradual. The volume delivered is varied in accordance with the requirements of the load by some such device as a wedge piece interposed between the plunger and the cam follower, the position of the wedge piece being under the control of the governor.

At full load the thick end of the wedge is interposed and the full lift of the cam is utilised ; at light load a thinner part of the cam is interposed, and since the return stroke of the plunger under the influence of the spring is limited by a stop, there is clearance between the cam follower and the wedge, and consequently a shorter stroke of the plunger. This arrangement and others equivalent to it suffer from the defect that the point at which delivery starts is retarded as load is reduced.

In the second method this objection is overcome. The plunger operates with a constant stroke determined by a cam

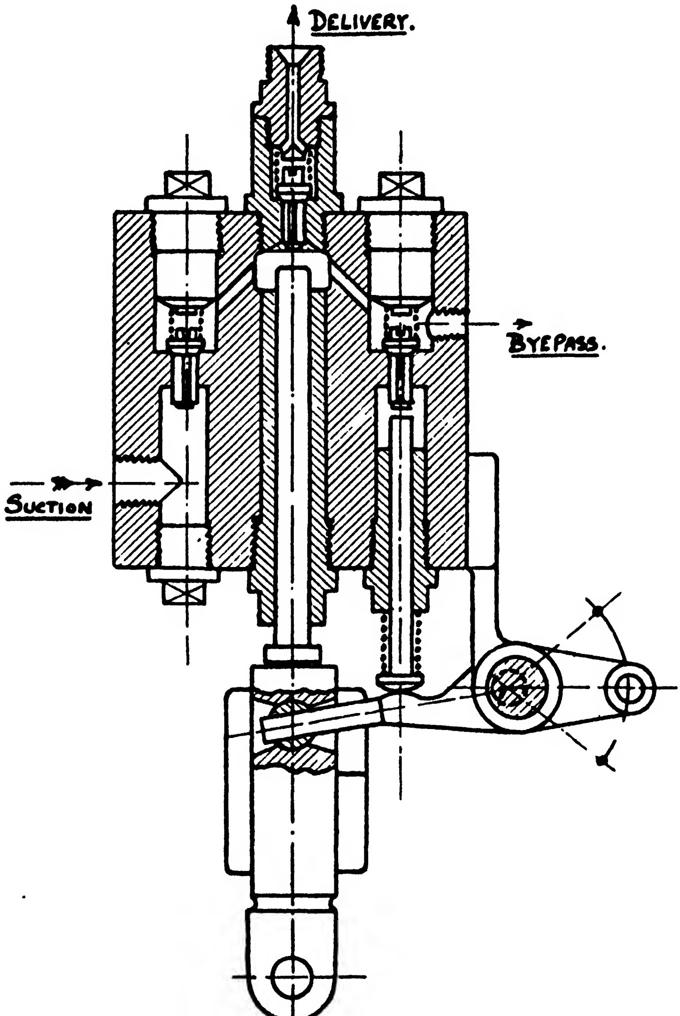


FIG. 242.

or eccentric, but the effective length of the stroke is shortened on light load by the forcing open of a so-called "spill" valve which allows the escape of fuel from the pump barrel to the suction side of the pump. The point at which the spill valve lifts is varied in accordance with the load by means essentially similar to those employed in the true Diesel fuel pump for regulating the point of seating of the suction valve (Fig. 242).

With this kind of pump the instant of injection remains constant at all loads.

With these pumps it is necessary to arrange the governing mechanism, whether spill valve gear or wedge mechanism or equivalent, in such a way that the operating forces do not appreciably react on the governor and are not subject to undue wear. Simplicity of adjustment should also be considered.

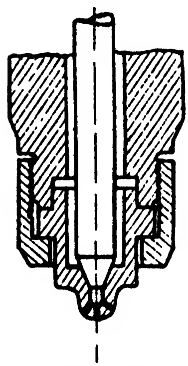


Fig. 243.

Fuel Valves.—The mechanically operated fuel valves associated with the rail system have been evolved from the air blast types of valve, and designs corresponding to the Augsburg and Swedish types have been used. Apart from the elimination of the blast air arrangements the essential differences are as follows :—

(1) The interior of the valve is filled with fuel oil instead of air, and plugs are arranged to vent air locks if necessary.

(2) The valve body is made of forged steel or bronze to withstand the high pressure.

(3) The spring pressure per unit of needle area is greater (in the "Augsburg" form) for the same reason.

(4) Drains are provided to carry away any gland leakage.

(5) Pulverisers are not required as the oil is broken up on emerging at high speed from the nozzle orifices.

(6) The needle is carried down as near to the nozzle orifices as possible to prevent after drip, i.e. the slow emergence of oil from the orifices after the valve has closed (Fig. 243).

(7) The nozzle or flame plate usually contains a plurality of minute holes of the order 20/000″ in diameter, or a single hole of larger size.

(8) A filter capable of stopping particles of grit of a diameter less than that of the nozzle orifices is put in circuit adjacent to the valve itself or even incorporated in the body of the latter. This is very desirable, if not

essential, as scale and grit in the connecting pipes are difficult to avoid.

(9) Means must be provided to vary the period of opening of the valve in accordance with the load. This can be achieved by eccentric mounting of the valve operating lever or equivalent means whereby roller clearance is increased. This, in general, retards the point at which injection starts as load is reduced, but this defect can be overcome by arranging that the movement of the cam roller during the increase of roller clearance shall follow the cam profile or even interfere with it, thus obtaining an advance of the point of injection.

A fuel valve of the automatic type may consist of a simple plug with a union at the outer end and a nozzle plate at the combustion end connected by a hole of small diameter, with or without a non-return valve near the nozzle end. With this type of valve very rapid building up and release of pressure in the fuel pump delivery pipe appears to be a necessary condition of successful working.

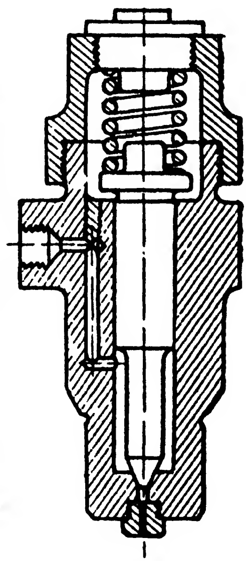


Fig. 244.

The preferred type contains a spring loaded needle valve seating close to the nozzle. The outer end of the needle passes through a bushed hole in the valve body, thus providing an unbalanced area on which the fuel pressure acts in lifting the needle against the spring pressure during the pump delivery period (Fig. 244). The spring tension per unit of needle area at the bushed hole determines the fuel pressure at the point of injection.

If the needle lifts against a stop the mean pressure during injection may be considerably greater than this if the nozzle orifices are sufficiently fine. If the latter are relatively coarse the needle lift will be very slight and the injection pressure will be sensibly constant.

Fuel Valve Nozzles.—A great deal of experimental work has been done by the various makers of mechanical injection engines to secure the most favourable type of fuel sprayers; the efficiency of the combustion process depends very largely on the degree of pulverisation and the character of the distribution of the fuel. Some of this work is referred to in the list of references at the end of the chapter. It seems to be established that the degree of pulverisation should strike a happy medium between the extremes of :—

(a) Too coarse pulverisation leading to slow combustion, and carbonisation due to liquid oil striking the piston or otherwise.

(b) Too fine pulverisation preventing the spray from adequately penetrating the charge and leading to undue concentration of oil in the neighbourhood of the injector, with the result that combustion is incomplete and smoke is formed.

In an ideal state of affairs the fuel in a fine state of division would be evenly distributed throughout the hot central core of the combustion chamber. When the latter consists of an elongated pocket or egg-shaped chamber the ideal programme can be approximately realised by the use of a single orifice nozzle.

If, on the other hand, the combustion chamber is wide and shallow a plurality of orifices is usually employed.

The size of orifice required may be determined approximately as in the following example :—

I.H.P. per cylinder (4 stroke cycle)	100
R.P.M.	300
Fuel consumption per I.H.P. hr. in lb. . . .	0·35
Density of fuel in lb. per ft.^{3}	55
Injection pressure lb./sq. in. (above ignition pressure)	4000
,, period in crank-shaft degrees . . .	30

Volume of oil injected per working stroke

$$=\frac{100\times 0{\cdot}35}{60\times 150\times 55}\text{ cubic feet.}$$

Duration of injection period

$$=\frac{60\times 30}{300\times 360}=\frac{1}{60}\text{ sec.}$$

Rate of discharge during injection period.

$$=\frac{60\times 100\times 0{\cdot}35}{60\times 150\times 55}=0{\cdot}00425 \text{ ft.}^3 \text{ per sec.}$$

Head corresponding to 4000 lb./in.^{2}

$$=\frac{4000\times 144}{55}=10{,}520 \text{ ft.}$$

Apparent velocity allowing coeff. of discharge$=0{\cdot}6$

$$=0{\cdot}6\sqrt{2\,g\times 10{,}500}=495 \text{ ft./sec.}$$

$\therefore$ Area required$=0{\cdot}00425\div 495$ ft.$^2=0{\cdot}00086$ in.^{2}

Required dia. for single orifice $=\sqrt{\dfrac{4}{\pi\times 860}}=39/1000''$

" " 6 equal orifices$=39/1000\div\sqrt{6}\bar{=}16/1000''$

Theory of the Jerk-Pump Injection System—Referring to the system illustrated in Fig. 240, if the fuel pump plunger is impulsively given a velocity which when multiplied by the ratio,

$$\frac{\text{Area of Plunger}}{\text{Sectional Area of Pipe}},$$

gives a velocity Vo to the oil at the pump end of the pipe, a pressure " wave " or surge is propagated with velocity,

$$V_w=\sqrt{Eg/\rho}\,\ldots\ldots\ldots\ldots\ldots\ldots(1)$$

in which E is the bulk modulus, i.e. the reciprocal of C, page 387, or

$$\frac{10^6}{69{\cdot}5}\text{ atmospheres}=\frac{10^6\times 14{\cdot}7}{69{\cdot}5}=210{,}000 \text{ lb./in.}^2$$

and ρ=density of oil, say 55 lb./ft.^{3}

$=0{\cdot}032$ lb./in.^{3}

$g=386$ in./sec.^{2}

Hence,

$$V_w=\sqrt{\frac{210{,}000\times 386}{0{\cdot}032}}=50{,}500 \text{ in./sec.}=4200 \text{ ft./sec.}$$

If the engine runs at 100 R.P.M. the time interval measured in crank-shaft degrees for the pressure wave to traverse one foot of pipe is :—

$$\frac{100\times 360}{4200\times 60}=0{\cdot}143 \text{ deg.}$$

If the engine runs at 1000 R.P.M. the corresponding interval is 1·43 degrees and so on.

It is best to have all the fuel pipes of a multi-cylinder engine nearly the same length. An engine running at 150 R.P.M. will work well with fuel pipes 15 feet long, the conditions with regard to lag being the same as those of an engine running at 1500 R.P.M. with fuel pipes 18 in. long.

A fraction of a second after the plunger has been started with impulsive velocity the pressure wave will have traversed a distance proportional to V_w, whilst the plunger will have displaced oil at the pump end of the pipe a distance proportional to V_0. The pressure generated is therefore :—

$$P_1 = E\frac{V_o}{V_w} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots (2)$$

$$= \frac{210{,}000\ V_o}{4200} = 50\ V_o \quad \ldots\ldots\ldots\ldots (3)$$

Values of P_1	=1000	2000	4000	6000	8000	lb./in.^{2}
,, ,, V_0	=20	40	80	120	160	ft./sec.

A pressure wave of the above intensity reaches the fuel valve and if it fails to open the valve, is reflected back upon itself, giving a pressure intensity at the valve of double the amounts tabulated above. If the valve lifts the pressure wave may be only reflected in part or not at all, depending on the flow through the valve. If the injection process lasts long enough in relation to the length of the pipe, the wave may be reflected backwards and forwards several times between the pump and the fuel valve.

The efflux of fuel at the fuel valve is equivalent to imparting a negative impulse at that end of the pipe. Similarly the opening of a spill valve at the pump initiates a negative pressure wave which cancels in due course the state of pressure in the pipe. Davies and Giffen have shewn how these processes may be calculated quite simply in steps.

In practice the fuel cam is usually so shaped that the fuel pump plunger is gradually accelerated up to a speed which is maintained nearly constant for a period and then decelerated (see Fig. 245). The injection takes place during the constant speed interval. In some designs of pump the beginning of the pump discharge is timed by the plunger overrunning ports and velocity is generated very rapidly. In others the building up of pressure and velocity may be gradual, in which case the spring loading of the fuel valves is relied upon to give a sharp opening. The pressure required to lift the fuel valve varies greatly as between different designs. When the fuel spray is arranged to discharge into a compact chamber in which the air charge acquires a high degree of ordered turbulence, the valve may be set to lift at a pressure of 1000 lb./in.^{2} or less; n fact a simple non-return valve or a capillary hole have been

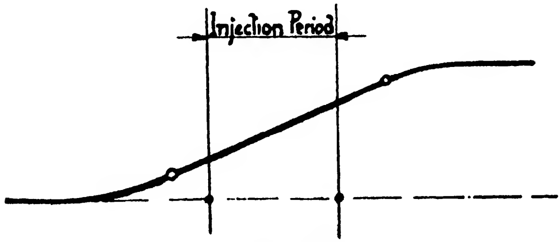


FIG. 245.

successfully used in such cases. If induction turbulence only is relied upon for mixing, a higher degree of pulverisation is usually necessary and the valve lifting pressure may be 2000 lb./in.^{2} and upwards. In some designs of fuel valve the needle rises against an adjustable stop so that the fuel pressure during injection is determined by the rate of discharge and may be, say, 6000 lb./in. when the valve is set to, say, 2500 lb./in. only. In other designs no stop is provided and the valve needle floats on the spring which controls the pressure during the injection period.

The duration of the injection period is usually about 15 to 20 crank-shaft degrees at rated full load to about 20° to 27° at maximum power. The whole of this period should come well inside the period of constant rate of rise of the cam. This rate of rise in relation to the diameter of the fuel pump plunger is a matter of prime importance.

Assume the following data :—

Normal Brake M.P.	73 lb./in.^{2}
Fuel Consumption	0·37 lb./B.H.P.
Specific gravity of fuel.	0·85

Then
$$\frac{\text{Stroke Volume of Cylinder}}{\text{Volume of fuel per injection}}$$
$$=\frac{33{,}000\times 62{\cdot}5\times 0{\cdot}85\times 60}{73\times 144\times 0{\cdot}37}=27{,}000$$

If the injection period at this B.M.P. is 15 crank-shaft degrees, the cam rise per degree (neglecting leakage),

$$=\left(\frac{B}{b}\right)^2 \cdot \frac{S}{15 \times 27{,}000} \text{ inches} \quad \ldots\ldots\ldots\ldots\ldots(4)$$

in which B/b = cylinder dia. ÷ plunger dia. and s = engine stroke, in inches.

Theory of Penetration.—The relation between time and distance penetrated by a jet of pulverised fuel when injected into compressed air is of fundamental importance in the

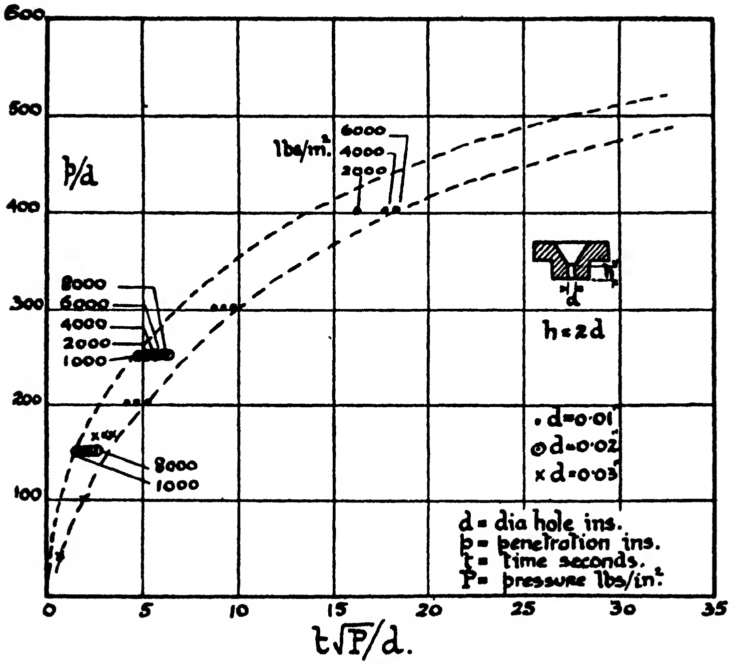


FIG. 246.

design of airless injection engines and some useful researches have been made on the subject. For a given size of orifice the higher the pressure the smaller are the particles into which the fuel is pulverised and the higher the initial velocity. The initial velocity varies as the sq. root of the difference of pressure between fuel and air. For a given pressure the larger the orifice diameter the coarser are the particles and the larger is the distance penetrated before the loss of any given fraction of the initial velocity.

Considerations of similarity suggest measuring penetration in terms of orifice diameter as the unit and plotting against a time scale $\times$ the square root of the pressure and $\div$ the orifice diameter.

Experimental data obtained by the National Advisory Committee for Aeronautics (U.S.A.) have been plotted by the author in this way on Fig. 246. The following points will be noted :—

(1) The points lie within a fairly narrow band.

(2) The influences of pressure and nozzle diameter as separate variables are small as compared with their joint effect in the variables chosen. This appears to justify the choice of these variables.

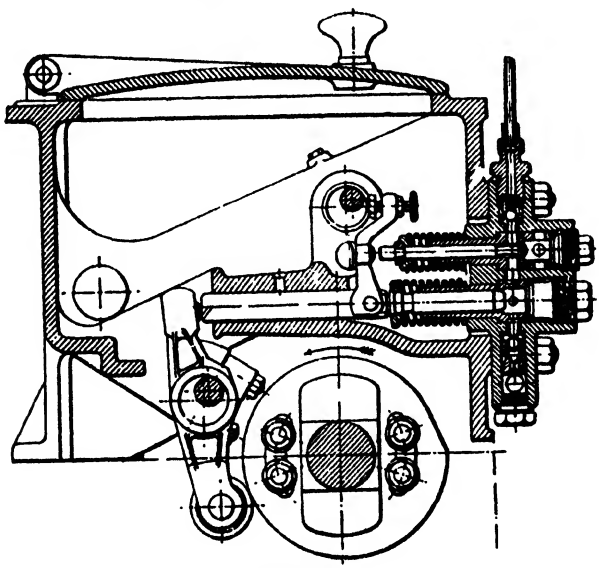


FIG. 247.

(3) The rate of penetration falls off very rapidly after a penetration of about 500 orifice diameters. If the nozzle is called upon to penetrate about half the cylinder diameter the minimum orifice diameter is about 1/1000 of the cylinder diameter.

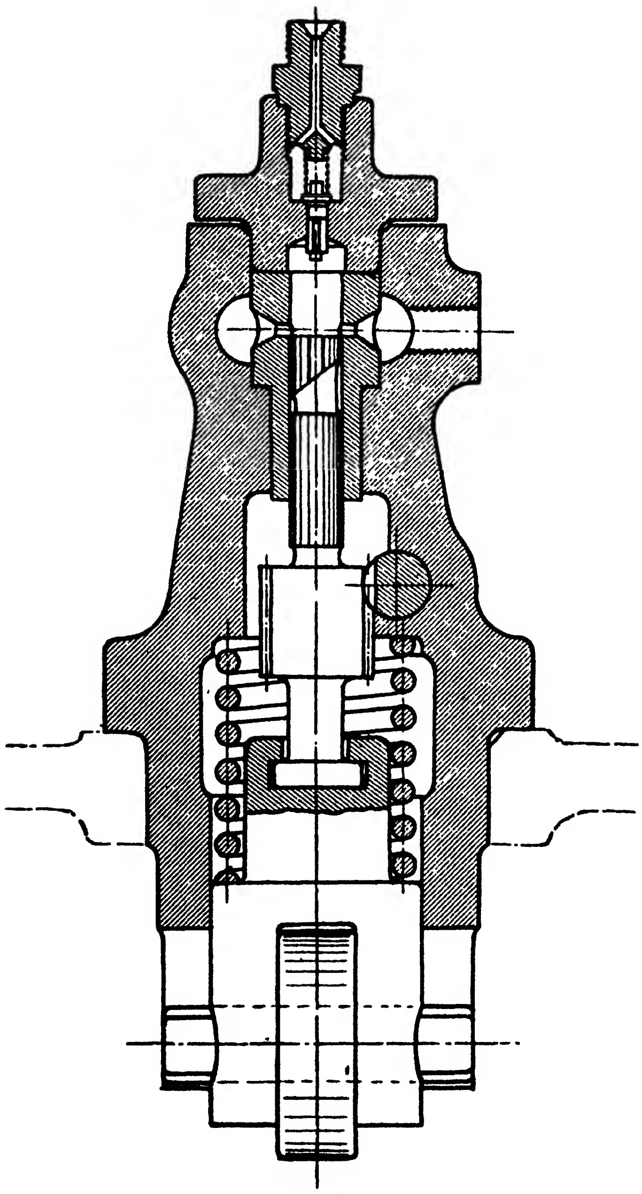


Fig. 248.

In practice cylinders of 24″ diameter give good results with orifices of 30/1000″ diameter or more and are insensitive to injection pressure over a fairly wide range.

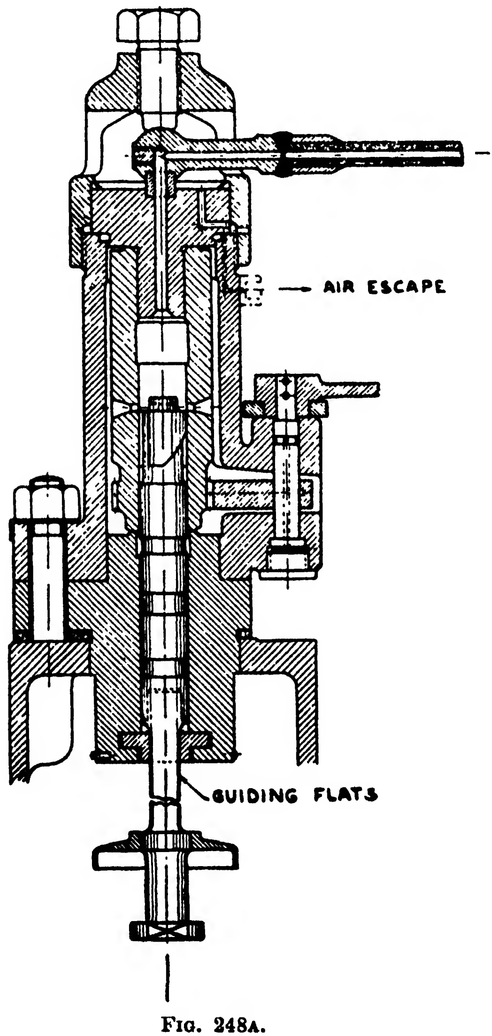


FIG. 248A.

With small cylinders of say 5″ diameter it is hardly practicable to use holes of say 6/1000″ and with direct injection by means of larger holes of say 10/1000″ this is a danger of the spray reaching the cylinder walls and causing dilution of the lubricating oil. In such cases the penetration may be reduced by

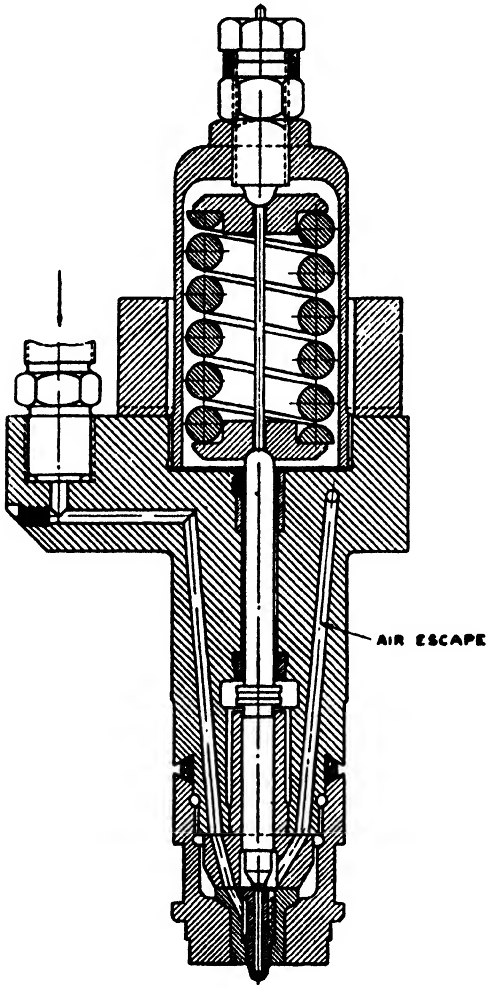


FIG. 249.

shortening the length of the orifice to say one diameter or less. Fig. 246 is drawn for holes 2 diameters long.

Typical Fuel Pumps.—Figs. 247 and 248 shew typical types of airless injection fuel pumps. The first (Ruston and Hornsby) is provided with suction, delivery and spill valves. The spill valve operates earlier or later according to the position of an eccentric shaft under the control of the governor, thus lengthening or shortening the delivery period in accordance with the load.

The second represents more or less diagrammatically a fairly numerous class of pumps of which the Bosch is a well-known example, in which the plunger acts as the spill valve. This is achieved by cutting a helical recess to which the pressure oil has access, in the surface of the plunger. The edge of the helical recess uncovers a hole leading to the suction chamber, thus terminating the delivery period. The instant of release of pressure is varied by governor or manual control by rotating the plunger relative to the sleeve or vice versa. There are several variations of scheme on the same general principle. In the example shewn the plunger is given a partial rotation by means of a rack engaging with teeth cut on the plunger. The manufacture of airless injection fuel pumps calls for a high degree of accuracy, particularly in relation to the plungers and sleeves. The plungers are made of case-hardened steel and the sleeves may be of toughened steel provided in some instances with liners of very close grained cast iron.

Fig. 248A shews a fuel pump of the above general type made by Burmeister and Wain.

Typical Fuel Valves.—Fig. 249 shews a typical fuel valve with spring loaded differential spindle, working in a guide which is separate from the valve body. This division facilitates giving these two parts the requisite high degree of precision of workmanship necessary for minimum leakage without sticking. The oil passage from the body to the guide passes through a ground joint which must be absolutely tight against the oil pressure.

The second connection at the top of the valve carries away any oil which leaks past the spindle. An air vent at the highest point of the pressure oil passage is frequently provided for priming. In some designs the lift of the valve is limited by a positive adjustable stop in the form of screwed pin passing through the top plug.

Fig. 250 shews a number of alternative fuel valve nozzles. In form A the spray emerges from a number (usually 4 to 6) fine orifices which are covered by the coned end of the spindle. In form B the orifices are not so covered but are drilled into a common hole controlled by the spindle. In form C the orifice by which the spray enters the cylinder takes the form of an annulus surrounding a " pintle " at the end of the spindle.

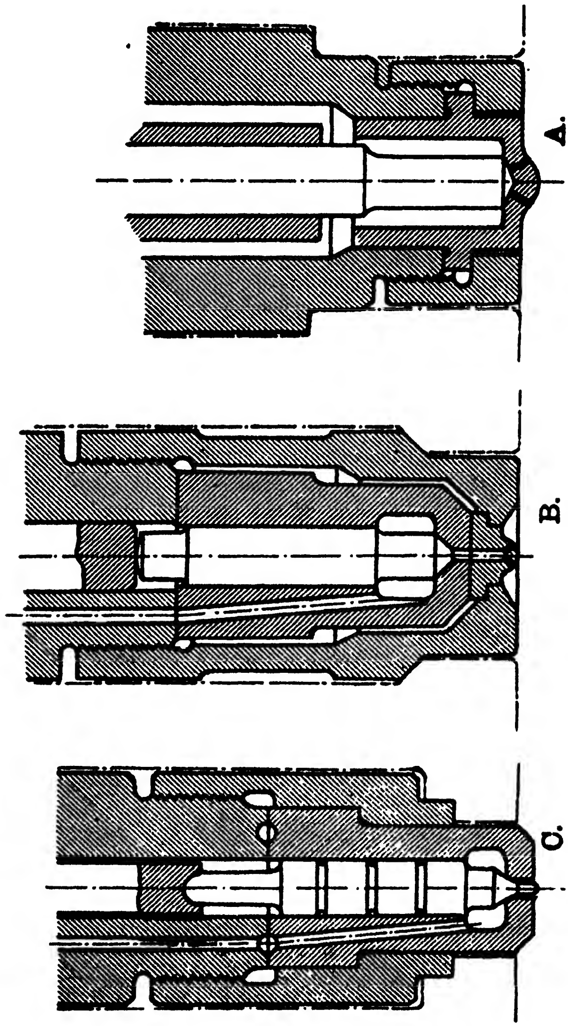


Fig. 250.

Combustion Chambers.—With so-called direct injection in which the compression space consists of a single chamber into which the fuel is injected, it is above all things important that the volume should be compact and of such a shape in relation to the sprays that the fuel is evenly divided. With a flat or

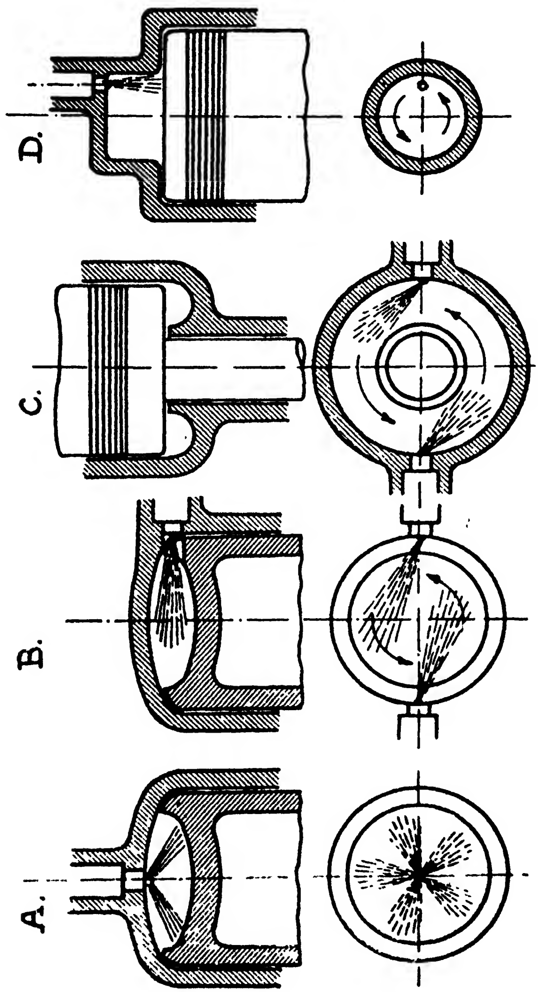


Fig. 251.

dished cylinder cover (Fig. 251 A and B), an oblate spheroidal or semi-spheroidal shape gives good results. If the fuel valve is central and vertical (A) the nozzle may be provided with

3 to 8 orifices (according to size), with directions making an included angle of about 100° to 150° depending on the depth of the combustion space in relation to the diameter. With side injection (B) it is advantageous to have two or more fuel valves giving oblique injection in the direction of rotational air swirl. A similar arrangement C may be used for the bottom of a double acting cylinder. Globular forms have already been illustrated in Figs. 238 and 239. An arrangement peculiar to a form of Ricardo engine is shewn in Fig. 251 D. In this case strong rotational air swirl performs the functions of

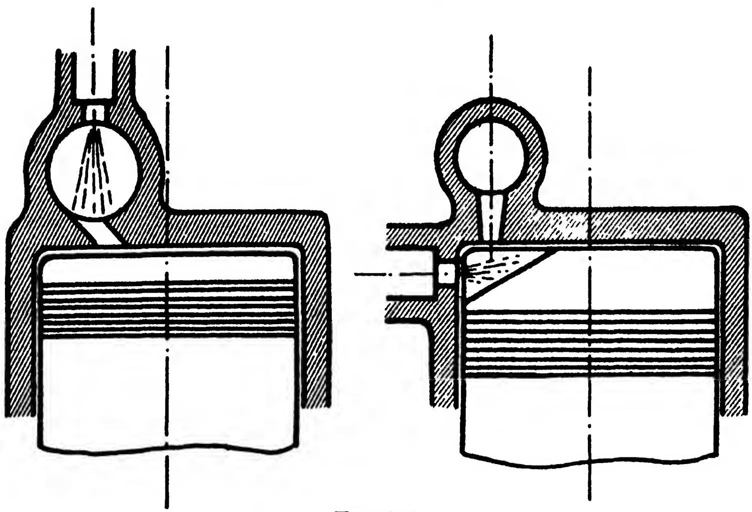


FIG. 252.

pulverisation and distribution of the fuel which is injected at comparatively low pressure (about 1000 lb./in.^{2}).

In other designs (particularly of small high speed engines) the fuel is sprayed into a chamber which communicates with the cylinder through a comparatively narrow neck, as in Fig. 252. The rush of gas through the neck at the end of the compression stroke and the beginning of the combustion stroke promotes pulverisation and turbulence. In yet another variation a chamber with a narrow neck is arranged in the cylinder cover or the piston or elsewhere for the promotion of turbulence, but the fuel is injected outside this chamber.

Miscellaneous Points.—An airless injection fuel pump is liable to become airlocked if the pressure on the suction side falls below atmospheric. This can arise on account of the suction pipes being too small in diameter or the head of fuel insufficient.

If it is inconvenient to arrange the feed tank sufficiently high the fuel pump suction line is charged to a pressure of 10 to 40 lb./in.^{2} by some simple type of pump.

On account of the small sizes of the fuel valve orifices, absence of grit is of special importance to airless injection engines. Where the size of the installation warrants the expense, centrifugal purifiers are installed for the purpose of fuel cleaning. In any case fine mesh filters are provided and a specially fine filter, capable of stopping grit of less diameter than the fuel valve orifices, is sometimes fitted in each pipe between fuel pump and fuel valve.

Typical airless injection indicator diagrams are shewn in Fig. 16, Chapter III. The compression is 400, the maximum pressure 610 and the mean pressure 90 lb./in.^{2}, and the fuel consumption 0·300 lb. per I.H.P. per hour.

Literature.—Bird, A. L., "The Ignition of Oil Jets."—I. Mech. E., December, 1927. Also "Experiments on Oil Jets and their Ignition."—I. Mech. E., 1926.

Davies, S. J., and Giffen, E., "Injection, Ignition, and Combustion in High Speed Heavy Oil Engines."—Diesel Users Assoc., March, 1931. Also "The Present Position of the High-Speed Heavy Oil Engine."—British Assoc., September, 1930.

Farmer, H. O., "Fuel Injection Systems for High Speed Oil Engines."—I. Mech. E., March, 1931.

Dicksee, C. B., "Some Problems connected with High Speed Compression Ignition Engine Development."—I. Mech. E., March, 1932.

Le Mesurier, L. J., and Stansfield, R., "Fuel Sprayer Action."—Diesel Users Assoc., March, 1933. Also "Combustion in Heavy Oil Engines."—N.E. Coast Inst. E. and S., February, 1932.

Bass, E. L., "Some Fuel Experiments on a Mechanical Injection Oil Engine."—Diesel Users Assoc., October, 1927.

Dicksee, C. B., "The High Speed Compression Ignition Engine."—Blackie.

CHAPTER XXI

AIR AND EXHAUST SYSTEM

Air Consumption.—If an engine has a brake mean pressure of 75 lbs./in., then the swept volume per B.H.P. hour, counting combustion strokes only is :

$$\frac{33{,}000 \times 60}{75 \times 144} = 183\cdot5 \text{ cub. ft./B.H.P. hour}$$

Defining " air supply ratio " as :—

Volume of air referred to ambient conditions of atmospheric pressure and temperature, divided by the swept volume, and assuming a four stroke engine with an air supply ratio of 0·85, the air consumption works out at 156 ft.^{3}/B.H.P. hour.

If the engine has a consumption of 0·375 lb./B.H.P. hour of fuel requiring 14 lbs. of air for the combustion of 1 lb. of fuel, and if the ambient air has a specific volume of 13·1 ft.^{3}/lb. corresponding to a pressure of 14·7 lbs./in.^{2} abs. and temperature 60° F., then the volume of air " burnt " is,

$$0\cdot375 \times 14 \times 13\cdot1 = 69\cdot0 \text{ ft.}^3\text{/B.H.P. hour}$$—

about 44% of the air supply.

The following table gives typical air consumptions of engines of various types.

FIG. 253.

TYPE.	Air supply Ratio.	B.M.P. lbs./in.^{2}	Air Consumption ft.^{3}/B.H.P. hr.
4 Stroke unblown .	·85	75	156
	·95	82	160
4 Stroke blown .	1·1	82	185
	1·3	97	185
4 Stroke turbo-charged . .	1·5	112	185
	2·0	150	185
2 Cycle . .	1·2	75	220
	1·5	75	275

Air and Exhaust System.—Road vehicle and tractor engines, portable engines for use on road repairs or building operations, stationary engines for use near cement works, or in sandy districts or other dusty situations are fitted with air filters to minimise the abrasive effects of mineral dust which would otherwise accelerate wear. Such filters may be of the cloth fabric type or the viscous type in which the dust is trapped by oily metallic surfaces which are cleaned and re-oiled from time to time. Apart from such filters, which may have an appreciable silencing effect, the air and exhaust system includes :—

Air suction inlets and silencers.
Blowers (see Chapter VII).
Air receivers and manifolds.
Air and exhaust valves or ports.
Exhaust manifolds.
Exhaust turbines (see Chapter VII).
Exhaust piping.
Silencers and waste heat boilers.

Air Inlets and Silencers.—The type of slotted pipe shown in Fig. 253 sometimes fitted to each inlet port of a normally aspirated four stroke engine has a moderate silencing effect and serves as a coarse filter. If the narrow slots (about 40/1000″) are allowed to become clogged, the breathing capacity of the engine may be seriously impaired.

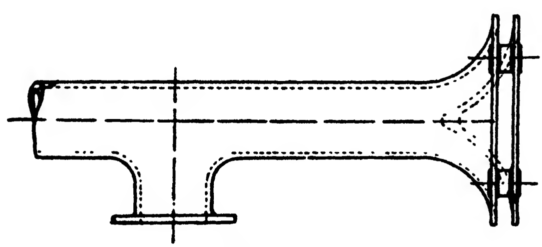


Fig. 254.

For multicylinder engines, the trumpet arrangement shown in Fig. 254 is an effective silencer particularly if lined with felt. A similar arrangement can be used as a suction silencer for centrifugal blowers.

Blowers of the roots type may emit a most intense highly pitched note unless a suitable silencer is provided. A very effective type of silencer consists of a labyrinthine felt-lined chamber through which air is drawn at a speed of about 30 feet second. Such silencers may with advantage be constructed as acoustic filters of the interference type, with two sound paths whose lengths differ by half a wave length of the dominant note.

The quietest type of roots blower seems to be the three lobed type with helical rotors. With suitable proportions the suction noise is slight and a simple form of inlet silencer is adequate.

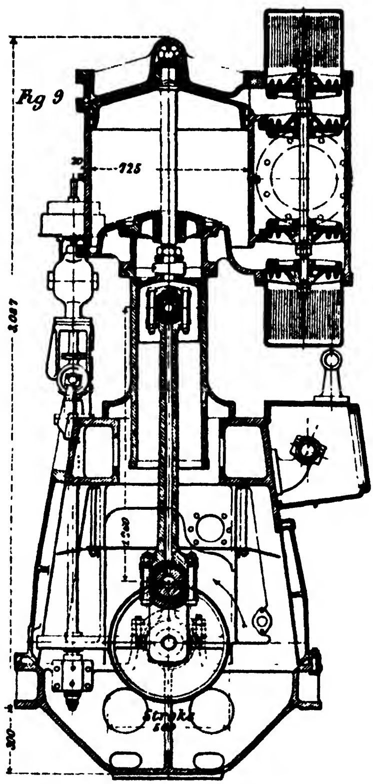


Fig. 255.

Blowers.—For the supercharging of four-stroke engines and the scavenging of two cycle engines, practically all known types of blower have been used, viz :—

(1) Reciprocating blowers (Fig. 255) :—

(a) Driven from a crank or cranks at the forward end of the engine ; such blowers may be simple or tandem, and the details of cylinders, pistons, guides, etc., somewhat resemble those of L.P. reciprocating steam engines.

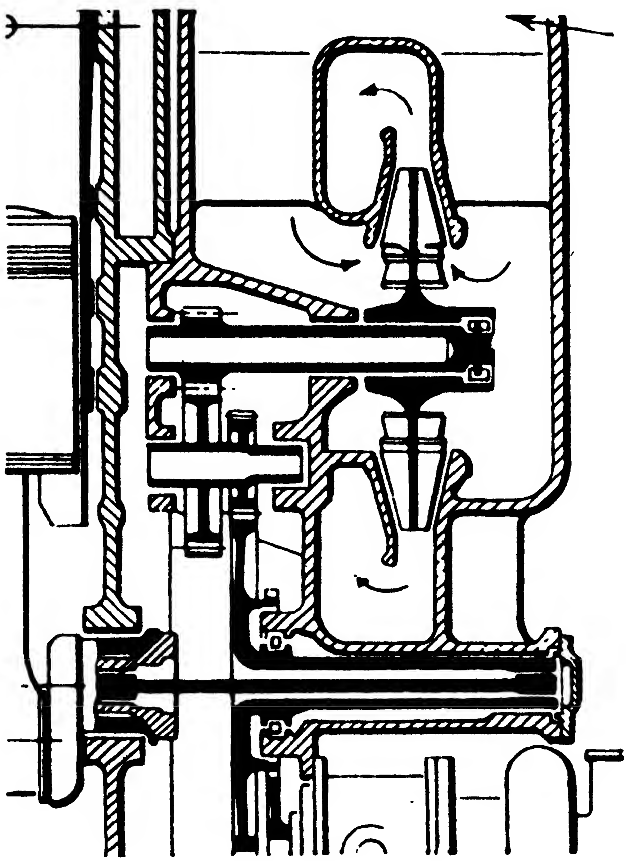


Fig. 256.

Various types of valves have been used to control the inlet and outlet ports, the favourite being thin plate valves of small controlled lift.

(b) Driven from the crosshead by links and levers as in the familiar air pump drive for a reciprocating steam engine.

(c) Formed by fitting suction and delivery valves to the enclosed bottom of the cylinder of a crosshead type four stroke engine (under-piston supercharge).

(2) Positive Rotary Blowers of the roots or other type; such blowers are usually driven by chain or spur gearing or a combination of both at two to three times crankshaft speed. The design of the blower drive offers little difficulty in the case of an engine running in one direction only and fitted with a flywheel giving a high degree of regularity. For reversible engines with light flywheels, the effects of manœuvring must be catered for by the provision of flexible couplings, slipping clutches and the like to mitigate shocks.

(3) Centrifugal Blowers (Fig. 256). These may be separately driven by electric motors or directly driven off the engine by Vee belts or chains or spur gearing or combinations of these at about ten times crankshaft speed. Spring drives and/or slipping clutches may be required to mitigate inertia effects due to starting, stopping and irregularity.

Air Receivers and Manifolds.—In four stroke engines with frames of monoblock type, an air receiver may be formed in the frame casting and connected to each cylinder cover air port by a pipe or elbow. The area of the passage should preferably increase steadily from the throat of the valve to about double at the entrance to the receiver, which it should join with a good radius to prevent contraction, and excessive wire drawing with consequent loss of volumetric efficiency.

An alternative to this arrangement consists of a sheet steel manifold of circular or rectangular section with a branch or connection to each cylinder cover port. The sectional area of the manifold should be sufficient to prevent undue pressure drop or local depression due to the demands of individual cylinders. Any such drop of pressure should be small in comparison with the unavoidable pressure drop through the inlet valve. For circular manifolds fed at one end only, a diameter of about 2½ times the diameter of the air inlet valve (assuming one valve) is adequate for six cylinders. For a 12 cylinder manifold fed at one end a diameter of about $3\frac{1}{4}$ times the diameter of the exhaust valve would be ample.

Similar arrangements may be used for supercharged four stroke engines.

For two cycle engines the air receiver between the blower and the air ports is made sufficient in volume to absorb fluctuations in supply and demand throughout the cycle with a negligible variation of pressure. With a cylindrical sheet steel receiver arranged along the back of the engine, the diameter for a six cylinder engine, with receiver fed at one end may be about $1{\cdot}2\sqrt{B.S.}$ (B = Bore, S = Stroke); for twelve cylinder fed at one end about $1{\cdot}5\sqrt{B.S.}$ These rough figures apply to single acting engines. For double acting engines the diameters are to be increased by some factor less than $\sqrt{2}$ since the demands of top and bottom are opposite in phase.

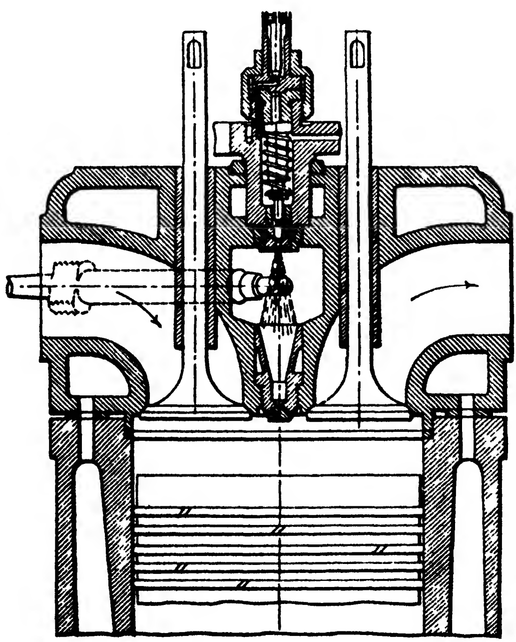


FIG. 257.

Air and Exhaust Valves.—Whilst sleeve valves have been successfully used for the control of air inlet and exhaust outlet, the majority of four stroke engines are fitted with inlet and exhaust valves of the poppet type. In general such valves are fitted in removable cages in the cylinder covers of large engines, whereas, in small engines they are fitted directly into the cylinder covers with the addition of renewable spindle guides and in some cases, heat-resisting seating rings. Since the inlet

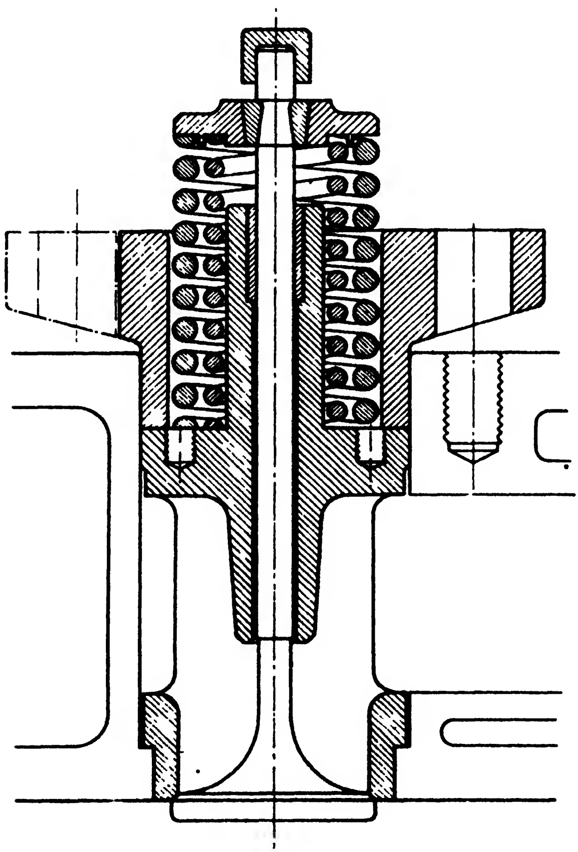


Fig. 258.

valves work under relatively cool conditions they may be made of low or medium carbon steel. Exhaust valves, on the other hand, are made of heat resisting steel such as silichrome or in high duty engines, of austenitic steels. Welded coatings of stellite or other heat-resisting alloys are sometimes applied to the working faces. The use of heat-resisting materials for exhaust valves greatly extends the periods between successive removals for grinding so that it has become practicable to dispense with valve cages in comparatively large cylinders up to 12″ diameter or more, thereby enabling valves of larger size to be used, so that high piston speeds are practicable without detriment to breathing capacity (Fig. 257).

Fig. 258 shows a typical air or exhaust valve in a cage. Two concentric valve springs are used and the stem itself is used as a guide. The spring cap is secured to the stem by split-cone washers and the valve actuating tappet makes contact either directly on the stellited end of the spindle or through the medium of a hardened steel cap. The contacting surface may be of point, line or surface type (Fig. 259).

Fig. 260 shows a water-cooled exhaust valve cage provided with a removable seating ring of heat-resisting material. The depth and thickness of metal under the outlet port must be

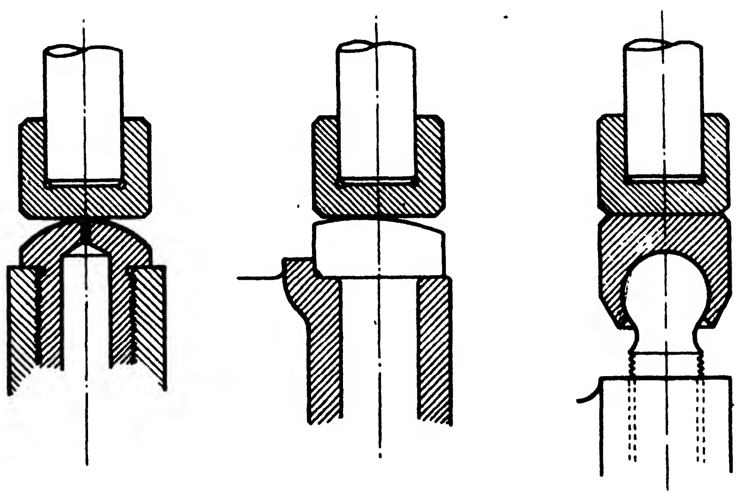


Fig. 259.

sufficient to prevent undue deflection, otherwise loss of compression and with that a loss of power may occur.

The cage is secured to the cover by two to four studs. The securing flange may be integral with the cast iron cage or a separate casting or drop forging.

For large slow-running engines it is possible to make the valve heads, separate from the spindles, of heat-resisting steel or special cast iron.

At the expense of some extra complication, two air and two exhaust valves of smaller size may be accommodated in each cylinder cover, thereby somewhat increasing breathing capacity at high speed, and reducing the noise of valve operation by reason of the reduced lift and closing velocity. This practice dates back over forty years, e.g. in the 560/780 Augsburg engines of the early 1900's. Such smaller valves ran cooler and longer between overhauls.

Several arrangements are in use for poppet exhaust valves of 2 cycle uniflow scavenge engines, viz., one or two valves in cages, one, two, or four valves without cages.

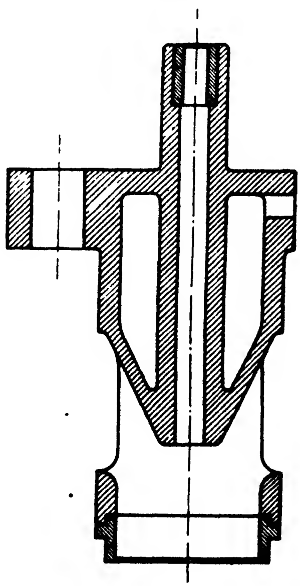


FIG. 260.

Such valves run at higher temperatures than the exhaust valves of four stroke engines, so that the most heat-resisting steels are necessary in high duty engines.

A single valve of this type is shown in Fig. 261. The outlet passage is carefully streamlined and carried clear of the cylinder cover. The cage is water-jacketed. Since the valve is operated once per revolution, special care needs to be taken in the design of the springs to avoid surging.

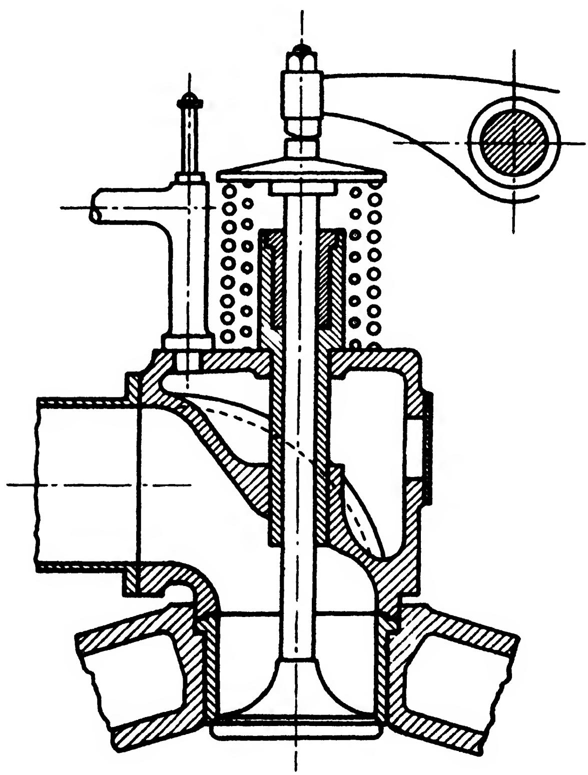


FIG. 261.

Dimensions of Air Suction Valves.—The requisite diameter for an air suction valve may be regarded as determined by the mean vacuum allowable on the suction stroke. Taking this to be 0·6 lb. below atmospheric pressure, the theoretical mean velocity is found from Fig. 28 to be about 280 feet per second, and taking the mean coefficient of discharge to be 0·70 this gives a mean apparent velocity of 195 feet per second. Now as regards the mean opening area of the valve, if we assume a harmonic cam opening and closing exactly at the upper and lower dead centres respectively, then the mean opening would be just half the maximum opening if the maximum lift is made $=\frac{1}{4}$ of the valve diameter. Actually the valve is always arranged to open before top centre and close after top dead centre, so that the mean area is usually more like 0·6 of the maximum area.

Adopting this figure, we may write :—

$$195 = V_a = V_p \times \frac{0{\cdot}785\ B^2}{0{\cdot}65 \times 0{\cdot}785\ d^2}$$

Where V_a = Apparent mean velocity of air in feet per second.
V_p = Mean piston speed in feet/seconds.
B = Bore of cylinder.
d = Diameter of suction valve.

From which,

$$\frac{d}{B} = \sqrt{1{\cdot}54\frac{V_p}{195}} = \sqrt{\frac{V_p}{127}}$$

Values of $\frac{d}{B}$ calculated from this formula for various piston speeds are given below and agree well with average practice.

Piston speed in ft. per min. .	500	600	700	800	900	1000	1100	1200
Ratio d ÷ B .	·257	·281	·303	·324	·343	·362	·380	·397

In the best practice the maximum lifts of the air and exhaust valves are frequently made as much as 0·28 of the valve diameter, as although the extra lift does not increase the maximum available area the mean area is increased and the opening at the dead centre is augmented without adopting an unduly long opening period or an awkward shape of cam.

Exhaust Lifting Devices.—Some form of exhaust lift for breaking compression is usually fitted to both land and marine engines. With the former the use of such a device on shutting down the engine obviates the tendency of the engine to swing in the reverse direction to that for which it was designed, just before stopping, and also facilitates turning the engine by hand. With marine engines some such device should come into operation automatically on reversing to prevent the possibility of compressing an unexhausted charge when motion is begun in the reverse direction. An expansion stroke executed in the ahead direction becomes a compression stroke in the astern direction and vice versa. Two hand devices are shewn in Fig. 262. Type A consists of a substantial steel lever provided with a slotted end which normally keeps it in a vertical position clear of the gear. To bring the lever into operation it is lifted to the extent of the slot and allowed to fall forwards.

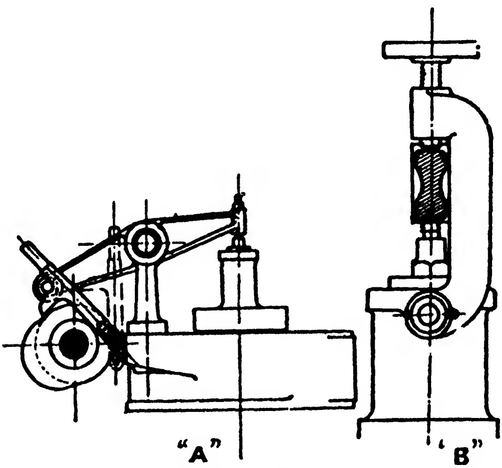


FIG. 262.

A projection on the lever then slips under an extension of the roller-pin on the first occasion of the valve being lifted and prevents its return. Type B consists of a link and screw, by means of which the valve may be depressed during the running of the engine and which normally lies alongside the casing. Fig. 263 shews three arrangements suitable for marine engines. In type A the valve is depressed by a pneumatic cylinder arranged above and in line with the valve. Type B consists of a series of cams (one cam over each exhaust valve lever) mounted on a shaft running from end to end of the engine. A turning movement of this shaft simultaneously depresses all the exhaust valves. Type C is appropriate to those engines in which the valve levers are operated by push rods. The first movement in reversing consists of swinging the lower end of the push rods out of the range of operation of the cams. By a scheme of linkwork, which is obvious from the illustration, the same movement introduces a lever with an inclined face under a roller provided for this purpose attached to the valve lever, with the result that the valve is held off its seat until the push-rods regain their normal working positions.

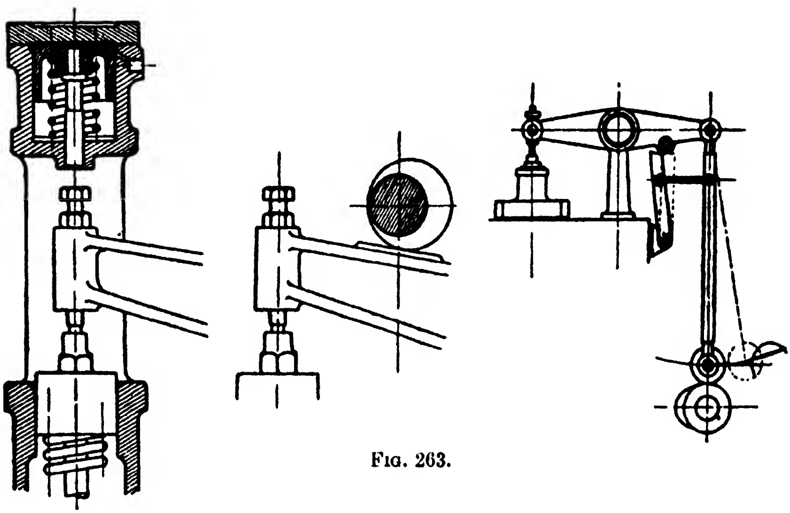


Fig. 263.

Exhaust Valve Springs.—Apart from the weight of the valve in itself there are causes tending to open the exhaust valve when it should be shut, viz.:—

(1) The vacuum on the suction stroke.

(2) With four-cylinder engines especially, the first rush of exhaust from the neighbouring cylinder if the latter discharges into the same collecting pipe.

In addition, the inertia of the valve and any levers, rods, etc., in connection therewith, tend to make the latter lose contact with the operating cam in the neighbourhood of maximum lift. These various influences are overcome by fitting a spring, the normal load of which is equivalent to a pressure of about 10 lb. per sq. in. in valve area in the case of slow speed engines, and anything up to about 20 lb. per sq. in. or more in the case of high speed engines. A method of computing the inertia effect will be dealt with in some detail as there is a tendency for higher speeds to be used in practice, and the principles involved have a wide application in the design of high speed machinery generally.

Inertia Effect of Valves.—Consider the simple arrangement shewn in Fig. 264, consisting of a valve directly operated by a cam without intermediate members. Let the weight of the valve guide and roller, etc., be "W" lb. The effect of the inertia of the spring may be allowed for by adding one-third of its weight to that of the other parts. Let "x" be the distance in inches of the valve from its seat at any instant. If the shape of the cam and the speed of the cam-shaft be known it is possible to express "x" in terms of the time "t" in seconds counted from the instant at which the valve begins to lift, by means either of an equation or a graph exhibiting the lift on a time base. If this equation (equation of motion) is available, then one differentiation with respect to "t" gives an expression for the velocity denoted by "$\dot{x}$" and a second differentiation gives the acceleration denoted by "$\ddot{x}$." If the relation between x and t is given by means of a graph, then the differentiation may be done by one or other of the graphical methods explained in books on practical mathematics. In either case, x being measured from the valve seat outwards, positive values of $\ddot{x}$ denote inertia effects tending to press the roller against the cam, and negative values of $\ddot{x}$ denote inertia effects tending to cause the roller to lose contact with the cam. Here we are only concerned with the negative values of $\ddot{x}$. If X denotes the resultant force on the valve, neglecting all effects except those due to the inertia,

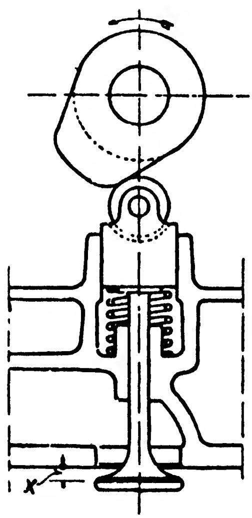


FIG. 264.

lines, and the follower is circular or flat, the motion may be divided into. parts, the accelerations of which are readily calculable thus :—

(a) Circular cam profile, flat follower—simple harmonic motion.
(b) Circular cam profile, circular follower—crank-connecting rod motion.
(c) Straight cam profile, circular follower—acceleration given by :—

$$\ddot{x}=w^2(R_1+R_2)\left[\frac{2\tan^2\theta+1}{\cos\theta}\right] \quad \ldots\ldots\ldots\ldots(5)$$

in which, R_1=radius of cam.
R_2= ,, ,, roller or other follower.
w=angular speed of cam-shaft.
$=\left(\frac{2\pi.\text{cam-shaft R.P.M.}}{60}\right)$
θ=angle turned through by cam-shaft measured from tangent point.

Values of θ	=	0	10	20	30	40	50 deg.
Values of $\left[\frac{2\tan^2\theta+1}{\cos\theta}\right]$		1·00	1·08	1·35	1·93	3·15	5·99

Effect of Levers and Push Rods, etc.—The simple case considered above of a cam operating directly on the end of the valve is seldom realised in practice, and a somewhat more general case, illustrated diagrammatically in Fig. 266, will be

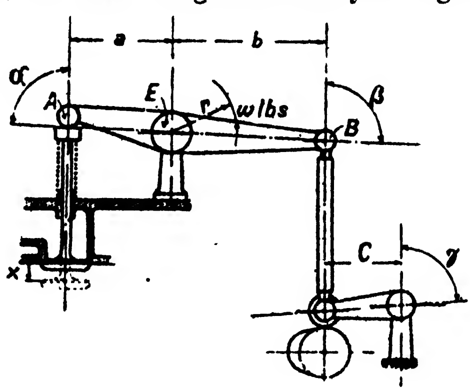


FIG. 266.

considered. We have now several different members, all participating in the motion and acquiring momentum which must be overcome by the spring. The problem is a simple case of the general theory of a system of one degree of freedom simplified by treating the " coefficient of inertia " as a constant instead of a function of x. Consider any particle of the lever AB situate at a distance " r " from the fulcrum E and having a weight " w " lb. If " s " denote the speed of this particle during any small displacement of the system, then :—

$$\dot{s}=\dot{x}.\frac{r}{a} \text{ and } \ddot{s}=\ddot{x}.\frac{r}{a}$$

The effective force acting on the particle is therefore equal to $\frac{w}{g}.\ddot{x}.\frac{r}{a}$ and the reaction X_1 at A due to all such particles of the lever is given by :—

$$X_1=\frac{\ddot{x}}{g.a^2}\Sigma w.r^2=\frac{\ddot{x}}{g}.W_1\frac{k_1^2}{a^2} \quad \ldots\ldots\ldots\ldots(6)$$

Where W_1 is the weight of the lever and k_1 is its radius of gyration about E. The expression $\frac{W_1k_1^2}{g.a^2}$ is the inertia co-efficient of the lever with respect to the co-ordinate x, and may be denoted by A_1. The total reaction X at A, due to the inertia effects of the valve itself, all the levers, push-rods, etc., is the sum of all the reactions due to the individual members, and therefore :—

$$X=\ddot{x}(A_0+A_1+A_2+A_3) \quad \ldots\ldots\ldots\ldots(7)$$

A_3 being $=\frac{W_0}{g}$ where W_0=weight of valve,

and A_2 is the inertia coefficient of the push-rod, and A_3 is that of the link.

By similar reasoning to that given above for A_1 it is found that $A_2=\frac{W_2}{g}\left(\frac{b}{a}\right)^2$, where W_2=weight of push-rod,

and $A_0=\frac{W_3}{g}\left(\frac{k_3b}{ac}\right)^2$, where W_3=weight of link, and k_3=its radius of gyration about its axis.

It will be seen at once that equation (6) is similar to (1), with inertia coefficient substituted for mass. The assumption made is that the angles $\alpha\ \beta\ \gamma$ do not deviate far from 90 degrees.

For ordinary practical purposes a deviation of 10 or 15 degrees on either side involves a negligible error.

Example: $W_0 = 6$ lb. $\quad a = 10''$ $\quad k_1 = 7''$
$W_1 = 15$ lb. $\quad b = 12''$ $\quad k_3 = 5''$
$W_2 = 8$ lb. $\quad c = 7''$
$W_3 = 5$ lb. $\quad \ddot{x} = 1600$ in./sec.^{2}

$$X = x(A_0 + A_1 + A_2 + A_3)$$
$$= \frac{1600}{386}\left[6 + 15\left(\frac{7}{10}\right)^2 + 8\left(\frac{12}{10}\right)^2 + \left(\frac{12\times 5}{10\times 7}\right)^2\right]$$
$$= 118 \text{ lb.}$$

For slow running engines the inertia may not amount to more than two or three lb. per sq. in. of valve area. In such cases the spring may be based on the inertia load plus about 6 lb. per sq. in. of valve area, to deal with the other effects which enter into the question.

With high speed engines the inertia load may be much greater and the additional allowance for friction, etc., as much as 15 lb./in.^{2} or more.

Strength and Deflection of Springs.—The usual formulæ for the safe load and the deflection of springs made of steel wire of circular section are given below for handy reference.

$$P = 0{\cdot}2f\cdot\frac{d^3}{r} \quad \ldots\ldots (8)$$

P = Safe load (maximum) in lb.
f = Safe stress, about 30,000 to 50,000 lb. per sq. in.
d = Diameter of wire in inches.
r = Mean radius of coils in inches.

And

$$\delta = \frac{64.n.r^3}{d^4} \times \frac{P}{G} \quad \ldots\ldots (9)$$

δ = Deflection in inches.
n = Number of turns (free).
G = Modulus of rigidity, usually taken to be about 12,000,000 lb. per sq. in.

The surging frequency F (per min.) of a helical spring is given approximately by:—

$$F = 200{,}000\,\frac{d}{r^2.\,n} \quad \ldots\ldots\ldots\ldots (10)$$

With cams subtending an angle of about 120°, violent surging seems to be almost inevitable if the cam-shaft R.P.M. is equal to or greater than $F \div 6$. Cam-shaft R.P.M. $\leqq F \div 16$ seem to involve little surging with ordinary cams. Values from $F \div 8$ to $F \div 12$ are in common use. The degree of violence of surging depends, with a given period, lift, and speed, on the lift curve. A pure cosine lift curve appears to be favourable on theoretical grounds but gives rather slow opening and closing.

By substitution equations (8), (9) and (10) give the useful relation,

$$F = 0{\cdot}214\frac{f}{\delta} \quad \ldots\ldots\ldots\ldots\ldots\ldots (11)$$

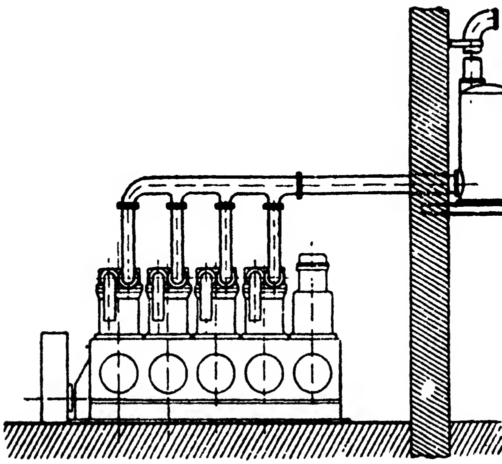


FIG. 267.

Exhaust Manifolds.—A once common arrangement of cast iron exhaust manifold and piping for a four stroke stationary engine is shown in Fig. 267. The connections between the covers and the manifold may be equal in area or a little larger than the area of the cylinder cover port. The diameter of the circular section manifold is preferably about 2½ times the diameter of the exhaust valve (one exhaust valve per cylinder) if weight and space allow, for a six-cylinder engine. For other numbers of cylinders the area may be *pro rata* with a minimum diameter of about twice the exhaust valve diameter for four cylinders or less. Rather less generous manifolds are used when weight and space considerations are important. Fig. 268 shows a similar arrangement with the manifold under the floor level; the connecting pipes must be water cooled or lagged and are liable to hamper access to the back inspection doors.

With either arrangement, if the manifolds are fabricated in steel, a circular section is advisable. More usually the exhaust manifold is arranged at or a little below cylinder cover level with short connections to the several cylinder covers. Frequently such manifolds are of cast iron, water jacketed, of circular or rectangular section (Fig. 269). Ample doors are provided to facilitate moulding and the removal of scale. So long as the jackets are clean, the metal temperature of the inner wall differs little from that of the water and no provision is required for expansion.

With uncooled manifolds connected to the covers by short connections, some provision for longitudinal expansion is necessary, in the form of bellows pieces, or glands (Fig. 270/1). The connection to each cover is usually provided with an exhaust thermometer or pyrometer and an exhaust gas test cock.

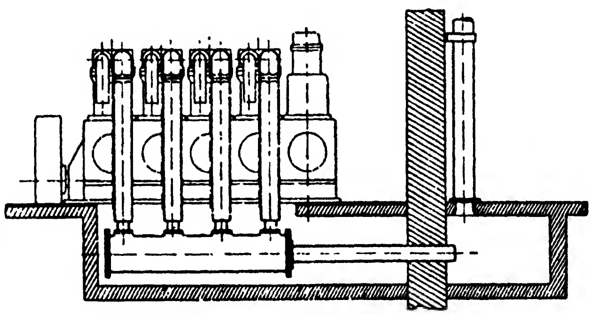


FIG. 268.

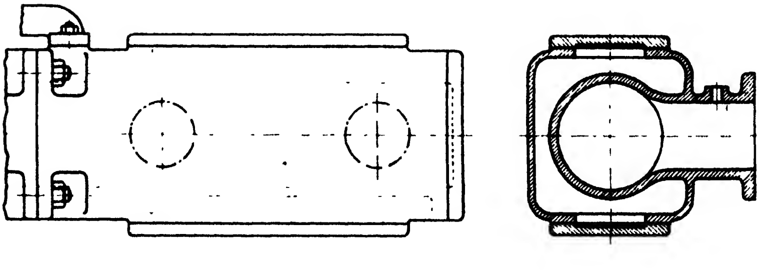


FIG. 269.

Two stroke engines with exhaust ports in the cylinder, uncovered by the piston are usually provided with water-cooled cast iron (smaller engines) exhaust manifolds arranged close alongside the cylinders with very short connecting branches or deflectors to guide the flow of exhaust in the direction of the outlet. It is difficult to formulate any rule for the diameter of such manifolds. If the timing of the air and exhaust ports is such that no reliance is placed on wave action in the manifold, then the larger the manifold the better. Diameters as much as $1{\cdot}2\sqrt{\text{B.S.}}$ are used. If, on the other hand, the exhaust impulse of one cylinder is used as a reaction to boost another cylinder, then too large a manifold may be detrimental. Questions of exhaust resonance and impulsive scavenge are involved (see Chapter VI).

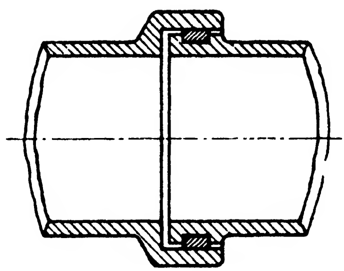


FIG. 270.

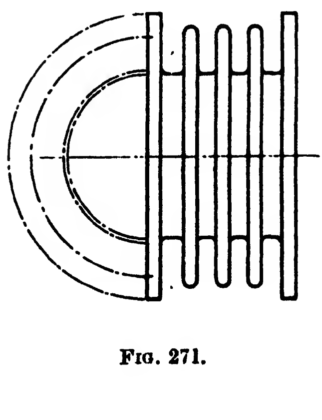


FIG. 271.

Something can be done to reduce scavenge resistance by special groupings of exhaust branches of calculated lengths and diameters. In particular two equal manifolds each served by the same number of cylinders firing at equal intervals, and such that the impulses in one manifold are 180° out of phase with the impulses in the other, if connected together behave somewhat as if exhausting into the atmosphere or a very large receiver. A similar arrangement has been used with four and eight cylinder four stroke engines to reduce effects of exhaust interference.

Silencers.—An unsilenced two cycle engine of, say, 150 B.H.P. can be heard three or four miles away. A receiver such as that shown in Fig. 272 having a capacity of about 50 or 60 cylinder volumes, is a very effective silencer, and its resistance is practically negligible. For many applications something less bulky is desired. For marine engines a cylindrical receiver having a capacity of about 20 to 30 cylinder volumes, with internal baffles giving three or four reversals of direction to the exhaust stream is usually sufficient. The area should nowhere be restricted to such an extent as to offer objectionable resistance. The resistance can be estimated approximately as described in Chapter VI by calculating the velocity head lost at each restriction. A silencer of elongated cylindrical shape is most convenient for accommodation in a funnel or engine room casing.

Diameter of Silencer	3·5 B
Length ,, ,, ...	5·5 B
Bore of Inlet Pipe	0·55 B

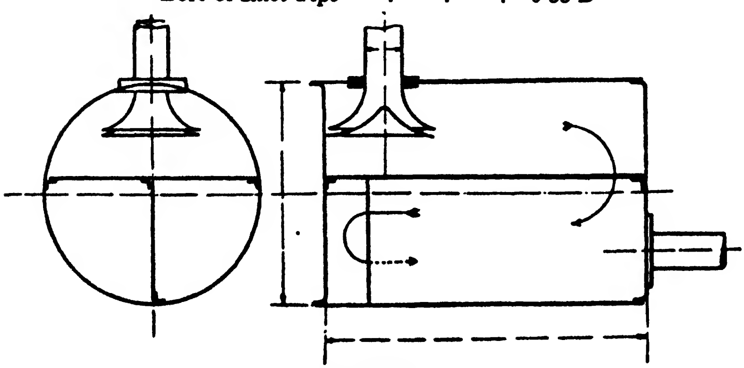


Fig. 272.

Still more compact silencers are constructed as acoustic filters consisting of chambers of various volumes and linear dimensions surrounding a central discharge pipe and communicating with it by rows of holes (Fig. 273). In a somewhat similar device a perforated pipe is surrounded by an annular chamber containing material with a high percentage of voids, e.g. steel borings, etc. (Fig. 274).

Silencers of the above types are supplied by specialist firms.

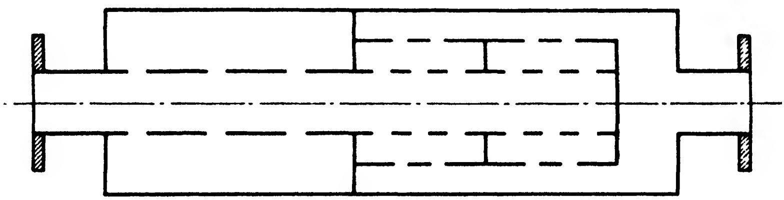


Fig. 273.

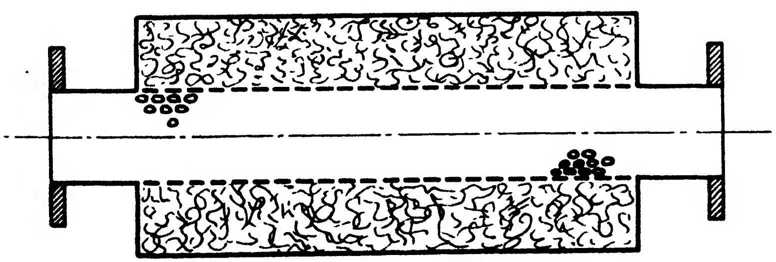


Fig. 274.

Literature.—For information on the mechanics of cam-operated mechanism, see :—

Goodman, J., "Mechanics Applied to Engineering."—Longmans.

Purday, H. F. P., *Motor-ship*, February, 1922.

Also numerous articles in the *Automobile Engineer*.

CHAPTER XXII

COMPRESSED AIR SYSTEM

As mentioned in Chapter I, the injection of fuel by means of an air blast is one of the outstanding characteristics of the Diesel Engine, and it seems probable that its use in the early experimental engines was suggested by the compressed air apparatus used to start the engine, and which still appears to be the most practicable method of doing this where large engines are concerned. The utility of the air blast is by no means confined to its function of injecting the fuel; in fact the widespread use of mechanical injection clearly indicates that effective atomisation can be obtained otherwise. As Guldner has pointed out, the use of an air blast probably secures a more efficient mixing of the cylinder contents than could be obtained in any other practicable manner. On this account it is possible that the air blast engine may continue to be built for using particularly refractory fuels, although the present tendency is to drop blast air injection in favour of airless injection on the scores of simplicity and economy of fuel consumption. There are two aspects from which heat efficiency should be viewed, viz. :—

(1) Economy in fuel consumption.
(2) The effect of efficient combustion in keeping the mean cycle temperature to a minimum.

The last consideration is a vital one from the point of view of reliability and durability, and experience abundantly proves that any considerable increase of the cycle temperature, due to overloading, leakage past valves, loss of volumetric efficiency or other causes is sufficient to convert an otherwise reliable machine into a source of trouble.

The blast air has a pressure which varies from about 900 to 1000 lb./sq. in. at full load, to about 600 lb./sq. in. at no load, and in land engines is usually supplied by a compressor forming an integral part of the engine. In marine installations the compressors are sometimes driven by separate auxiliary engines. An arrangement adopted by one maker was to drive the lower stages of the compressors by auxiliary engines, the last stage being performed by a high pressure plunger driven by the main engine. From the compressor the air passes, via coolers, to a blast reservoir or bottle of sufficient capacity to absorb fluctuations of pressure, and fitted with suitable distributing valves, one of which communicates with the fuel injection valves and another enables surplus air to be passed to the storage reservoirs provided for starting purposes. Cam-operated valves in the covers of one or more cylinders enable the stored air to be used, to give the engine the initial impetus which is necessary before firing can begin. With land engines of blast injection type the starting bottles are generally charged to a pressure of about 900 lb. per sq. in., and with the fly-wheels commonly used it is not necessary to provide starting valves for more than one cylinder out of three. With marine engines, storage pressures of about 350 lb. per sq. in. are more common, and in order to secure prompt starting from any position starting valves are fitted to every cylinder. With airless injection land engines the starting air pressure is about 300 lb./in.^{2} and upwards.

The air system also includes certain servo-motors or air engines, frequently used to perform operations of reversing the valve gear. The various organs will now be considered in more detail.

Air Compressors.—Four stroke land engines of the slow speed type using blast air injection, require compressors having a capacity of about 15 cubic feet per B.H.P. per hour, which assuming a volumetric efficiency of 80% corresponds to a stroke volume capacity of about 19 cubic feet per hour. Small high speed engines appear to require about 25% more than this allowance. The above method of basing the compressor capacity on the B.H.P. is not a very satisfactory one, as different makers have different views as to power rating. A better plan is to express the L.P. stroke volume as a percentage of the aggregate cylinder volume, and the following figures are representative of average practice for land engines. In view of the demands made on the system when manœuvring, marine installations are provided with a special manœuvring air compressor separately driven.

Bore of working cylinders.	Ratio L.P. stroke vol. ÷ Stroke vol. of working cylinders.	
	Four Stroke Engines.	Two Stroke Engines.
10	0·08	0·16
15	0·07	0·14
20	0·05	0·09

Number of Stages.—For small slow-running compressors two stages are sufficient, but a 9-inch diameter of low pressure cylinder appears to be about the safe limit; and even with this restriction it appears wise to abandon the principle of equal distribution of work between the stages. The small diameter of the H.P. cylinder affords little cooling surface for the dissipation of heat, and this consideration points to the advisability of arranging for the greater part of the work to be done in the L.P. stage, a conclusion which has been anticipated by experience.

Three-stage compressors are being increasingly used, even for the smaller sizes, but four stages have not come into general use even in the largest sizes. With three or four stages the principle of equal division of work is open to less objection, owing to the smaller ratio of compression in each stage.

Compressor Drives.—Almost every conceivable type of drive has been adopted at one time or another, and only the commonest are mentioned below:—

(1) Tandem two- or three-stage compressor driven off the crank-shaft. This arrangement appears to have the balance of advantages for most purposes.

(2) Tandem two-stage compressor driven by links and levers from each connecting rod or crosshead. This arrangement is expensive, but has the advantage of distributing the work amongst a number of small compressors, which are subject to less heat trouble than one compressor of the same total capacity. The suction pressure of the H.P. stage is usually sufficient to prevent reversal of thrust due to inertia, and consequently sweet running is secured.

(3) Similar to (2), but stages separate. This arrangement is bad, as the cooling surface is less than in case (2), and

the L.P. gear is subject to reversal of thrust due to inertia.

(4) Twin tandem three stage compressors driven off a pair of cranks at the forward end of the engine. This is a very suitable arrangement for large marine engines.

Constructive Details.—The construction of air compressors being a specialised branch of mechanical engineering, it is not proposed to give here more than a very brief reference to the subject. The cylinders of tandem two and three stage machines are frequently cast in one piece, including the water-jacket. The relatively low temperatures obtaining justify this procedure, provided sound castings can be obtained with reasonable regularity. The foundry work may be simplified in the case of two-stage compressors by the following division of material :—

L.P. Cylinder and Jacket—one casting.
L.P. Cover and H.P. Jacket—one casting.
H.P. Liner and H.P. Cover—separate castings.

One advantage of this scheme is the possibility of renewing the H.P. liner when worn. The latter is peculiarly subject to rapid wear on account of the high pressure behind the rings.

Similar arrangements are of course possible with three-stage machines. The possibility of increasing the cooling surface of the H.P. liner by means of ribs does not appear to have received very much attention. The trunk pistons serve as admirable crossheads, being almost entirely free from the heat trouble to which the pistons of internal combustion engines are subject. For the L.P. and intermediate stages Ramsbottom rings are usually fitted. In very large machines the latter can also be fitted to the H.P. plunger. Small H.P. plungers are usually fitted with some arrangement similar to that shewn in Fig. 275. The only thing which need be said about the connecting rods is that on account of the thrust being always in one direction, special care is required in the details of lubrication.

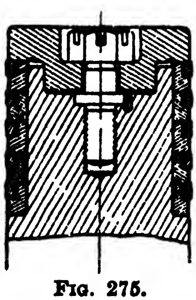


FIG. 275.

The design of valves has an important bearing on the success or failure of a compressor. The chief evils to be avoided are :—

(1) Sticking of the valves off their seats, due to deposits of carbonised oil.
(2) Damage to valves or valve seats, due to hammering.

The first is influenced more by the efficiency of the cooling arrangements and the compression ratio than with the design of the valves themselves. For obvious reasons the H.P. valves are most subject to this trouble.

The second trouble is usually due to the valves being too heavy, having too much lift, or the failure to provide adequate cushioning, and in successful designs is avoided by one or more of the following means :—

(1) Making the valves in the form of very light plates or discs with a very small lift.
(2) Providing a large number of very small valves in place of one or two large ones.
(3) Where large valves of considerable weight are used, arranging for some sort of dash-pot action.

All the valves should be easy of access and removal. Experiments with existing types of compressor seem to indicate that makers are inclined to base their valve dimensions on an air speed very much lower than is necessary. One or two per cent loss of efficiency is of small importance, if such a sacrifice enables the size of the valves to be reduced.

In some designs the intercoolers are separate from the compressor cylinder, and in others take the form of pipe-coils arranged round the compressor cylinders inside a removable water-jacket. Vibration of the coils should be prevented by adequate clamps and stays, and no sharp bends are allowable, on account of a scouring action (presumably due to turbulent flow) which in acute cases may cause fracture of the pipe in a short time. L.P. intercoolers are sometimes made similar to tubular condensers, and in other designs take the form of a cast-iron vessel provided with internal helical baffles which give rise to turbulent flow and increase greatly the efficiency of the cooling surface. It is desirable in all cases to fit a final cooler, to reduce the temperature of the fully compressed air before entering the blast receiver. Each receiver should be fitted with safety valve and drain. One or two isolated cases of explosion, traceable to accumulation of lubricating oil in the intercooler system, emphasise the necessity for these fittings. Some makers fit special " purge-pots " in communication with each receiver for the collection and discharge of condensed water and oil.

Calculations for Compressors.—In calculating the L.P. stroke volume required to furnish a given free air capacity, allowance must be made for the volumetric efficiency, which depends mainly on the clearance space and the delivery pressure. For example, suppose the clearance to be 3% of the stroke volume and the receiver pressure to be 150 lb. per sq. in. On the suction stroke, the suction valve will not begin to lift until the air left in the clearance space has expanded down to atmospheric pressure. If this clearance air expands according to the law :—

$$P.V.^{1.2} = \text{constant},$$

then its expanded volume expressed as a percentage of the stroke volume will be :—

$$3 \times \left(\frac{164 \cdot 7}{14 \cdot 7}\right)^{\frac{1}{1 \cdot 2}} = 22 \cdot 5\%$$

Subtracting its original volume, viz. 3%, the amount by which the effective stroke is shortened is 19·5%, and the volumetric efficiency is 80·5%. A further deduction should strictly be made for the fact that at the beginning of the compression stroke the cylinder contents are in general at a pressure slightly less than atmosphere. One or two per cent will usually cover this contingency. This example is sufficient to shew the importance of reducing the clearance volume of the L.P. cylinder to a minimum. The efficiencies of the L.P. or H.P. cylinders may be found similarly, but only influence the volumetric efficiency of the compressor as a whole indirectly by raising the receiver pressure above the value it would have if there were no clearance. It will be evident on reflection that leakage past the H.P. delivery valves will also raise the receiver pressures, and for this reason it is desirable to fit pressure gauges to all receivers so that the condition of the valves may be inferred from the gauge readings.

Assuming perfect intercooling and equal volumetric efficiency in all the stages, the pressure of the atmosphere and the receiver pressures (absolute) will be in inverse ratio to the stroke volumes of the cylinders, and equal division of work between the stages will be secured by the proportions given below :—

Two Stages.

Atmospheric Pressure.	Intermediate Pressure.	High Pressure.
1 at. = 14·7 lb./in.^{2}	8 at. = 118 lb./in.^{2}	64 at. = 940 lb./in.^{2}

L.P. volume = 8 H.P. volume.

Three Stages.

Atmospheric Pressure.	1st Intermediate Pressure.	2nd Intermediate Pressure.	High Pressure.
1 at. = 14·7 lb./in.^{2}	4 at. = 58·8 lb./in.^{2}	16 at. = 235 lb./in.^{2}	64 at. = 940 lb./in.^{2}

L.P. volume : I.P. volume : H.P. volume = 16 : 4 : 1.

In practice better results are obtained with two-stage machines by the following proportions:—

Atmospheric Pressure.	Intermediate Pressure.	High Pressure.
1 at. = 14·7 lb./in.^{2}	12 at. = 177 lb./in.^{2}	64 at. = 940 lb./in.^{2}

L.P. volume = 12 H.P. volume.

Assuming that the L.P. stroke volume has been determined by some such considerations as the above, the actual cylinder dimensions are found by selecting suitable values for the piston speed and the L.P. stroke to bore ratio. The piston speeds commonly used lie between about 300 to 600 feet per minute, the lower speeds being usually associated with small machines. With a little care it is possible, by making small variations in the strokes, to design a series of four or five compressors suitable for engines covering a wide range of powers. Such a scheme involves sacrifices in some cases which, however, would appear to be quite outweighed by the advantages of standardisation. The ratio of stroke to bore of the L.P. cylinder will generally lie between about 0·7 and 1·7. The valve areas for each stage are based on some figure for the mean velocity obtained, in the case of suction valve, by multiplying the mean piston speed by the ratio of piston to valve area, and in the case of delivery valves the mean piston speed during the delivery period by the same ratio. Certain continental authorities recommend speeds not exceeding 80 and 115 feet per second for the suction and delivery respectively, but it appears that these figures may be doubled or even trebled with impunity, and sometimes to advantage.

The calculations of the strength of the various parts are straightforward, involving no special principles, and are therefore passed over. The cooling surface to be provided for intercooling is a very important matter, and the following figures from a successful design may be useful as a basis of comparison in the absence of first-hand experimental data.

THREE-STAGE COMPRESSOR. Free air capacity, 130 ft.^{3}/min.

Cooling surface, copper-pipe coils. $\begin{cases} \text{L.P.} & 8{\cdot}7 \text{ ft.}^2 \\ \text{L.P.} & 3{\cdot}6 \text{ ft.}^2 \\ \text{H.P.} & 3{\cdot}6 \text{ ft.}^2 \end{cases}$

The subject of heat transmission being a very important one in connection with internal combustion engines, a brief reference to the usual theory is inserted below.

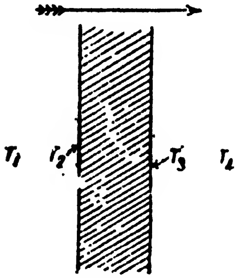


FIG. 276.

Transmission of Heat through Plates.—Referring to Fig. 276, the direction of heat flow is indicated by the arrow and the symbols t_1, t_2, t_3, t_4 denote the temperatures of the hot fluid, the hot side of the plate, the cold side of the plate, and the cold fluid respectively. The total heat drop consists of three stages :—

(1) An apparently sudden drop from the hot fluid to the plate.
(2) A steady gradient across the thickness of the plate.
(3) An apparently sudden drop from the plate to the cold fluid.

The assumption is that the rate of heat flow is dependent only on the temperature drop, the thickness of the plate, and the speeds of the fluids employed. On this assumption, if Q is the amount of heat transmitted per hour per unit of area, then :—

$$Q=a_1(t_1-t_2)=\frac{\lambda}{d}(t_2-t_3)=a_2(t_3-t_4)\ldots\ldots\ldots(1)$$

from which

$$Q=\frac{(t_1-t_4)}{\frac{1}{a_1}+\frac{1}{a_2}+\frac{d}{\lambda}}\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(2)$$

Where d=thickness of plate and a_1, a_2, and λ are constants.

For compressed air in pipes a_1 is given approximately by :—

$$a_1=\text{about } 3{\cdot}0\,\frac{w}{a}\ \ldots\ (\tfrac{1}{2}''\text{ pipe})\text{ to } a_1=\text{about } 2{\cdot}2\,\frac{w}{a}\ \ldots\ (2''\text{ pipe}),$$

where w/a=mass flow in lb. per ft.^{2} of sectional area.

For nominally still water a_2=about 100 (B.T.U./hr. ft.^{2} F.°).

Values of λ are given below for various metals :—

Iron	460	B.T.U./ft.^{2} deg. F. per inch of thickness.
Mild steel	320	
Copper	2100	
Brass	740	
Scale (average)	12	

The values of λ are well determined, but unfortunately the term involving this constant is the least important of the three as the bulk of the heat drop occurs at the surfaces of the plates. The values of a_1 and a_2 must be used with caution as an examination of published data indicates that these "constants" are subject to enormous variation under different circumstances. In particular, the value of a_2 is greatly increased by the eddying motion produced by high speed flow through narrow channels or tubes. If the water boils at the surface a_2 may be about 2000 even if the water is nominally "still."

The value of a_1 for compressed air is somewhat higher at high temperatures than at low. For further information on the subject of emissivities and heat transmission generally the reader is referred to the sources of information mentioned below.*

Equation (2) deserves careful consideration as it contains the crux of many heat transmission problems. The term involving "d" and "λ" is usually negligible (except in cases of very rapid transmission) unless due allowance is made for scale or deposit. It sometimes happens that either a_1 or a_2 is almost negligible in comparison with the other.

Air Reservoirs.—The usual arrangement of H.P. air bottles for land engines is shewn in Fig. 277. This scheme was devised in the very early days of the development of the Diesel Engine and no substantial improvement has been made on it in recent years. Two starting and one blast-air bottles are provided, all designed for a working pressure of about 1000 lb. per sq. in. One of the starting bottles serves as a reserve, in case of a failure to start the engine, due to any derangement. In the

* "High-speed Internal Combustion Engines," Judge. "Notes on Recent Researches," paper by Prof. Petavel: Manchester Assoc. of Engineers, Oct., 1915. "The Laws of Heat Transmission," lecture by Prof. Nicholson: Junior Inst., Jan., 1909. "Heat Transmission," by Royds: Constable. "The Calculation of Heat Transmission," by Fishenden and Saunders: H.M. Stationery Office.

event of such failure, every care is taken to make certain that the engine is in perfect order before using the reserve bottle, and it seldom happens in practice that the bottles require to be replenished from outside sources of supply. The connections between the bottles, the air compressor and the engine should

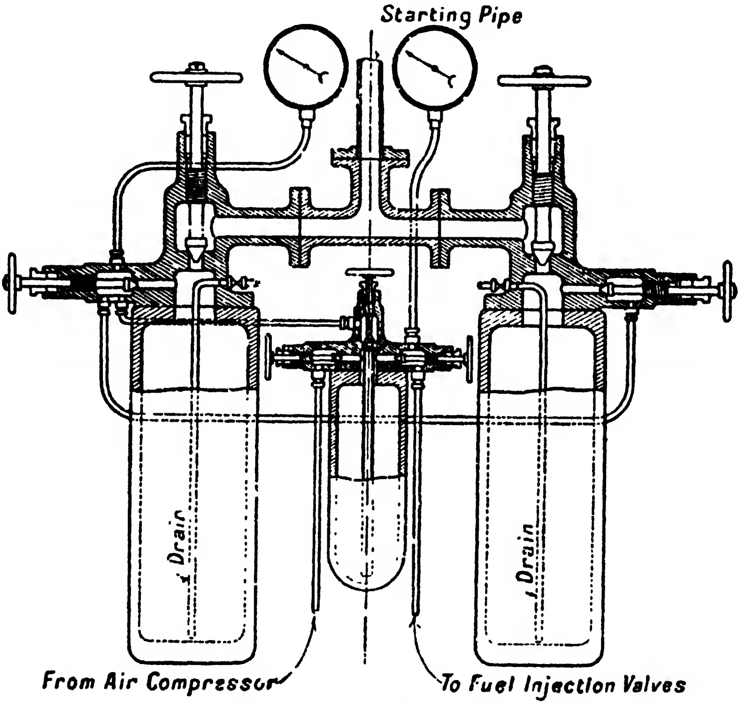


FIG. 277.

be quite clear from the diagram. Only one or two points will be mentioned.

(1) Before starting up, it is possible to ascertain the pressure in each of the three bottles by opening up the appropriate valve on each bottle-head in rotation. In each case the pressure is recorded on the left-hand gauge.

(2) The pressure in any pair or all three bottles may be equalised by opening up a pair or all three such valves.

(3) The right-hand gauge registers the blast pressure on the engine side. By throttling the blast control valve on

the blast bottle-head the injection pressure may be regulated below that of the bottle. This is done when replenishing the starting bottle on light load. It is thus possible to pump up the starting vessel to 1000 lb./in.^{2} whilst the blast pressure is only 600 lb./in.^{2}, as required for light running.

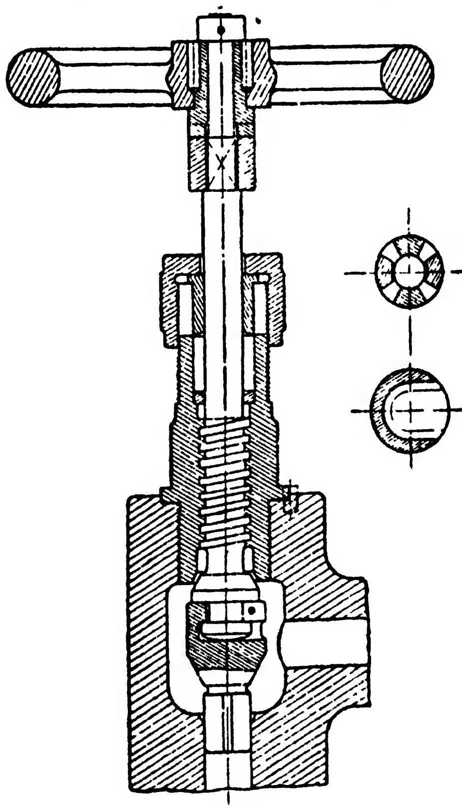


FIG. 278.

The bottle-heads containing the various valves are usually machined from a solid block of steel. A detail of one of the valves is shewn in Fig. 278.

The bottles themselves are of weldless steel, and a neck is frequently screwed on, as in Fig. 279. Some idea of the capacities of the bottles commonly provided may be gathered from the following table :—

Total Capacity of H.P. Starting Air Bottles
(four stroke engines)

Engines of about 9" bore, having one to six cylinders—about fourteen times the stroke volume of one cylinder.

Engines of about 24" bore, having one to six cylinders—about seven times the stroke volume of one cylinder.

Owing to the expensive machinery required to manufacture weldless reservoirs, only a certain limited number of standard sizes are available at reasonable prices, and in very large installations it is sometimes necessary to provide groups of four or more starting vessels. The usual working stress is about

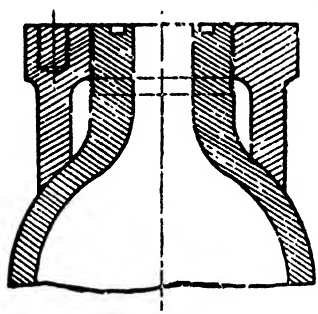


Fig. 279.

8000 lb./in.^{2}, and it is customary to specify a water-test pressure of double the working pressure.

With airless injection engines the starting air pressure is 250 to 500 lbs. per sq. in. and the system is much simplified.

Riveted Air Reservoirs.—For marine installations of high power it is usual to use a lower air pressure of about 350 lb./in.^{2} for starting purposes. The air reservoirs now require to have a very much larger cubic capacity, but the reduced pressure permits of the employment of riveted reservoirs. The construction of the latter need not be dealt with here, being comparable with that of the steam drums of large water-tube boilers. Adequate drainage for condensed water and oil vapour, and also a manhole for inspection and cleaning, should be provided. These matters, as well as others dealing with the strength of the riveted joints, the quality of material to be used and the tests to be carried out on completion, form the subject matter of regulations by the various insurance societies and the Board of Trade.

Blast Piping System.—From the blast bottle the injection air passes to a main running along the back of the engine, where it is distributed by short lengths of pipe to the several fuel valves, as in Fig. 280. Where one fuel pump is provided for a number of cylinders the fuel distributors may be made to serve as distributing tee-pieces for the blast air. In marine

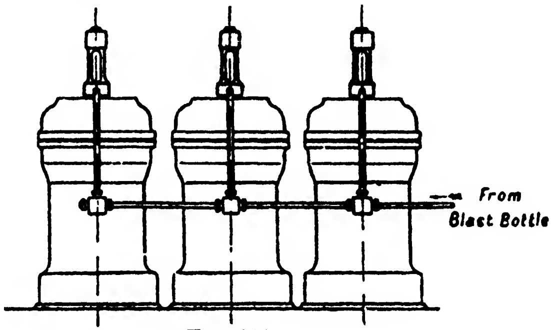


Fig. 280.

engines it is usual to provide a shut-down valve, as in Fig. 281, to each tee-piece, so that the supply to any individual cylinder may be cut off.

In addition, it is sometimes necessary (see Chapter XXIII) to provide a valve whereby the whole supply of blast air is automatically cut off when the manœuvring gear is put into the stop position. The bore of the blast air supply pipe to each cylinder need not exceed about 2% of the cylinder bore, but is usually greater than this in small engines, to avoid the multiplication of standard sizes of unions. The blast air main may be about 4% of the cylinder bore for any number of cylinders up to about six. The same type of union may be used as has already been illustrated in Fig. 207 in connection with the fuel system. Other types of union are in use, notably the Admiralty Cone Union, which is also very serviceable.

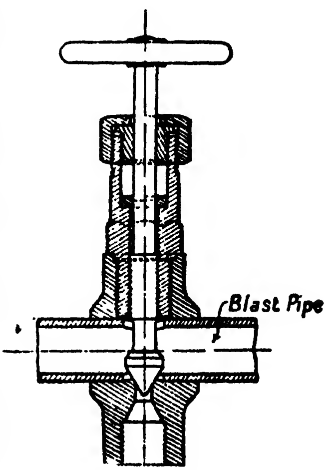


Fig. 281.

The Starting Air Pipe System.—With land engines it is quite common to provide two cylinders only with a starting valve when the number of working cylinders does not exceed

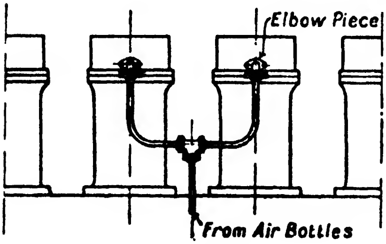


Fig. 282.

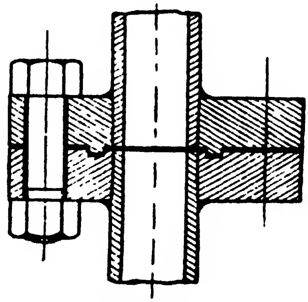


Fig. 283.

four. With six cylinders and upwards three or more cylinder units are provided with air-starting arrangements. A neat arrangement of the starting pipe is shewn in Fig. 282 for a four-cylinder engine. In this case the design of the starting

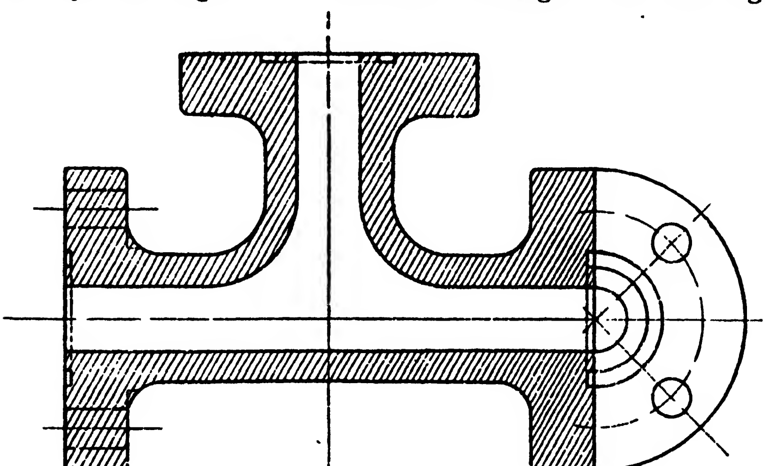


Fig. 284.

valve is such that air is admitted through a port cast in the side of the cylinder cover. Any arrangement of piping is to be avoided which renders difficult the removal of a cylinder cover, hence the provision of an elbow on the latter. In other designs this elbow is cast integrally with the cover itself.

With marine engines all cylinders are provided with starting valves, to which the air is led through a steel main pipe line running the whole length of the engine. Fig. 283 shews the type of pipe flange most commonly used, the material being steel. The tee-pieces for distribution to the several cylinders may be of cast iron or cast steel. If the former material is used, the design should be very substantial, as in Fig. 284. The pipe lines should be securely clipped to the framework of the engine, otherwise there is liable to be severe vibration, due to the surging of pressure within the pipe.

In large slow speed engines the diameter of the starting pipe may be about 0·07 to 0·1 of the cylinder diameter. In small high speed engines it is advisable to give the main distributing pipe a diameter of about 0·15 to 0·17 of the bore, in order to secure rapid acceleration.

Starting Valves.—These are usually located in the cylinder cover and operated by cams and levers, in the same way as the other valves. An arrangement less frequently used consists of a centralised air distributing box of rotary or other type remote from the cylinder covers but connected to them by distributing pipes. Loss of compression is obviated by the provision of non-return valves in the cylinder cover. The centralised distributing box may consist of a sleeve rotating in a casing in such a manner that the slot in the sleeve admits air successively to a number of ports communicating with the several cylinders. In other arrangements a set of cam-operated mushroom valves is used. A similar scheme is used to distribute air for servo-operated starting valves.

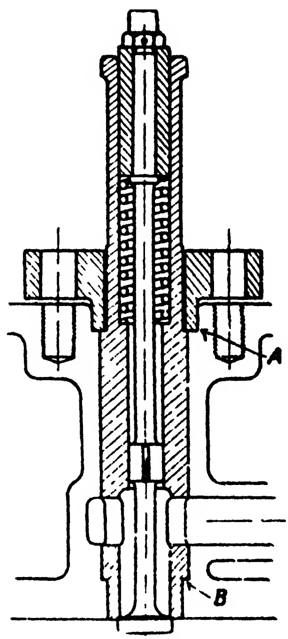


Fig. 285.

A common type of starting valve is shewn in Fig. 285. No provision has been made here to prevent leakage past the spindle, and if the latter is a good ground fit in the casing the leakage should not be serious in amount. In large sizes of valve additional tightness may be secured to advantage by the provision of a number of small Ramsbottom rings. It is usual to make the diameter of the piston part of the spindle the same as the smaller diameter of the valve head. The minimum spring compression should be equivalent to the maximum starting air pressure acting on an even area equal to that of the valve seat. The valve casing should be substantially proportioned, to prevent distortion and consequent leakage at the seat or binding of the spindle. It will be noticed that with this design of valve, joints have to be made at A and B simultaneously. There is no practical difficulty about this. Joint A is usually made with a copper or white-metal ring. A slight modification is sometimes made by the introduction of two cone joints, as in Fig. 286. This also works well. Fig. 287 shews a type of starting valve in which the air is led to the top of the valve casing, instead of being introduced through a port in the cylinder cover.

A useful type of starting valve, devised by the Burmeister and Wain Company for Diesel Marine Engines, is illustrated diagrammatically in Fig. 288. With this design the valve becomes inoperative when the air pressure is removed and resumes working as soon as the pressure is restored. This results in a great simplification of the manœuvring gear (see Chapter XXIII) by the elimination of mechanism which in some other designs is provided for the purpose of throwing the starting valves out of gear when the fuel is turned on. The desired result is achieved by attaching to the upper end of the valve spindle an air cylinder and piston, kept in constant communication with the air supply by means of holes through the spindle. In the absence of air pressure, the spring A is sufficiently strong to keep the piston B at the bottom of the cylinder C, thus removing the roller from the range of operation of the cam D. When pressure air is turned on the piston is forced to the top of the cylinder, and the valve remains operative so long as the force required to open the valve is less than the difference between the pressure load and the spring load on the piston B.

Diameter of Starting Valves.—On theoretical grounds, the necessary diameter of starting valves would appear to depend on the pressure of the air supply, amongst other things. It so happens, however, that in those cases where a low pressure air system is the most convenient (viz. in large marine installations) the multiplicity of cylinders to which starting air is

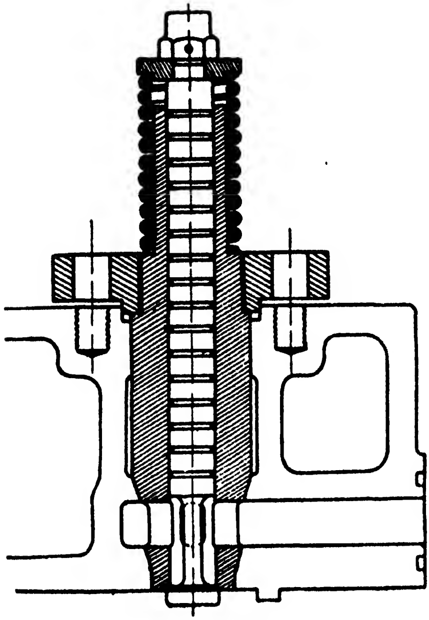


Fig. 286.

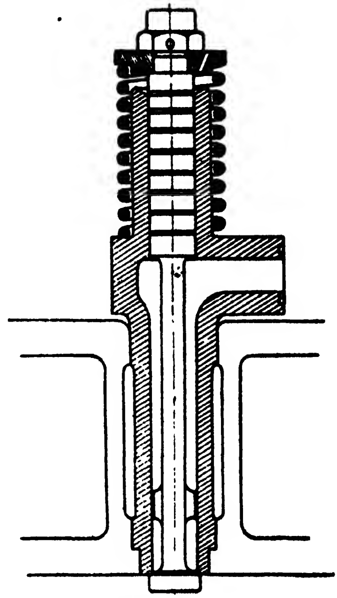


Fig. 287.

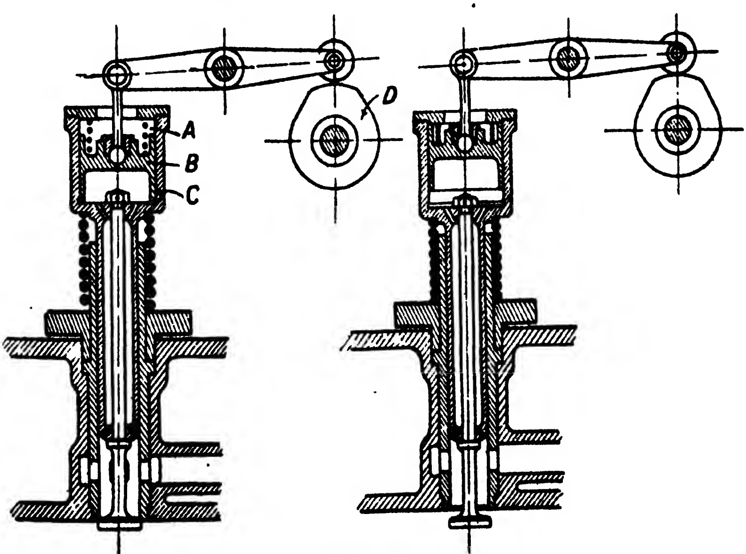


Fig. 288.

supplied affords adequate starting torque with a relatively low mean starting pressure in each cylinder. The result is that roughly the same diameters of starting valve are used in either case, i.e. whether a low or high pressure starting system be adopted, typical figures being from about 0·1 of the bore in the case of large engines to 0·13 in small engines.

With slow speed land engines it is very desirable to obtain a "fat" starting card, to overcome the inertia of the heavy flywheels which are usually necessary. An engine in good working order should start firing in the first or second revolution on

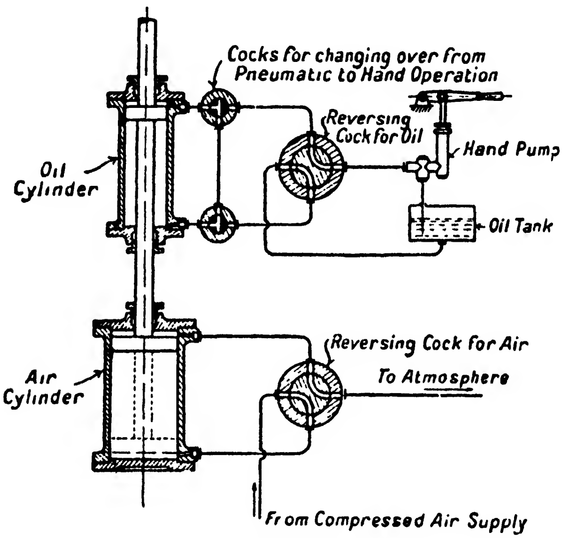


FIG. 289.

starting up cold. Airless injection stationary engines start up most readily from cold if starting air is not admitted to one or more cylinders.

Air Motors.—In large marine engines the work required to effect reversal of the valve mechanism when going from ahead to astern, or vice versa, is generally too great to be done with sufficient rapidity by a hand-gear, except in case of a breakdown of the air motor which is usually provided for the purpose. In different designs the air motor takes various forms, of which some are mentioned below :—

(1) A small reciprocating engine (double acting), with two cylinders and cranks arranged at right angles. The

arrangement is almost exactly similar to the small auxiliary steam engines used on steamships for the reversing gear or the steering gear. Low pressure air is used, and the air motor is geared down by worm and worm-wheel, so that it makes a considerable number of revolutions for one movement of the reversing gear.

(2) A single cylinder, with piston and rod, the reversing motion being performed in one stroke. This arrangement is suitable for high or low pressure air, and in either case the piston-rod must be extended into an oil dashpot cylinder to reduce shock. If the two ends of

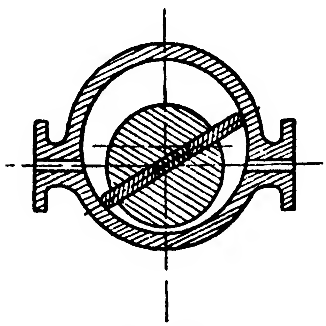


FIG. 290.

the oil cylinder be connected to a hand pump, the latter may be used for reversing, in the event of a failure of the air cylinder. This scheme is illustrated diagrammatically in Fig. 289.

The reciprocating motion of the piston-rod may be converted into rotary motion (one complete revolution or more) by a rack and pinion.

(3) A rotary engine of the type which is frequently used as a pump in connection with machine tools, motor-cars and other small machines, and which is shown diagrammatically in Fig. 290. This type of motor is only suitable for low pressures and is arranged to do its work in a considerable number of revolutions by means of worm gearing.

Types (1) and (3) would appear to have the advantage of greater adaptability to varying pressures. It is an easy matter to gear the motor down so that it will turn under the lowest air pressures anticipated. At higher pressures wire drawing at the ports prevents the attainment of an undesirably high speed.

In type (2) hydraulic leather packings are used, and considerable care should be taken in the design and the workmanship to eliminate all unnecessary sources of friction. The strict alignment of the two cylinders and the guided end of the rod deserve special attention.

Literature.—Ford, J. M., "Compressor Theory and Practice."—Constable.

CHAPTER XXIII

VALVE GEAR

Cams.—With few exceptions the valves are operated by external profile cams made of steel, hardened and ground on the face. In order to facilitate the removal of valves and cylinder covers, it is usual to arrange the cam-shaft to one side of the cylinders and to transmit motion from the cams to the valves through levers or a combination of levers and push-rods or links. The arrangements in general use give an approximate one to one leverage between cam and valve, and the figures

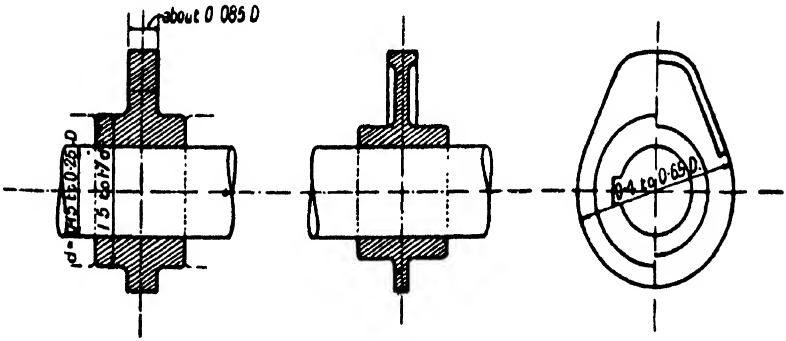


FIG. 291.

given above for the width of cam face are based on this proportion.

Two forms of cam body for the air and exhaust valves of four stroke non-reversible engines are illustrated in Fig. 291, the dimensions being expressed in terms of the cylinder bore. Fig. 292 shews a combined ahead and astern cam for a large marine engine. The bosses should be bored a hard-driving fit on the cam-shaft, and their lengths should be machined accurately to dimensions, so that the complete group of cams required for one cylinder give correct spacing when driven hard up side by side.

The fuel cam has to be of special construction, on account of the necessity for precise adjustment of the timing, and a typical form is shewn in Fig. 293. The toe-piece is preferably made of hardened steel, but chilled cast iron is sometimes used.

Profile of Cams.—The design of cam profiles for air, exhaust and scavenge valves is a matter of reconciling the claims of the following desiderata :—

(1) Rapid and sustained opening.
(2) Absence of wear and noise.

In slow speed engines the question of noise hardly arises, and wear is easily kept to a reasonable minimum by adequate width of face. For such engines the tangent cam shewn in Fig. 294 is suitable. For high speeds a smoother shape, as shewn in Fig. 295, is desirable, and the relatively slow opening may be compensated by earlier timing. Such profiles are easily drawn by deciding on some arbitrary smooth curve of roller lift, and plotting corresponding positions of the roller with respect to the cam, as in Fig. 296. Some designers favour a sinusoidal form of roller-lift curve. On theoretical grounds the cam profile should be based on the roller clearance circle, as in Fig. 297; this plan is adopted for high speed engines in the interest of noise reduction.

The starting air cams are best given a sudden rise on the opening side to minimise wire-drawing (Fig. 298).

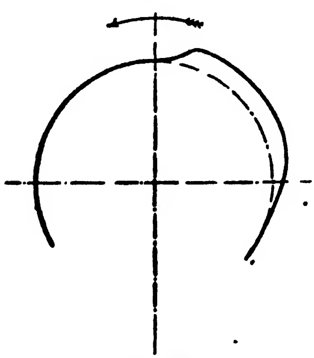


Fig. 298.

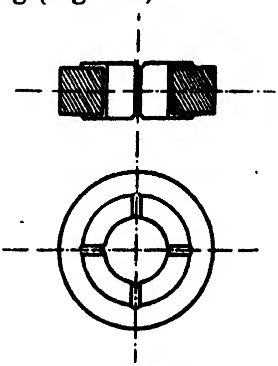


Fig. 299.

Blast air fuel cam profiles are a study in themselves, and the final decision rests with the test-bed engineers. The effective period is usually about 48 or 50 crank-shaft degrees,

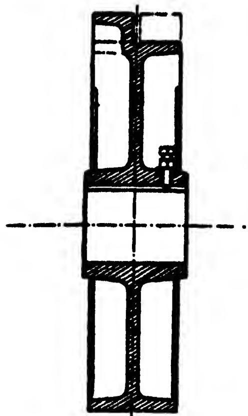


FIG. 292.

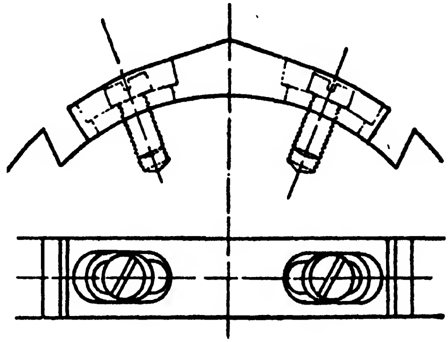


FIG. 293.

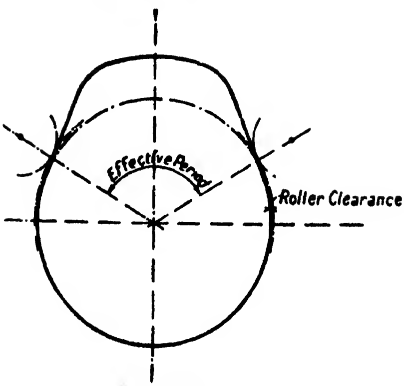


FIG. 294.

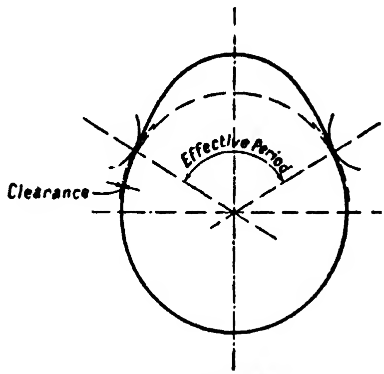


FIG. 295.

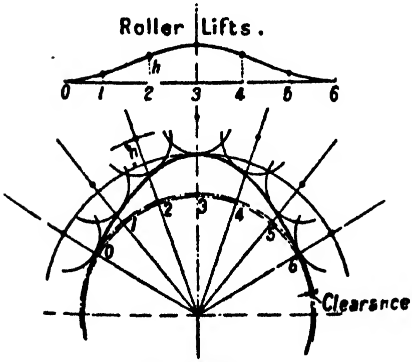


FIG. 296.

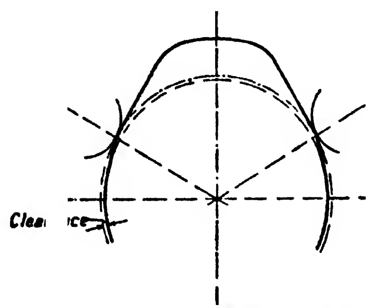


FIG. 297.

or 24 to 25 cam-shaft degrees. The cam-piece should, however, give a range about 25% in excess of this, after allowing for the normal roller clearance to allow for lost motion in the gear. The tangent profile shewn in Fig. 294 is usually found quite satisfactory, but other shapes are used.

Cam Rollers.—Engines having longitudinally fixed cam-shafts are usually provided with cam rollers of steel case-hardened and ground inside and out (Fig. 299) and having a diameter of about one-third that of the corresponding cams. The grooves provided for lubrication of the spindle should be noted. In marine engines in which reversal of rotation is effected by the provision of ahead and astern cams mounted on a longitudinally movable shaft, the rollers require to be of large diameter (about 60% of the cam diameter), in order that the idle cam may clear the lever, as in Fig. 300. Rollers of this size may be of cast iron bushed with phosphor bronze.

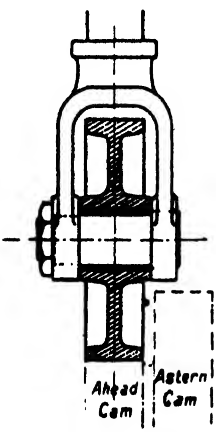


Fig. 300.

Valve Levers.—A common arrangement of valve levers and lever fulcrum shaft for four stroke land engines is shewn in Fig. 301. The fulcrum brackets are secured to the cylinder cover, and the latter may be lifted complete, with all valves and gear, and replaced without disturbing the valve settings. With the arrangement shewn it is necessary to lift away the fulcrum shaft and levers before the various valves can be removed for regrinding. This very slight inconvenience is sometimes overcome by means of split levers or by provision of horse-shoe shaped distance collars on the fulcrum shaft, which when removed leave sufficient space to allow the levers to be moved sideways clear of the valve casings. These devices are desirable in the largest engines only. Referring to Fig. 301 below, it will be noted that the fuel and starting levers are mounted on an eccentric bush A, connected to the handle B. The latter is provided with a spring catch engaging with notches in the fixed disc C, in accordance with the following scheme and the diagram shewn in Fig. 302.

Top notch.—Running.—Fuel lever in its normal running position. Starting valve roller out of range of cam.

Middle notch.—Neutral.—Both fuel and starting valve rollers out of range of cams.

Bottom notch.—Starting.—Starting air valve lever in its working position. Fuel valve roller out of range of cam.

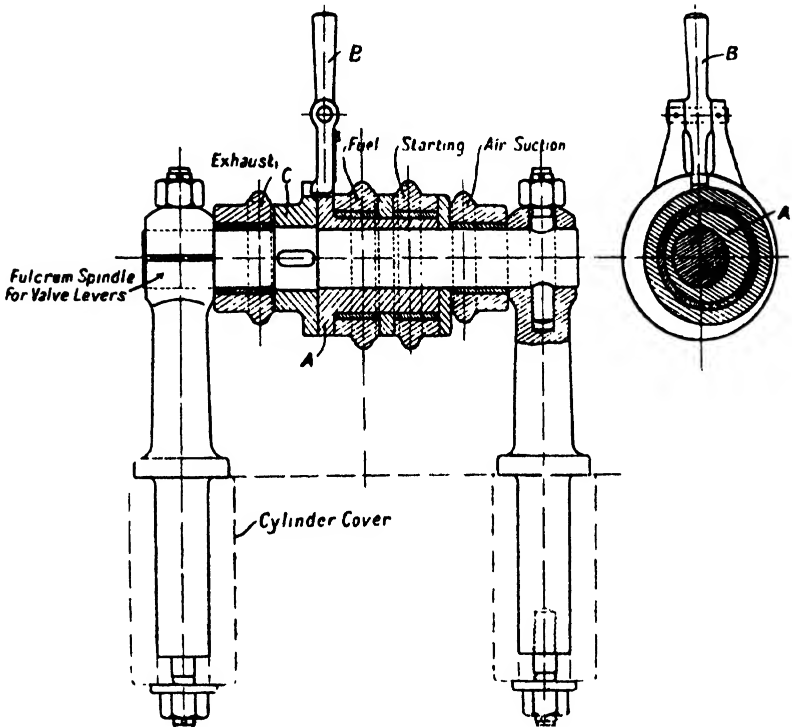


FIG. 301.

Sometimes this arrangement is modified by keying the eccentric bush and handle to the fulcrum shaft and allowing the latter to turn in its supports. This scheme is useful when the disposition of the gear is such that an air suction or exhaust lever separates the fuel and starting levers. This eccentric mounting of levers may also be used for exhaust lifting or to remove all the levers out of range of the cams during the axial displacement of the cam-shaft of a reversing engine.

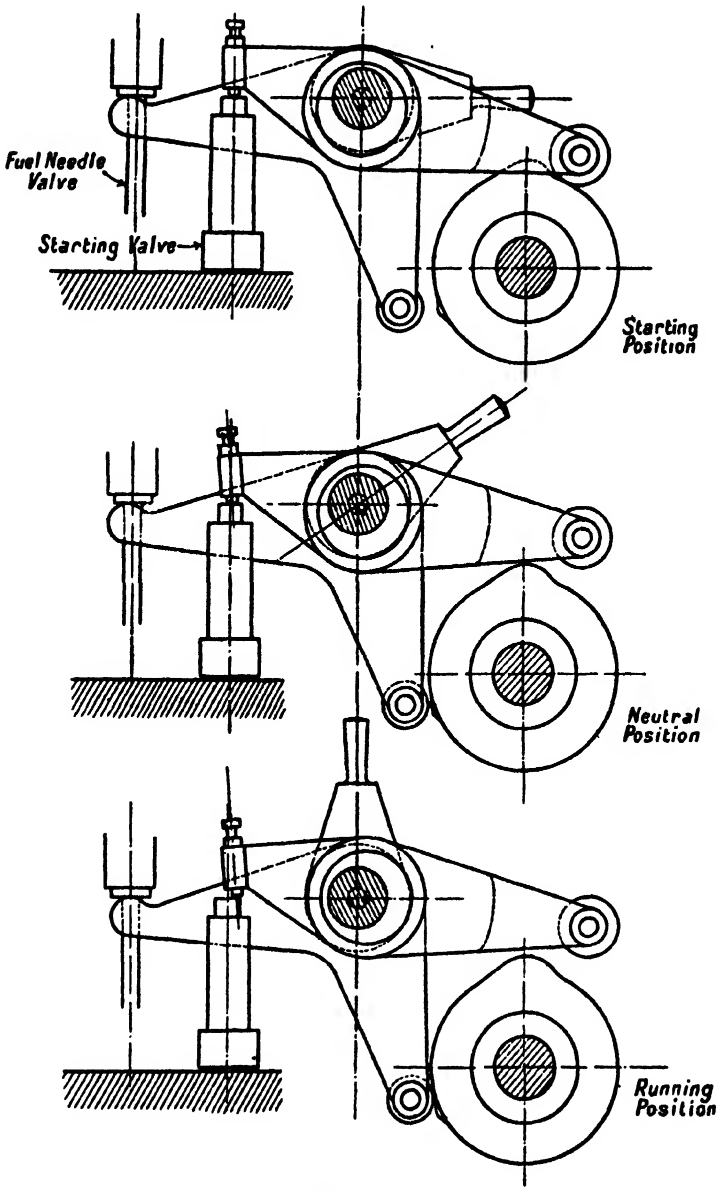

Fuel Needle Valve
Starting Valve
Starting Position
Neutral Position
Running Position

FIG. 302.

Certain well-known marine makers of great repute do not consider it necessary to put the fuel valve out of action whilst the engine is running on compressed air, and content themselves with suspending the supply of fuel to the valves during this period.

The levers are generally of cast steel or malleable iron; but good cast iron may be used if the stress is confined to about 2000 lb./in.^{2} Some alternative sections are shewn in Fig. 303. The forked end of the lever calls for very little comment. Type A (Fig. 304) is a good design, but expensive. Type B is very commonly fitted and is open to little objection. Type C is the cheapest existing construction, and if accurate castings (machine moulded) are obtainable the only machining operation required is to drill and reamer the hole for the roller-pin. The two grooves for the taper-pin may be cast.

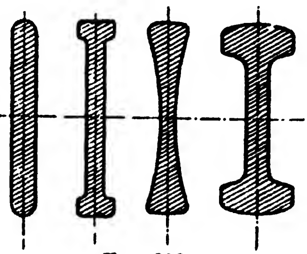


Fig. 303.

In small engines the tappet-end may consist of a plain boss screwed to receive a hardened tappet-screw and lock-nut, as in Fig. 307. In larger engines the more elaborate arrangements shewn in Fig. 308 are usually adopted. The bosses of

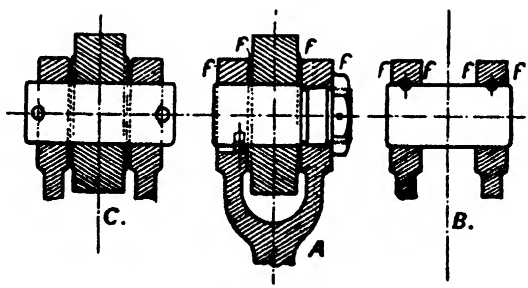


Fig. 304.

the levers should be bushed with good phosphor bronze and provided with a dustproof oil cup of some description, in order to reduce wear to a minimum. With these precautions the bush should only require renewal at widely distant intervals and means of adjustment are unnecessary even in the largest sizes of engines.

The valve lever system for a small or medium sized four stroke airless injection engine is shown in Fig. 305. The levers are made of drop forged steel of H section and the fulcrum bosses are bushed with bronze tubes. Each end of each lever is provided with a hardened ball-ended pin. One of these operates the valve through the intervention of a flat-faced

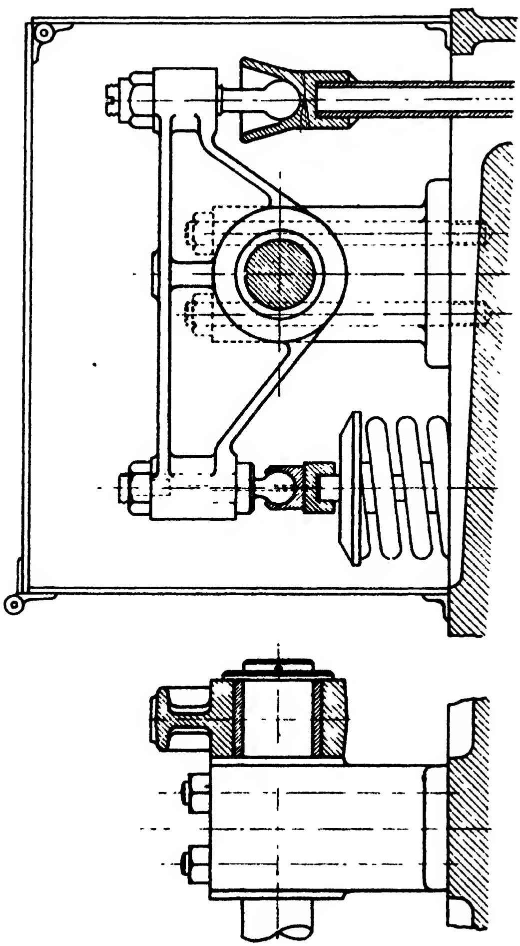


Fig. 305.

swivel piece and the other is screwed for the adjustment of clearance and is operated by a cup-ended push rod. If forced lubrication is used, provision must be made to drain oil back to the crankcase. Numerous modifications of this arrangement are in use; for example roller or needle bearing mounting of the levers with grease lubrication. With forced lubrication the levers may be drilled with small holes for conveying oil to the ball ends.

Fig. 306 shows two alternative methods of operating the push rods :—

(a) by roller tappet,
(b) by flat-faced cam follower.

If the roller tappet is used, the tappet body usually slides in a bronze bush in the frame, slotted to engage with the roller and keep its axis parallel to the camshaft. Since the thrust is entirely transverse, the sides of the frame bossing may be flattened to save end space and weight. Flat cam followers frequently take the form of high duty cast iron pots chilled on the under face and allowed to revolve.

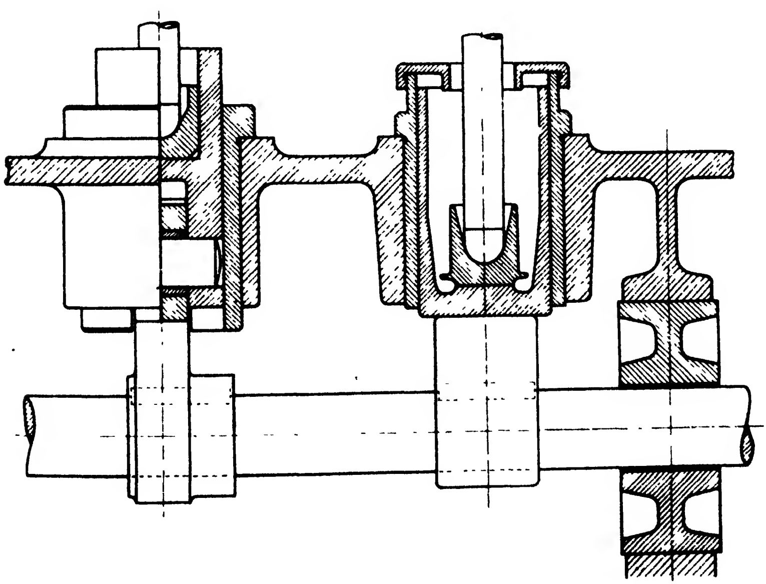


Fig. 306.

Strength of Valve Levers.—In four stroke engines the exhaust valve lever is the most heavily loaded. Although the force required to operate the suction valve is relatively small it is usual to make the inlet valve lever of the same section as the exhaust lever for the sake of uniformity of appearance, and

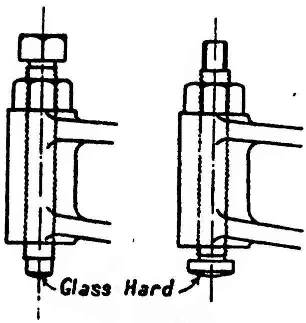


FIG. 307.

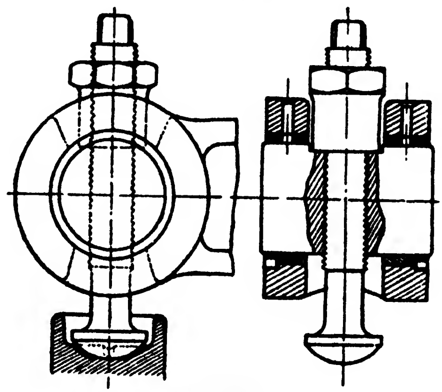


FIG. 308.

the same pattern may frequently be used for both. The loads imposed on the tappet-ends of the various levers at the points of valve-opening are given below :—

FOUR STROKE ENGINES

Exhaust Valve. About 45 lb. per sq. in. of exhaust valve area + spring load + inertia of valve.

Suction Valve. Spring load + inertia of valve + a maximum of about 5 lb. per sq. in. of valve area if the exhaust valve happens to be closing too early, due to excessive roller clearance.

Starting Valve. 500 lb. per sq. in. of valve area + spring load.

Fuel Valve. (a) Swedish type.

air injection:

1000 lb. per sq. in. of needle area + spring load + inertia, all reduced by the leverage employed.

(b) Augsburg type.

Difference between the spring load and 1000 lb. per sq. in. of needle area at stuffing-box.

Two Stroke Engines

Scavenge Valves. Spring load less scavenge air pressure into area of valve + inertia of valve.

Fuel, Exhaust and Starting Valves. As for four stroke engines.

All the above are of course subject to slight correction for friction.

By way of example, the main dimensions of the exhaust valve lever for a 20" four stroke cylinder are calculated below :—

Data :

Diameter of exhaust valve, 6·5 in.

Fulcrum spindle and exhaust valve lever centres, as in Fig. 309.

Pressure load on exhaust valve $= 0{\cdot}785 \times 6{\cdot}5^2 \times 45 = 1490$ lb.

Spring load at, say, 8 lb. per in.^{2} of valve area $= 0{\cdot}785 \times 6{\cdot}5^2 \times 8 = 265$ lb.

Inertia load, say . . 40 lb.

Total load to open exhaust valve . . 1795 lb.

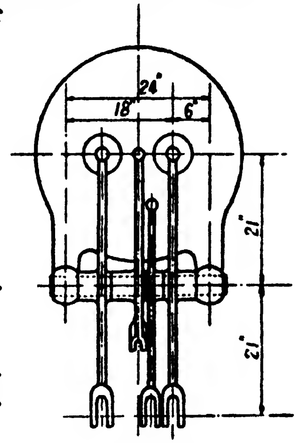


Fig. 309.

Reaction at fulcrum spindle, about 3600 lb.

Bending moment at fulcrum spindle $= \dfrac{3600 \times 6 \times 18}{24}$ in. lb.

Allowing a stress of 6000 lb./in.^{2},

$$\frac{d^3}{10} = \frac{3600 \times 6 \times 18}{6000 \times 24} = 2{\cdot}7$$

$$d = 3 \text{ in.}$$

Allowing for a bush ½" thick and about ½" metal at the boss the external diameter of the latter will be 5".

Sketching in the approximate outline of the lever, as in Fig. 310, it is seen that at the weakest section AA, the bending moment is about $1800 \times 18{\cdot}5$ in. lb., and taking a stress of

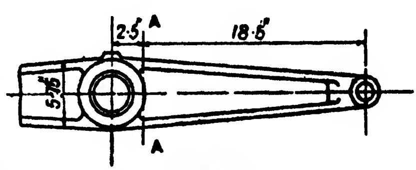


Fig. 310.

5000 lb./in. for cast steel, the modulus Z of the section AA should be $\frac{1800 \times 18{\cdot}5}{5000} = 6{\cdot}66$ in.^{3}

If the section AA is approximately I-shaped, as in Fig. 311. then Z=b.t.h. nearly. "h" is 5·75, and therefore

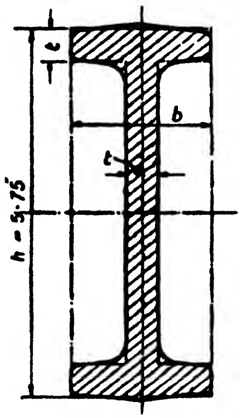


Fig. 311.

$$b.t = \frac{6{\cdot}66}{5{\cdot}75} = 1{\cdot}16 \text{ in.}^2$$

which is satisfied by b=2·25 and t=0·515″.

The lever may be made of approximately uniform strength by tapering towards the ends, both in width and depth, as in Fig. 312, whilst the flange and web thicknesses are kept constant. If a double bulb or other section is required, for the sake of appearance, and on casting considerations, it is a simple matter to sketch in such a section approximately equivalent to the simple I-section to which the calculation applies.

Push-rods.—In some designs a push-rod is introduced between the lever and the cam-roller, as shewn diagrammatically in Fig. 266 *ante*, in order to enable the cam-shaft to be located at a low level. For this purpose bright hollow shafting, or even black lap-welded steam tubes are suitable, if not too highly stressed.

For handy reference in designing such push-rods the following table, taken from Prof. Goodman's "Mechanics Applied to

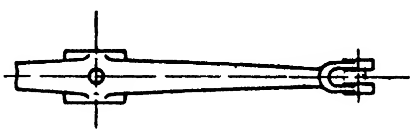


Fig. 312.

Engineering," is given for the buckling loads of tubular struts. In using these figures it is advisable to use a factor of safety of not less than about 3 or 4, and in no case to employ stresses exceeding 10,000 lb./in.^{2}

BUCKLING STRESS (FREE ENDS), LB. PER SQ. IN.

Ratio $\frac{\text{diameter.}}{\text{length.}}$	Mild Steel (Tubes).	Mild Steel (Solid).
10	59,000	56,000
20	42,000	33,000
30	29,000	21,000
40	20,000	13,000
50	14,000	9,000
60	10,500	7,000
70	8,200	5,200
80	6,500	4,000
90	5,500	3,000
100	4,500	2,500

The jointed ends of the rods may be of forged steel bar or malleable cast iron bushed with bronze, as in Fig. 313.

Cam-shafts.—In modern shops the cam-shaft may be rapidly and cheaply ground to size from black bars. In order to facilitate the driving on of the cams for a multi-cylinder engine the enlarged diameters are usually made of increasing sizes, differing by successive thirty-seconds of an inch, or thereabouts, as shewn exaggerated in Fig. 314. The same figure which represents the cam-shaft for a four stroke generating set of three cylinders also shews the method of supporting the shaft by means of a continuous trough with one bearing between each bank of cams. This arrangement has a very neat appearance and makes provision for catching the oil which drips off the cams and rollers. In some designs the cams are allowed to dip into an oil-bath, the level of which is maintained constant by a small pump provided for the purpose, or by a connection taken from the forced lubrication system. A copious supply of oil to the cams has the advantage of securing quiet running.

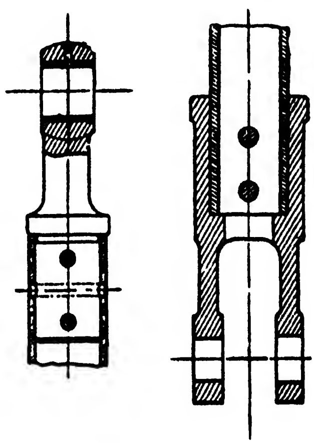


FIG. 313.

In other designs the cam-shaft is supported by bearing brackets secured to the cylinders as in Fig. 315. In this case it is very desirable to fit light cast or sheet iron guards round each bank of cams. The cam-shaft bearings are divided horizontally for adjustment, and the shells may be of cast iron

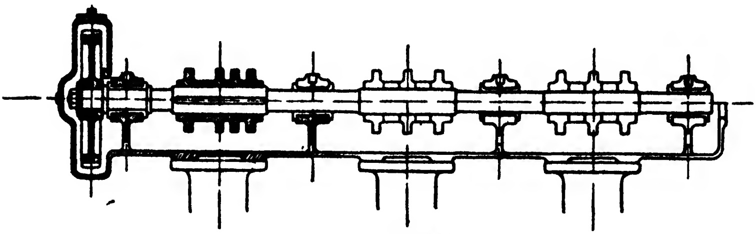


FIG. 314.

lined with white-metal, solid gun-metal, or in small engines, where the cost of material does not outweigh the advantage of simplicity, of solid die-cast white-metal. Owing to the slow peripheral speed and the intermittent character of the loading, grease lubrication has been successfully used in the past on slow speed engines. Forced lubrication is now usual.

A slightly different arrangement is shewn in Fig. 316.

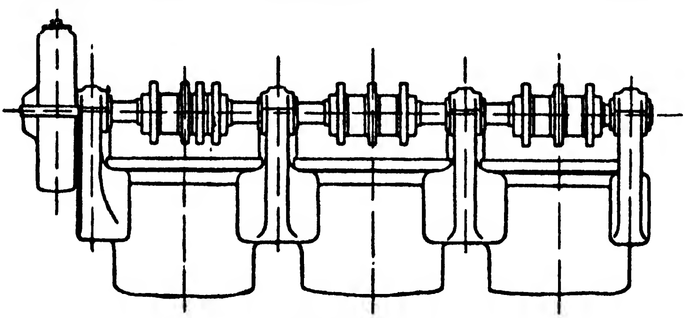


FIG. 315.

Here the shaft is supported by a series of cam-troughs, one to each cylinder, each trough having two bearings. The extra rigidity of this arrangement allows of the cam-shaft diameter being reduced below the figure required with the other arrangements described. This division of the trough into segments is advantageous from the manufacturing point of view as the smaller parts are easier to cast and handle in the shops ; also one pattern serves for engines of any desired number of cylinders.

With many designs, particularly of trunk engines with monoblock frames, the valves are most conveniently operated with push rods from a cam-shaft located in the crank-case or frame casting (see Figs. 7, 9 and 10).

Strength of Cam-shaft.—The size of cam-shaft required for a given engine would appear to depend not so much on the stresses to which it will be subject, as on the rigidity necessary to secure sweet running of the gear. For a four stroke engine the opening of the exhaust valve against the terminal pressure in the cylinder is the severest duty which the cam-shaft is

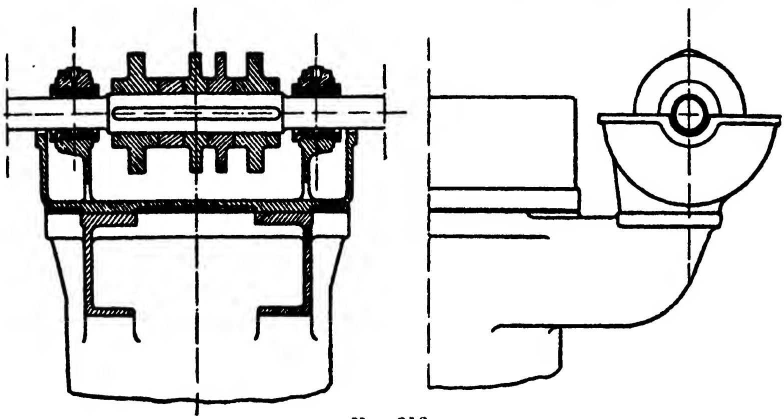


FIG. 316.

called upon to perform. The load is applied and released fairly suddenly, and a cam-shaft lacking in torsional and transverse rigidity would undoubtedly be subject to oscillations, which in an acute case would give rise to the following evils :—

(1) Noisy action of cams, due to torsional recoil of shaft after each exhaust lift.
(2) Interference with the timing of valves (particularly the fuel valves) of cylinders remote from the gearing end of the cam-shaft.
(3) Chattering of the gear-wheels by which the shaft is driven.

In view of the fact that as shaft diameters are increased the stiffness increases at a greater rate than the strength, it seems just possible that strength considerations might outweigh those of stiffness in very large engines. On the other hand, if angular deflection of the shaft between contiguous cylinders be accepted as the criterion, then considerations of similitude give shafts of diameters bearing a constant ratio to the cylinder bores (or rather exhaust valve diameters) and constant stresses in all sizes if the terminal pressure is always the same. The fact that in practice relatively thinner cam-shafts are used in large engines may be due to the lower terminal pressures and moderate piston speeds obtaining in the cylinders of the latter.

The following table shews the approximate diameters of cam-shafts used in practice on four stroke air blast injection engines of different sizes :—

Bore of Cylinder in inches	6	10	15	20	25	30
Diameter of Cam-shaft in inches	$1\frac{1}{2}$	$2\frac{1}{4}$	$2\frac{7}{8}$	$3\frac{3}{8}$	$3\frac{7}{8}$	$4\frac{1}{4}$

The above figures hold for any number of cylinders up to four with the cam-shaft drive at one end, or eight with the cam-shaft drive at the centre.

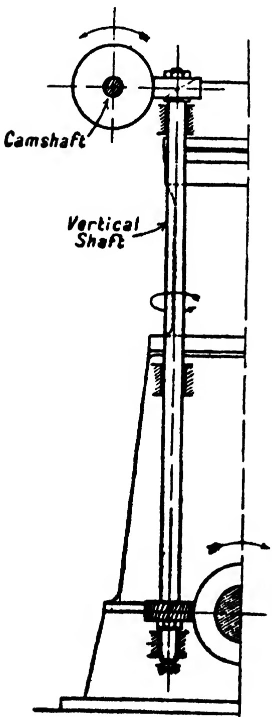


FIG. 317.

Stiffer cam-shafts are advisable if airless injection fuel pumps are driven from the cam-shaft ; the above diameters may be increased about 20 to 30%.

For two stroke engines the above diameters may be materially reduced in the absence of exhaust valves. Average figures for existing practice appear to be about 25% lower than those given above for four stroke engines. The fuel pumps and cylinder lubricating pumps are frequently driven off the cam-shaft, but any auxiliary gear, such as circulating pumps, etc., requiring appreciable power are best driven otherwise.

Cam-shaft Drives.—For non-reversible engines, the spiral drive shewn diagrammatically in Fig. 317 is widely used. This drive comprises the following components :—

(1) Lower spiral wheels.
(2) Footstep bearing for vertical shaft.
(3) Vertical shaft and couplings.
(4) Upper spiral wheels.

The lower spiral wheels generally have a 1 : 1 ratio, so that the vertical shaft runs at engine speed. In some designs the ratio is $1\frac{1}{2}$: 1, and the vertical shaft runs at 50% above engine speed. With the former arrangement the upper wheels have a ratio of 1 : 2 and with the latter 1 : 3.

The construction of the footstep bearing has already been commented on in Chapter XI. The vertical shaft is usually made the same diameter as the cam-shaft, or a trifle less, and for convenience in dismantling is sometimes made in two or three pieces, connected by couplings, examples of which are shewn in Fig. 318. The spiral drive has also been used for reversible two stroke engines, as it lends itself to a particular type of reversing motion, to be described later.

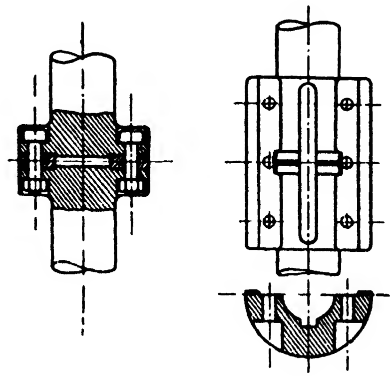


FIG. 318.

A combination of spiral and bevel drive is also used as shewn diagrammatically in Fig. 319, and leads to a compact arrangement of valve gear.

Reversible marine four stroke engines are usually fitted with some form of spur wheel drive, in order to enable the cam-shaft to be moved longitudinally, without undue complication. A floating cam-shaft with spiral drive necessitates the use of a splined seating for the upper spiral wheel.

The drive shown in Fig. 320 consists of a train of spur wheels from crankshaft to camshaft. With trunk engines having valves operated by long push rods the camshaft is located relatively low down, and one idler wheel generally suffices.

Fig. 321 shows a simple scheme of chain drive for a four stroke engine and Fig. 322 a more elaborate chain drive for a two cycle engine having a rotary blower running at about two or three times engine speed; a single chain drives the camshaft and the blower, but the final drive for the blower includes a spur wheel speed increasing gear.

Roller chains are in use for the camshaft and blower drives of the largest engines, including reversible marine engines, as well as small engines for road traction, etc. With small engines it is not unusual to arrange the chain drive at the end of the

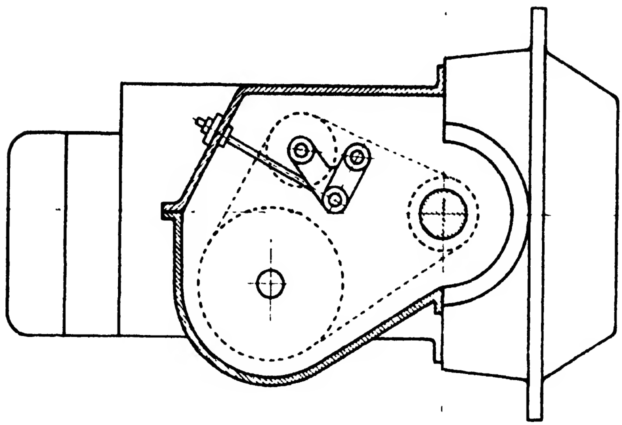


FIG. 321.

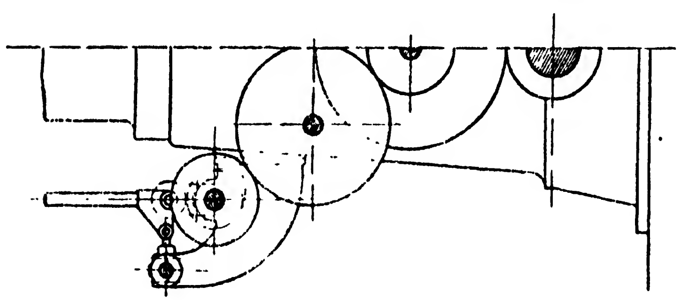


FIG. 320.

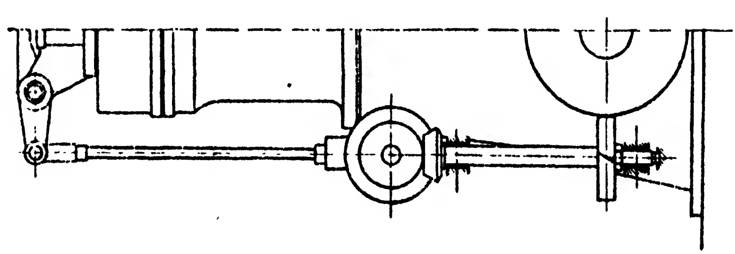


FIG. 319.

engine remote from the fly-wheel in spite of this being the position of greatest irregularity. With larger engines it is usual to arrange the chain drive at the flywheel end. For the larger sizes of marine engines in which flywheels are not fitted (apart from light turning wheels of negligible flywheel effect), the nodal point is at the middle of the engine, and it is here that the drive for camshaft and blower is fitted. Even so it is necessary to include flexible couplings and/or slipping clutches to mitigate shock at reversals.

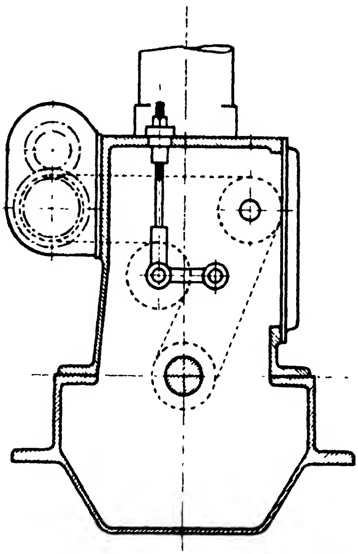


FIG. 322.

Spur and Spiral Gear for Cam-shaft Drives.—The question of the strength of the teeth hardly arises in this case, and the problem consists in the selection of materials and proportions giving quiet running and absence of wear. The following pairs of materials are in common use :—

	Driver.	*Follower.*
(1)	Cast iron.	Cast iron.
(2)	Steel.	Cast iron.
(3)	Steel.	Bronze.

Of these, the pairs (1) and (3) appear to give the best results, with proper proportions and adequate lubrication, etc.

In good practice, the normal circular pitch of the teeth is made about equal to one-twelfth of the cylinder bore for both spur and spiral gears, and the width of face about one-fifth of the cylinder bore in the case of four stroke engines. It is a fairly safe rule to make the pitch as coarse as the smallest wheel will allow in the case of spiral wheels. With spur wheels fine pitches are not so objectionable as the sliding between the teeth is much less.

For satisfactory running, the teeth must of course be properly cut and the wheels accurately centred. The tooth clearance should not exceed about 2/1000″, and should be uniform all round.

For particulars of tooth-gearing calculations the reader is referred to the special books devoted to this subject. The diagram shewn in Fig. 323 is very useful in the preliminary stages of spiral drive calculation. Suppose it is desired to design a pair of right-angle spiral wheels to say 1 : 2 ratio; first calculate the diameters of a pair of spur gears of the desired pitch and giving the desired ratio, viz. 1 : 2. Draw OA and OB equal to the pitch radii of the follower and driver respectively. Complete the rectangle OBCA and draw any line DCE, cutting the axes in D and E. Then DC and CE will be equal to the pitch radii of equivalent spiral wheels having

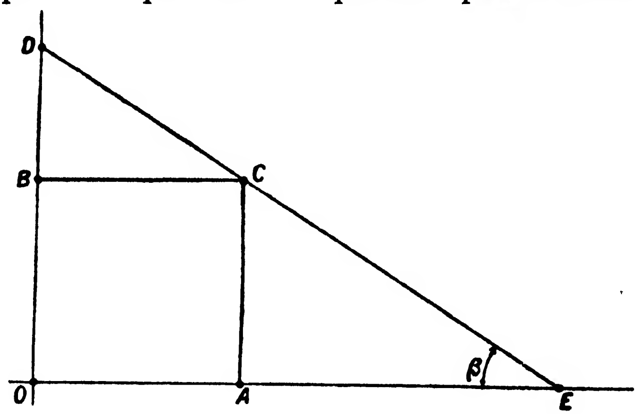


Fig. 323.

spiral angles α and β and having a normal pitch the same as the circular pitch of the spur wheels first calculated. It usually happens that the wheel centres (DE) are fixed within approximate limits by space considerations, and a process of trial and error is required to find suitable values for the number of teeth and the spiral angles. The latter should not be less than about 27°, as the efficiency falls off rapidly as this figure is reduced. Having obtained an approximate solution by the above method, the angles should be determined to the nearest minute by logarithmic trial and error calculation by means of the following relation :—

$$\frac{AC}{\sin\beta}+\frac{BC}{\cos\beta}=DE=\text{required wheel centres.}$$

Reversing Gears.—In spite of early anticipations of difficulty, reversing gears for Marine Diesel Engines have attained a high degree of efficiency. On the score of simplicity, reliability and quick action they compare favourably with the corresponding parts of steam engines. A very great number of different gears have been suggested and patented, but those in widespread use fall into two or three well-defined classes, which will be described below.

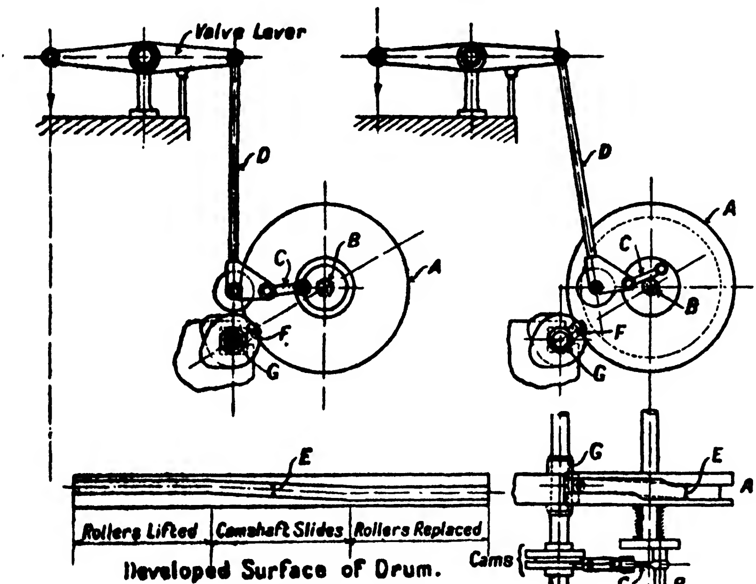


Fig. 324.

Sliding Cam-shaft Type of Reversing Gear.—This type of gear is the favourite for four stroke engines, though it is equally applicable to those working on the two stroke cycle. Ahead and astern cams side by side are provided for the operation of each valve. Reversal is effected by sliding the cam-shaft a few inches endways in its bearings, so that the ahead cam is removed from the action of the roller and replaced by the astern cam and vice versa. It is in general necessary to arrange means whereby the valve rollers may be swung clear of the cam noses during the longitudinal movement of the shaft, otherwise fouls would occur. In some very small engines the necessity for such provision is obviated by employing curved-faced rollers adapted to slide up and down inclined faces between the ahead and astern cams respectively.

The method adopted in some of the Burmeister & Wain engines is shewn in Fig. 324. A drum A is mounted on a cranked shaft B, on which are hinged drag links C, connected to the roller end of the valve push-rods D. Drum A is provided with a groove E, the developed shape of which is shewn in the figure. This groove accommodates a roller F, attached to a movable collar bearing G. Shaft B is rotated in the direction desired ("ahead to astern" or "astern to ahead") by

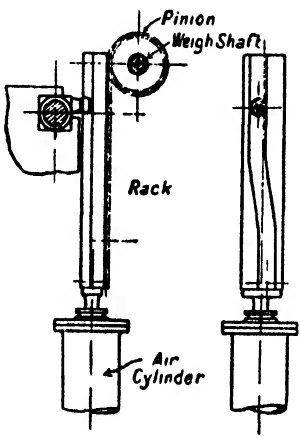


Fig. 325.

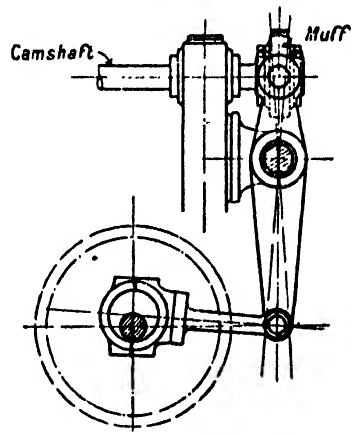


Fig. 326.

suitable gearing in connection with a reversing servo-motor or the like. Approximately one-third of a revolution of the shaft suffices to swing the rollers clear of the cams; meanwhile the cam-shaft is stationary. Another approximate one-third of a revolution causes the groove E to shift the cam-shaft from ahead to astern positions, or vice versa, whilst the valve rollers execute a harmless movement a little further out and back again. The remainder of the revolution of the weigh-shaft B replaces the rollers in their running position.

In other engines of the same make, the developed shape of the groove E is executed on the back of a rack by means of which the straight line motion of a vertical servo-motor is converted into rotary motion of the weigh-shaft. This variation is shewn in Fig. 325.

If separate means be adopted for removing and replacing the rollers, it is obviously possible to devise very simple means of shifting the cam-shaft, as in Fig. 326 for example. In such cases the two mechanisms must be interlocked to prevent a false manœuvre.

Experiments shew that the force required to move the cam-shaft longitudinally is about one-third of the weight of the cam-shaft, plus cams and other gear keyed thereto, and this figure may be used as a basis of calculation for this type of gear. It is advisable, however, to allow a fair margin of power, as the resistance to motion must always be a matter of some uncertainty. When the axial motion of the shaft has the

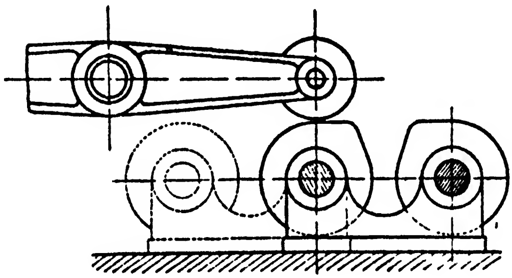


FIG. 327.

effect of opening one or more of the valves, the resistance due to this cause must be added to that of the shaft itself.

Twin Cam-shaft Type of Reversing Gear.—With this type of gear, once a speciality of the Werkspoor Company, not only are separate cams provided for ahead and astern running, but the latter are mounted on separate cam-shafts capable of being slid into and out of action as required. Fig. 327 illustrates the arrangement diagrammatically. The cam-shaft drive is usually by means of coupling rods. The chief advantage of this gear would appear to be the absence of special gear for swinging the rollers out of operation, this process being unnecessary. In recent Werkspoor engines this gear has been superseded by an arrangement of oblique eccentric bushes whereby the rollers are moved from the ahead to the astern cams and vice versa.

Twin Roller Type of Reversing Gear.—This gear depends on some form of link-work such as that shewn in Fig. 328. Rollers A and B lie in the planes of the ahead and astern cams respectively. In the position shewn the timing of the valve is controlled by the ahead cam and roller A; roller B is meanwhile outside the radius of action of its cam. Rotation of the weigh-shaft C through a predetermined angle throws roller A out of action and brings roller B into action with the astern cam. This type of gear has been applied to both four and two stroke engines.

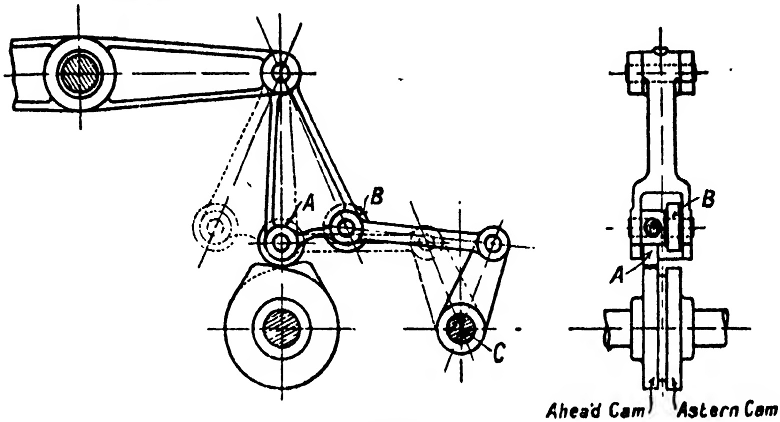


Fig. 328.

A different arrangement, having some slight resemblance to the above, is shewn in Fig. 329. In this case there is only one roller which is swung from the ahead to the astern cam by a motion in a plane at right angles to the plane of the gear. The roller face is curved to allow of this slight angular displacement from the vertical. The inherent defects of this mechanism probably render it unsuitable for use in conjunction with any but the air starting valves.

Selective Wedge Type of Reversing Gear.—This ingenious gear, illustrated diagrammatically in Fig. 330, has been devised by Carels Frères and used in connection with the starting air and fuel valves of two stroke marine engines designed by them. Ahead and astern cams are provided side by side, and the valve roller A is wide enough to cover both. Between the cams and the lever is interposed a roller-wedge piece B, under control of a cam C, mounted on a manœuvring shaft D. The latter is capable of independent rotary and endway motion. A suitable rotary motion of the shaft D withdraws the wedge B to an extent which renders inoperative the ahead cam on which it rests. A longitudinal movement of shaft D carries the

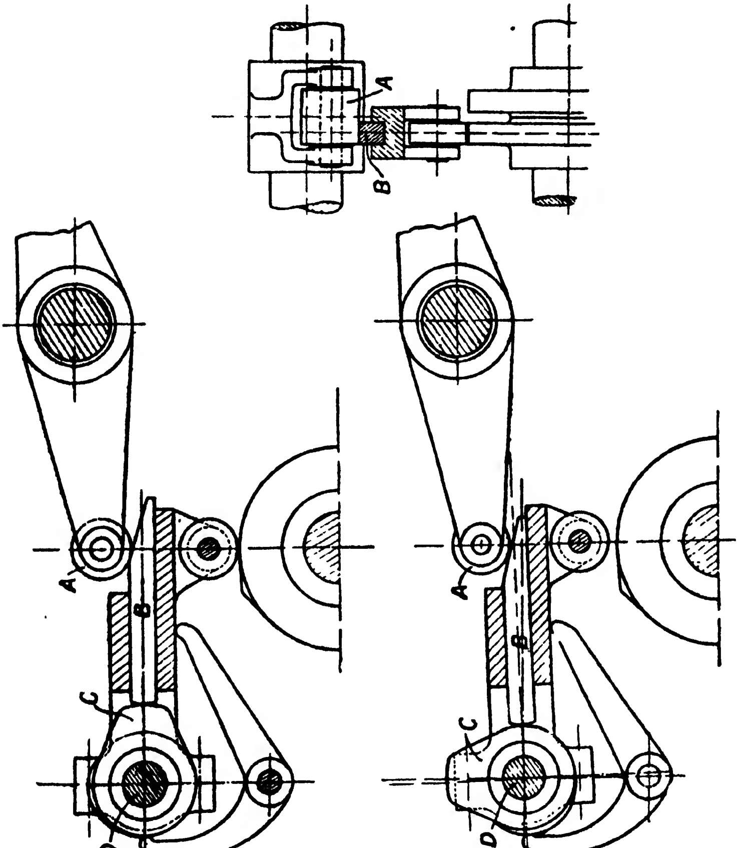


FIG. 330.

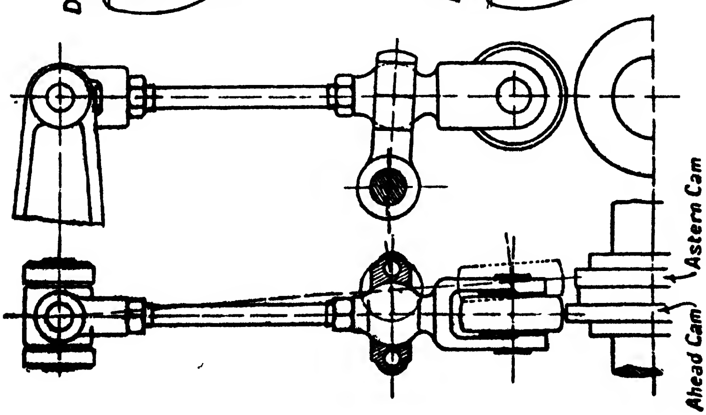


FIG. 329.

wedge B with it, and a further rotation of D introduces the wedge between roller A and the astern cam, and vice versa for astern to ahead. It is to be noticed that by a suitable arrangement of the durations and sequences of the cams by which the wedges are operated the engine is caused to start up in any predetermined manner, as for example :—

Position (1) Six cylinders on air. Fuel valves inoperative.
,, (2) Three cylinders on air. Three cylinders on fuel.
,, (3) Six cylinders on fuel. Air valves inoperative.

Special Reversing Gear for Two Stroke Engines.—With two stroke engines there are a number of means by which the duplication of cams may be avoided. Considering the case of

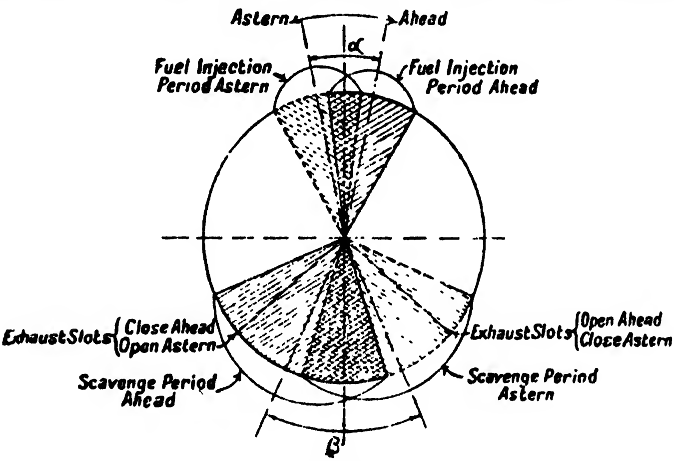


FIG. 331.

an engine fitted with scavenge valves and neglecting the starting air valves for the moment, the valve settings for ahead and astern will be somewhat as shewn in Fig. 331. It will be seen that for both fuel and scavenge valves all that is required to effect reversal is the rotation of the cam-shaft through a certain angle α for the fuel valve and β for the scavenge valves. In some early engines it was decided to select $\alpha=\beta=$about 30° to 35°, and so effect reversal of both valves by one movement. In later engines, however, it is more usual to use the rotation of the cam-shaft to reverse the scavenge valve only and adopt independent means, such as duplicate cams, etc., for the fuel and starting valves. The effect of the rotation of the cam-shaft on the settings of the latter must of course be allowed for in fixing the angular positions of the fuel and starting cams.

A simple method of effecting the desired rotation of the cam-shaft of a small engine is shewn diagrammatically in Fig. 332. Spiral drives are used and the vertical shaft is in two pieces, C and D, connected by a splined coupling permitting vertical movement of the upper half D. The vertical

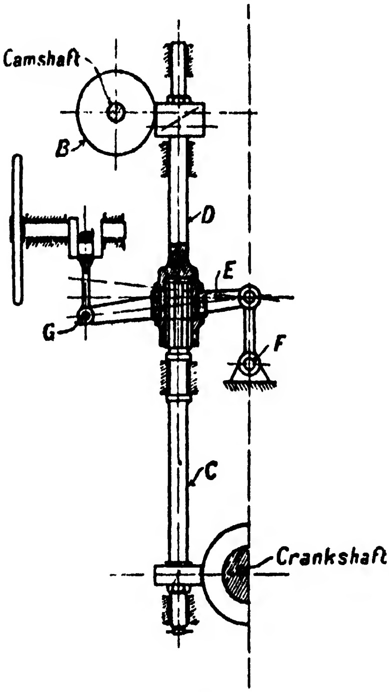


Fig. 332.

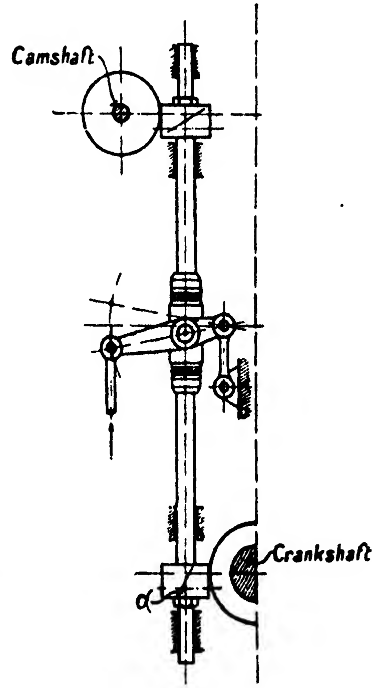


Fig. 333.

position of D is determined by the lever E, which is hinged at F and connected at G to an eccentric or other suitable means of transmitting motion from the hand-wheel. The extreme upper and lower positions of D determine the ahead and astern running positions.

If h = Lift of vertical shaft,
r = Pitch radius of wheel B,

Then $\frac{h}{r}$ = Reversing angle in radians;

In other arrangements the vertical shaft is moved as a whole, as in Fig. 333, and in calculating the amount of motion required for a given reversing angle it is necessary to take into account the rotation of the vertical shaft due to the sliding between the lower helical wheels.

Consider the case where both upper and lower gears have a ratio of 1 : 1. Let a be the spiral angle of the crank-shaft gear-wheel.

Note that $a < 45°$.

Let h = Lift of vertical shaft.

r_1 = Pitch radius of vertical shaft lower wheel.

r_2 = Pitch radius of cam-shaft wheel.

Then

Rotation of cam-shaft due to axial movement of vertical shaft $= \frac{h}{r_2}$ radians as before.

Further,

Rotation of vertical shaft due to sliding of lower spiral wheels $= \frac{h \cdot \tan a}{r_1}$

Therefore,

$$\text{Reversing angle} = \frac{h}{r_2} \pm \frac{h \tan a}{r_1}$$

With the arrangement shewn the positive sign applies when the upper and lower spirals have the same hand and the negative sign when they are of opposite hand.

An inspection of the valve settings for ahead and astern, as shewn in Fig. 331 *ante*, reveals the fact that the " reversing angle " is always described in the direction opposite to the previous direction of motion. Advantage has been taken of this fact to obtain self-reversing valve settings by arranging, between the cam-shaft drive and the cam-shaft proper, a claw clutch having angular clearance between the jaws equal to the reversing angle. With this arrangement independent reversible gearing must be used for the starting air valves. A suggested improvement on the above is to provide mechanical means for taking up the slack between the jaws whilst the engine is standing, instead of allowing it to be suddenly taken up on starting.

Manœuvring Gears.—In this connection the term manœuvring gear is applied to those mechanisms apart from the reversing gear which come into operation on starting up a marine engine.

The procedure differs in different designs, but in general the following remarks are applicable :—

(1) When the engine is standing the blast air supply should be cut off, to prevent accumulation of pressure in any cylinder the fuel valve of which happens to be open. If means be provided for putting the fuel valves out of operation in the stop position the blast cut-out is not so essential, but is still desirable as a safeguard.

(2) The blast air should be turned on automatically immediately the engine is started, although it is quite advantageous to provide an independent shut-off and regulating valve under the control of the engineer.

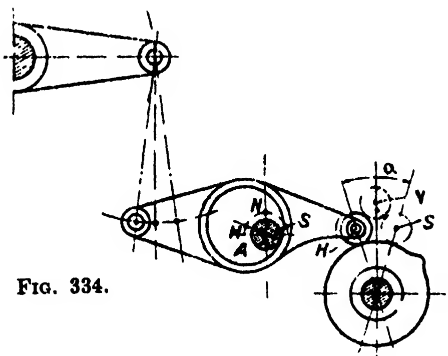


FIG. 334.

(3) When the engine is standing the starting air should be cut off, as there is otherwise great loss of air due to leakage past the starting valves. The starting air shut-off may be automatic or hand operated ; if the latter, it should be opened and closed by one simple motion. An ordinary high pressure globe valve, fitted with a quick-threaded spindle, is suitable for this duty.

(4) The fuel pump suction valves should be held off their seats until such time as the fuel valves are in running position, independent of the position of the fuel control.

(5) The fuel control should be a handle (not a wheel) with a wide range of movement between no oil and full oil.

(6) A wheel, or better still a lever, is provided in connection with suitable mechanism for putting the starting valves into

operation at starting, and subsequently putting them out of operation when sufficient speed has been attained to ensure firing in the cylinders. The same mechanism may or may not (in different designs) throw the fuel valve mechanism out of and into operation. Furthermore, in some designs the operation of this gear is graduated, as in the following scheme, which refers to a six-cylinder engine :—

First notch.—Six cylinders on air (starting).
Second notch.—Three cylinders on air. Three cylinders on fuel.
Third notch.—Six cylinders on fuel.

The gear under consideration is connected with the fuel pumps and with the blast starting air supply, so that the following conditions are secured :—

(a) Movement of the lever towards the starting position automatically turns on the starting and blast air.
(b) Suction valves of all fuel pumps held off their seats.

Further movement of the lever puts the starting valves of some or all of the cylinders out of operation, and simultaneously allows normal operation of the corresponding fuel pumps and also of the fuel valves if these latter are arranged to be out of operation during the time the starting valves are working.

(7) Some simple type of interlocking gear is usually fitted to prevent the following false manœuvres :—

(a) Starting the engine before the reversing gear is in either the full ahead or full astern position.
(b) Operating the reversing gear before the manœuvring gear has been put into the stop position.

Some of the means adopted to secure the conditions outlined in sections (1) to (7) will now be described. Further reference need not be made to the reversing gear, as with the exception of the interlocking arrangements mentioned above the reversing arrangements are entirely independent of the manœuvring gear.

A simple type of manœuvring gear is shewn diagrammatically in Fig. 335. The fuel and starting levers are eccentrically mounted on fulcrum shafts, as described earlier in this chapter. Each fulcrum shaft A is connected by links and levers to a manœuvring shaft B, under the control of a hand lever C. In the upper position of lever C all the fuel valves are in operation, and in the lower position the starting air valves. A link D connects the manœuvring lever to an eccentric fulcrum on the fuel pump, by means of which the suction valves are lifted by suitable tappets provided for this purpose, during such time as the starting air valves are in operation. Another link performs a similar operation on the blast air control valve, but in this case the connection is such that the blast air is only cut off in the neutral or stop position of the manœuvring lever.

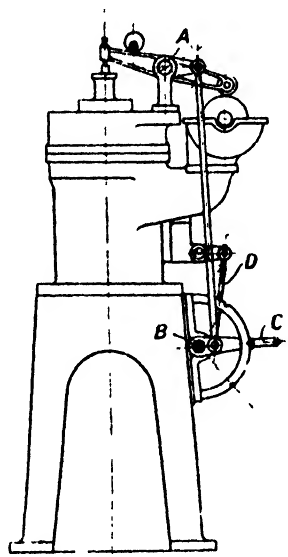


FIG. 335.

In some engines the above arrangements are adopted in principle, but two separate control gears and levers are provided for the forward and aft halves of the engine. The two control levers are placed close together, so that the engineer can work one with either hand. On starting he pulls both towards him, thus putting all cylinders under starting air. As soon as sufficient speed has in his judgment been attained, he pushes one lever towards the fuel position. If firing starts, he then pushes the other lever into the fuel position. If on the other hand firing does not ensue, he may pull back the lever into the starting air position and try the other lever in the fuel notch. With an engine in good order it is probably advantageous to throw over both levers simultaneously.

In other designs it is not necessary to operate the fulcrum shafts, as the starting valves automatically throw themselves into operation when starting air is turned on, and become inoperative when the starting air pressure is released.

A great deal of ingenuity has been expended on the design of gears for throwing successive combinations of cylinders from "air" to "fuel" positions by a continuous movement of a wheel. Some designers have even gone the length of combining the reversing and manœuvring mechanisms so that all positions ahead and astern are secured by clock-wise and anti-clock-wise rotation of this wheel.

Interlocking Gears.—The precise form which an interlocking gear takes in any design depends on the forms of mechanism adopted for the reversing and manœuvring gears respectively, but the problem very frequently reduces to that of two shafts, either of which shall only be capable of movement in prescribed positions of the latter. A simple interlock for two parallel shafts subject to partial rotation, is shewn in Fig. 336. It will be observed that the manœuvring shaft A can only be rotated when the reversing shaft B is in one of two positions (ahead and astern) defined by the positions of the gaps cut in the circumference of a disc keyed thereto. Furthermore, the shaft B can only be rotated from its ahead to its astern position (or vice versa) when the manœuvring shaft A is in one position—the stop position. The solution when the shafts are at right angles, as in Fig. 337, is equally obvious. An indefinite number of other schemes could easily be devised to meet the requirements of different arrangements of gear.

Hand Controls.—It is essential that all the wheels and levers by means of which the engine is controlled should be grouped together so that they may be manipulated by one

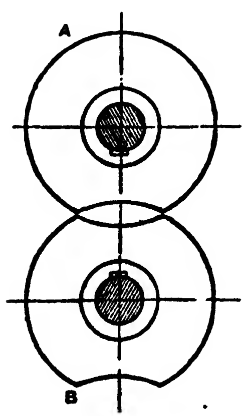


Fig. 336.

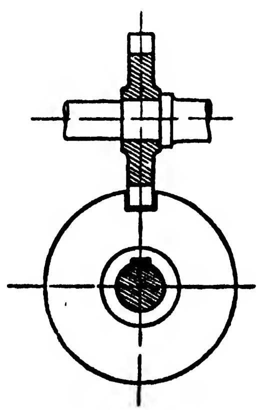


Fig. 337.

man in one position. In the best known designs there are a pair of long levers for controlling the groups of cylinders in starting and passing over on to fuel. In the Burmeister & Wain engines the same levers control the quantity of fuel delivered to the engine cylinders. In some other designs a separate lever or wheel is provided for this purpose. In almost every case separate wheels or levers are used for reversing. The usual positions for the control station are at the centre or forward end of the engine on the bottom or top platforms.

Literature.—Holmes, V., "Reversing Systems of Large Marine Oil Engines."—Inst. Automobile Engs., 1924.

CHAPTER XXIV

LUBRICATION

Forced Lubrication.—The principal bearings of most Diesel Engines are now forced lubricated. The system may briefly be described as follows (Fig. 338).

Oil from the bearings drains to the bottom of the crank chamber by gravity and is collected in a tank or depression called the sump. The suction of the lubricating oil pump is connected to the sump and a strainer of perforated metal is fitted to prevent rags or other matter being sucked into the pump. The pump discharges through a filter of fine wire gauze or other elements capable of trapping all solid matter above a certain small grain size. From the filter the oil passes to a cooler (if this is required) and thence to a pipe communicating with each main bearing. The crank-shaft and connecting rods are drilled to enable oil to pass from the main bearings to the connecting rod big end bearings and thence to the gudgeon pins. The pressure of oil in the distributing pipe depends on the rate at which the pump delivers oil, the viscosity of the oil and the resistance which is offered to escape of oil from the bearings. The pressure is commonly from 5 to 25 lb./in.^{2}. A pressure gauge is fitted to the discharge side of the pump to indicate satisfactory operation and give warning of failure.

Such a system, provided the pump discharge is adequate, ensures a copious supply of oil to each bearing and at the same time provides a means of carrying away the heat generated by friction of the bearings. The quantity of oil supplied is always greatly in excess of what is required for the formation of a lubricating film. Nevertheless the provision of a copious system of forced lubrication is no absolute guarantee of successful operation of the bearings. The oil pressure is a small fraction of the pressure which the bearings have to support and the formation of an adequate film depends on the design of the bearings in relation to the loading.

In large marine installations the sump takes the form of a double bottom tank to which the oil drains from the crank-pit and from which oil is pumped by a separately driven centrifugal or rotary oil pump. In smaller engines the sump may take the form of a small tank or vessel placed a foot or two below the

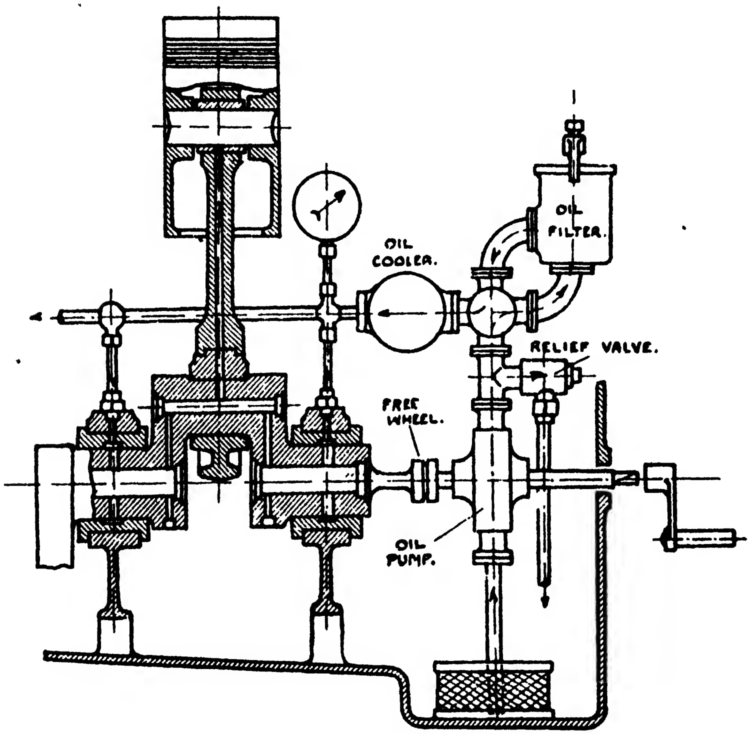


FIG. 338.

crank-case or merely a depression in the bedplate or bottom-half crank-case.

Duplicate filters are frequently fitted so that an element may be cleaned without stopping the engine or bye-passing the filter (Fig. 338.)

Lubricating oil coolers are sometimes dispensed with in relatively slow speed stationary engines up to about 300 B.H.P. when used in cold or temperate climates ; also with road vehicle engines in which the natural cooling of the bottom half of the crank-case may be sufficient.

In some engines oil is used for piston cooling and a common system may be used for this purpose as well as for lubrication. In such cases an oil cooler is essential and a larger oil pump is required.

It is highly desirable that the bearings be flooded with oil before starting an engine. If the oil pump is driven off the engine, this preliminary flooding may be done by a small hand pump. Alternatively means may be provided for turning the engine-driven pump by hand. This is easily accomplished by fitting a free wheel device in the pump drive and extending the pump spindle to receive a removable handle.

In course of time the lubricating oil becomes contaminated with solid matter too fine to be trapped by the filter. There are two ways of dealing with this :—

(1) Removing the oil, completely or in batches, from the system for treatment in a cleaning plant consisting of centrifugal purifiers or fine grained filters.

(2) Continuously passing a small flow of oil through purifiers or fine filters.

Lubricating Oil Pumps.—For large marine installations vertical centrifugal pumps or rotary displacement pumps are commonly used. The discharge, not including standby units, amounts to about :—

25 to 40 tons per 1000 B.H.P. when the oil is used for piston cooling as well as lubrication,

12 to 16 tons per 1000 B.H.P. when used for lubrication only.

Stationary engines, small marine engines and engines for rail or road traction are fitted with lubricating oil pumps driven off the engine. The well-known cog-wheel pump is a favourite form and may be driven directly off the forward end of the crank-shaft or by chain from the crank-shaft or by gearing from the cam-shaft. It is not advisable to mount the pump above the level of the crank-shaft otherwise difficulties may be experienced with the suction. The pump is sometimes immersed in the sump in quite small engines. This position is too inaccessible for engines of large or medium size. The capacity, for lubrication only, is about 0·8 to 1·4 tons per hour per 100 B.H.P. The pump discharge should be provided with a relief valve discharging to suction or sump to prevent excessive overpressure when starting up with thick cold oil. The relief valve is preferably set to a fairly high pressure of 30 to 60 lb. per sq. in. to prevent starvation of bearings when the oil is cold and viscous and the pressure gauges should have ample overload capacity.

Oil Filters.—Apart from special proprietary designs, lubricating oil filters usually consist of well-supported fine wire gauze diaphragms, with interstices measuring about 5/1000″ across, suitably housed. The greater the area available for flow the less the resistance and the longer the time which may elapse between cleanings. The gross area available for flow (i.e. interstices + wires) may be upwards of ½ in.^{2} per B.H.P. The filtering elements must be easily removable for cleaning without interruption to the oil flow. For this purpose a bye-pass is sometimes arranged; alternatively a duplicate filter may be supplied with change-over valves.

Oil Coolers.—When oil is used for piston cooling the cooler should be capable of transferring about 8% of the total heat supplied by the fuel, whilst maintaining the lubricating oil at the desired temperature. The oil outlet temperature from the pistons should preferably not exceed about 140° F. The oil temperature leaving the cooler may be about 120° F. under tropical conditions. If local circumstances with stationary engines render these temperatures difficult of attainment, they may be exceeded by about 30° F if a suitable oil is used.

When oil is used for lubrication only, and circumstances render a cooler necessary, it is advisable to reckon with a heat transfer of about 3% of the total heat supplied by the fuel. This is usually an over-generous estimate of the heat generated by bearing friction, but the oil picks up heat in other ways. The temperature of oil leaving the cooler need not be less than 120° F., and if a suitable oil is used may be as high as 145° or even more in circumstances where cold water is not available for cooling. Stationary engines of about 300 B.H.P., having moderate piston speed and mean pressure, working 12 hours a day in a temperate climate with cooling water entering at, say, 90° F., can work without oil coolers and without the oil exceeding a temperature of about 120° F. Sometimes a compromise is made for moderate duty by fitting a simple cooler capable of transferring about 1% of the total heat supplied.

One form of such a simple cooler consists of a "hair-pin" oil pipe introduced into the cooling water manifold, or the water jacket.

The usual type of oil cooler for dealing with considerable rates of heat transfer consists of a bundle of tubes with water flowing through the tubes and oil flowing transversely across them (Fig. 339). Small tubes with narrow spaces are more

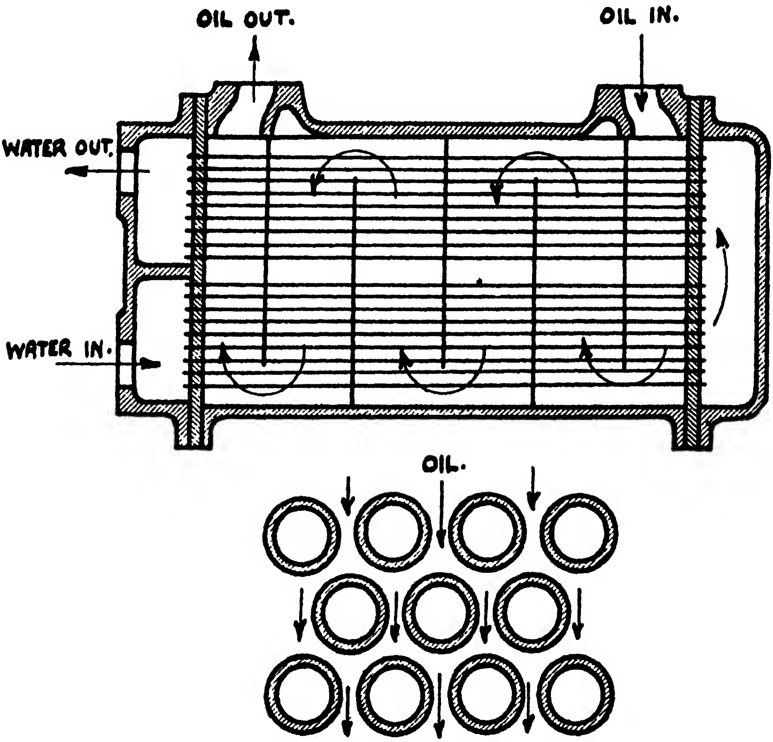


FIG. 339.

effective than large tubes with wide spaces. With the same ratio of space width to diameter, the coefficient of heat transfer varies inversely as the tube diameter to the power of about $\frac{1}{3}$, so that $\frac{1}{2}''$ O.D. tubes with $\frac{1}{8}''$ spaces are about 26% more effective than 1″ tubes with $\frac{1}{4}''$ spaces. The coefficient of transmission depends upon the velocity of oil in the spaces, the viscosity of the oil, and the velocity of the water. The water speed has little influence provided it exceeds about 2 ft./sec.

Assuming that the width of spaces between tubes is a constant fraction, say $\frac{1}{4}$ of the tube diameter, the equation for the heat transmission coefficient on the oil side takes the form :—

$$\frac{\alpha d}{k} = \text{const.}\left(\frac{Vd}{\upsilon}\right)^{\frac{2}{3}}\left(\frac{\upsilon}{h}\right)^{\frac{1}{3}} \text{approx.} \quad \ldots\ldots\ldots(1)$$

Where, α = transmission coefficient.
d = tube diameter.
V = oil speed measured at minimum space —d/4.
υ = kinematic viscosity of oil.
k = conductivity of oil.
h = diffusivity of oil (see p. 80).

Assuming k and h approximately constant for lubricating oils we have :—

$$\alpha = \text{const.}\ (d)^{-\frac{1}{3}}.(V)^{\frac{2}{3}}.(\upsilon)^{\frac{1}{3}}$$

Within the practical range υ is nearly proportional to the Redwood number.

Experiment shows that with :—

$$\left.\begin{array}{l} d = \frac{1}{2}'' \\ V = 4 \text{ ft./sec.} \\ \upsilon \text{ corresponding to Redwood No. } 1 = 150 \end{array}\right\} \text{then } \alpha = 120 \text{ approx.}$$

A reasonable value of α with the above data to allow for a certain amount of fouling and incrustation is

$$\alpha = 70 \text{ B.T.U./ft.}^2 \text{ hour deg. F.}$$

Example 1.—A 500 B.H.P. stationary engine consumes 0·38 lb. of fuel per B.H.P./hr. The cooling water inlet temperature is 140° F. and the outlet 160° F.; 20% of the total heat is absorbed by the water. The lubricating oil flow is 6 tons per hour and absorbs 3% of the total heat; specific heat of oil 0·5. It is required to find the surface area of a cooler to maintain an oil temperature of 155° at the cooler outlet.

Total heat supply	$0{\cdot}38 \times 500 \times 18{,}000$	$= 3{\cdot}42 \times 10^6$	B.T.U./hr.
Heat to water	$0{\cdot}20 \times 3{\cdot}42 \times 10^6$	$= 0{\cdot}68 \times 10^6$	,,
,, ,, oil	$0{\cdot}03 \times 3{\cdot}42 \times 10^6$	$= 0{\cdot}103 \times 10^6$	,,

Water flow $= 0{\cdot}68 \times 10^6 \div (160-140) = 34{,}000$ lb./hr., i.e. 6·8 gals./B.H.P./hr.

Temperature rise of water in cooler $103{,}000 \div 34{,}000 = 3°$ F.

Mean water temperature in cooler 141·5° F.

Drop of temperature of oil in cooler

$$=\frac{103{,}000}{0{\cdot}5\times 6\times 2240}=15^\circ \text{ F.}$$

Mean oil temperature $155+7{\cdot}5=162{\cdot}5^\circ$ F.

Mean temperature difference $162{\cdot}5-141{\cdot}5=21^\circ$ F.

Required surface of cooler $=\frac{103{,}000}{70\times 21}=70$ ft.^{2}.

Example 2.—A marine engine of 6000 B.H.P. consumes 0·36 lb. of fuel per B.H.P. hr. The cooling water inlet temperature in the tropics is 95° F., and the flow of water is 250 tons per hour. The lubricating oil flow which is also used for piston cooling amounts to 200 tons per hour, and absorbs 8% of the total heat supplied. It is required to find the surface area of a cooler to maintain an oil temperature of 125° F. at the cooler outlet.

Total heat supply $0{\cdot}36\times 6000\times 18{,}000=39\times 10^6$ B.T.U./hr.

Heat to oil $=0{\cdot}08\times 39\times 10^6=3{\cdot}1\times 10^6$ B.T.U./hr.

Temperature rise of water in cooler,

$$3{\cdot}1\times 10^6\div(250\times 2240)=6^\circ \text{ F.}$$

Mean water temperature in cooler $=98^\circ$ F. (i.e., $95+3$).

Drop of oil temperature in cooler,

$$\frac{3{\cdot}1\times 10^6}{0{\cdot}5\times 200\times 2240}=14^\circ \text{ F.}$$

Mean oil temperature $=125+7=132^\circ$ F.

Mean temperature difference $132-98=34^\circ$ F.

Required surface of cooler,

$$\frac{3{\cdot}1\times 10^6}{34\times 70}=1300 \text{ ft.}^2\text{.*}$$

It is not strictly correct to use the arithmetic mean temperatures of oil and water in evaluating the mean temperature difference, but the error in the above two examples is small. Correcting factors are tabulated under :—

$R_1=$	1	2	3	4	5	6	7	8	9	10
$R_2=$	1·00	0·96	0·91	0·87	0·83	0·80	0·77	0·75	0·73	0·71

$R_1=$ Ratio of maximum to minimum temperature difference between fluids.

$R_2=$ Ratio of true mean temperature difference to arithmetical mean temperature difference.

* For 6000 B.H.P. a more usual allowance of oil cooler surface is about 1500 to 2000 ft.^{2} giving a wider margin against incrustation, fouling, thicker oil, etc.

For Diesel locomotives oil coolers are frequently fitted of a type similar in all essential respects to motor car radiators. These are usually designed and built by specialist firms.

Forced Lubricated Bearings.—Bearings supplied with the bare minimum of oil required to maintain a film against continuous end leakage require pockets and grooves to distribute the oil to best advantage. To some extent pockets and grooves interfere with the formation of a film of maximum thickness by

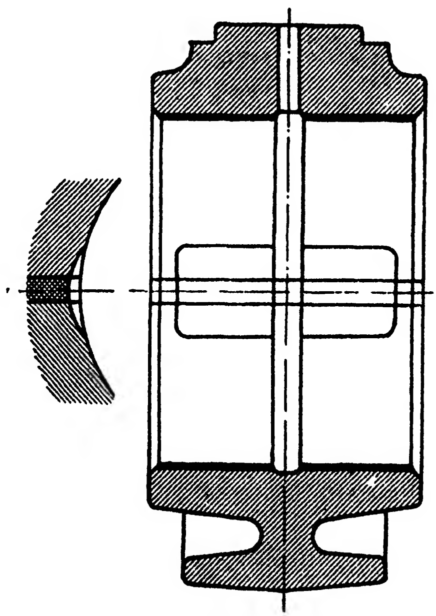


Fig. 340.

providing additional paths for the loss of the oil pressure, generated by the dragging in of oil by viscous adhesion. With forced lubricated bearings advantage should be taken of the copious oil supply to reduce pockets and grooves to the minimum consistent with adequate distribution.

Earlier practice consisted in providing each bearing with,

(a) Side pockets,

(b) A complete circumferential groove plain or staggered (see Fig. 340).

This arrangement provided for continuous communication of all bearings with the source of supply. Unfortunately the circumferential groove greatly reduces the resistance offered by the film to being squeezed out under pressure. It is therefore becoming increasingly common practice to curtail the circumferential groove in connecting rod big ends, but small side pockets appear to be desirable for the purpose of distributing the oil supply over the width of the bearing. With this arrange-

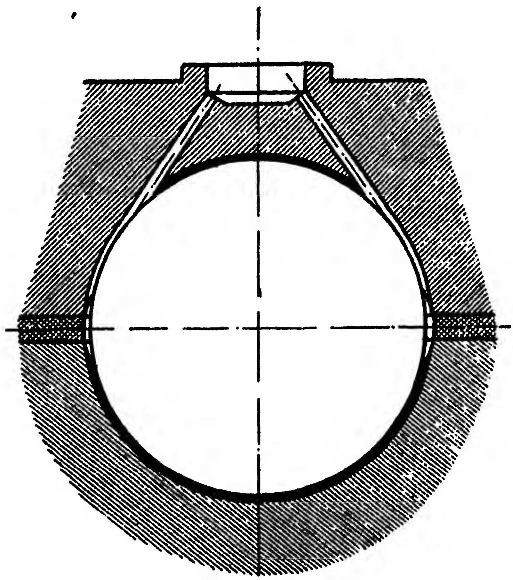


FIG. 341.

ment the oil supply to the big end is intermittent, being a maximum when the oil hole in the crank-pin coincides with a groove in one of the side pockets. The oil supply to the connecting rod small end may pass through a single hole at the crown of the bearing or a pair of holes communicating with the side pockets (see Fig. 341). The oil supply both to the big and small ends of the connecting rod is materially reduced by these arrangements as compared with the earlier system, but experience seems to shew that on small and medium size engines, at any rate, the oil supply is sufficient. Furthermore the reduction in oil supply to the connecting rod helps to reduce the oil consumption of trunk engines. The omission of the circumferential oil grooves in the main bearings would still further reduce the flow, but there would appear to be some advantage in interrupting the circumferential oil groove at the points of maximum pressure. In some examples the top main bearing bush is grooved and the bottom left plain.

Considerations of torsional vibration of crank-shafts tend towards the use of journals of large diameter and narrow width (see Fig. 342). A circumferential groove in the bearing reduces the surface to a pair of narrow bands from which oil is more easily squeezed out by pressure, than with wider bearings. Some mitigation of this condition by curtailment of grooving is worth consideration.

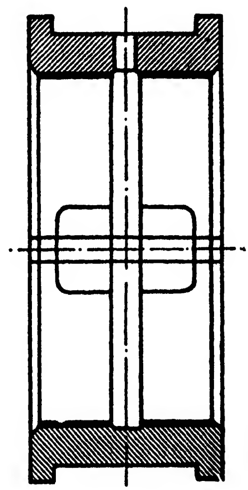


FIG. 342.

With slow revolution main bearings of large diameter it is usual to provide some oil grooves in a longitudinal direction to prevent dry patches. Such provision would appear to be particularly appropriate in two stroke single-acting engines in which the pressure is mainly downwards. Longitudinal grooves (in conjunction wi h a circumferential groove) are also used for gudgeon-pin bearings, the grooves being sometimes spaced at an angular distance apart equal to the angular swing of the rod (Fig. 343). All longitudinal grooves should be well rounded at the edges to prevent scraping of the oil film.

In many of the above respects practices differ widely in different designs and the above notes are indications of the considerations involved.

Cylinder and Piston Lubrication.—Many trunk engines running at about 300 R.P.M. (possibly less) and upwards rely entirely upon oil thrown from the connecting rod or oil mist for cylinder lubrication. Slow running trunk engines and all crosshead engines are fitted with positive sight feed lubricators, usually supplied by specialists. There is great diversity of practice with regard to the number and location of the quills, i.e. fittings at which the oil is introduced to the surface of the cylinder liner.

The following are representative :—

Four stroke S.A. engines	One to six quills per cylinder with or without circumferential distributing grooves in liner. Quills are level with second piston ring from top at bottom dead centre.
Two stroke S.A. engines	As above with quills sometimes arranged above level of cylinder ports.
Double acting engines	Four to eight quills per cylinder arranged at two levels about half-stroke above and below centre of cylinder.

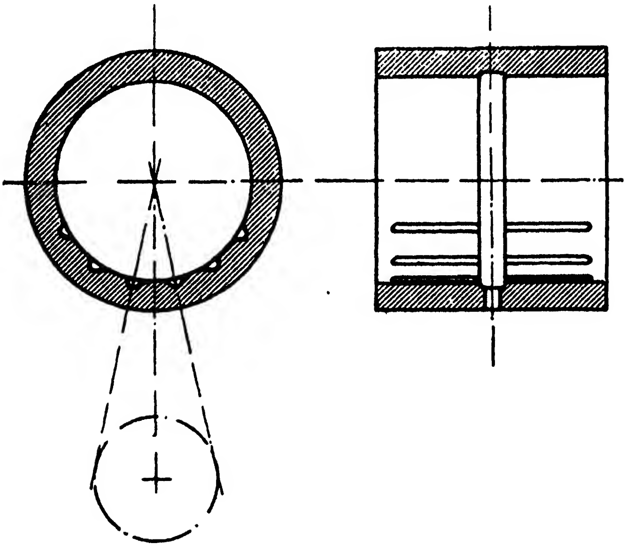


Fig. 343.

Cam-shaft Lubrication.—Cam-shaft bearings are usually forced lubricated. In large marine engines the cams require little oil and may be hand lubricated. In smaller engines the cam-shafts may be totally enclosed and the cams allowed to dip into an oil bath. If the cam-shaft is housed in the crank-case the cams may receive sufficient oil by splash from the connecting rod and crank-webs. Roller pins and the like are sufficiently lubricated by capillary action provided there are sufficient grooves or slots to give access to the wearing surfaces and the neighbouring metal is drenched in oil. The intermittent loading of such pins gives rise to a pumping action in the clearance spaces between the rubbing surfaces and this action gives a sufficient supply of oil to the surfaces provided the above conditions are satisfied.

Lubrication of Valve Gear.—Inlet, exhaust and starting air valve spindles, also valve levers, require a certain amount of lubrication. If the valve gear is exposed a few drops of oil may be applied from time to time by hand, preferably through dust proof oil cups at each point. If the valve gear is totally enclosed the valve levers may be forced lubricated, in which case it is advisable to make arrangements to catch the oil extruded from the lever bushes and lead it back to the crank-case; the aim being to keep the cylinder tops as dry as possible. If the valve levers are mounted on roller or ball bearings it is only necessary to provide a grease gum connection for use as required. The starting valves can be given a few drops of oil from time to time by hand. Inlet and exhaust valve spindles may be catered for by drip feeds outside the casing and connected by pipes to the required spots.

Miscellaneous Gear.—Governor gear, indicating gear, cam-shaft driving gear and all other working parts should be carefully considered from the point of view of lubrication, leaving nothing to chance. A connection to the forced lubrication system is usually the best source of supply. If splash from other moving parts is relied upon, arrangements must be made to guide the oil to each rubbing surface (see remarks under cam-shaft lubrication). It is unwise to provide oil holes which can be blanked off by the accidental turning of a bush, even if a locking pin is provided. This contingency is easily avoided by turning a groove in the outside of the bush (see Fig. 344). Small pins are preferably case-hardened and ground.

Theories of Lubrication.—The type of journal bearing which has been most fully studied in theory and experiment is the so-called half bearing subtending an angle of 30° to about 180° and subject to a constant load. For any such bearing it can be shewn that the relation between the coefficient of friction μ, the pressure intensity P, the angular speed n and the viscosity Z takes the form,

$$\mu = f\left(\frac{nZ}{P}\right)$$

This relation is independent of absolute size so long as relative proportions remain unchanged and the difference between bearing radius and journal radius is proportional to either.

With this proviso the same formula applies to any design of journal bearing, even if subject to loading variable in intensity and direction, provided the frequency of the cycle of pressure variation varies as "n." The formula shews that angular speed is of fundamental importance rather than rubbing speed.

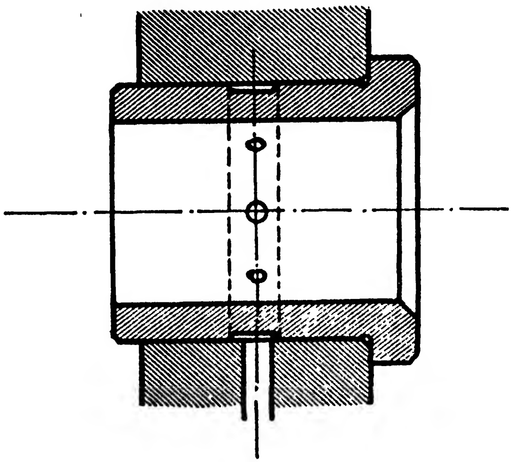


FIG. 344.

Fig. 345 shews graphically a typical relation for a fully lubricated half bearing under constant load. The region A to B is unsafe and seizure is liable to occur as a stable oil film is not fully established. The region beyond B toward C is the safe region corresponding to a stable film separating the "rubbing" surfaces. As the film passes through the bearing the viscosity of the oil forming it is reduced by temperature rise and the viscosity Z should refer to a mean value of the viscosity of the oil forming the film. Supposing the point "K" refers to particular valves of n, Z and P, applying to a particular bearing. If now n be increased (say doubled) whilst P remains constant, the rate of generating heat by friction will be increased and Z will fall, so that the value of $x = nZ/P$ will be less than doubled. It is conceivable that the reduction of Z might be such that x would actually be decreased, but this does not appear probable. With a constant high value of P the bearing is more likely to seize at low revolutions than high. With reciprocating machinery increased revolutions eventually lead to increased values of P, due to inertia and centrifugal force, which vary as n^2, so that if n is steadily increased the value of x will at first increase, then reach a maximum and then decrease again.

It is unfortunate that complete journal bearings subject to pressure varying in direction and/or magnitude have not been much studied under laboratory conditions, but there seems to be no doubt that the successful operation of such bearings under very high pressures in reciprocating machinery is due to the periodical partial refilling of each part of the bearing clearance when the thickness of the clearance at that part is

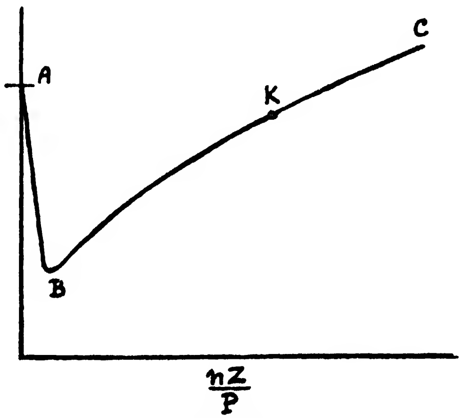


FIG. 345.

momentarily increased by the fluctuation in the direction of resultant pressure. Even in two cycle single acting engine main bearings, in which the pressure is mainly outward from the cylinder, there are forces which vary the direction of resultant pressure through a certain angle, and such bearings under forced lubrication work excellently.

With gudgeon pin bearings the angular variation in the direction of pressure is small in the case of two cycle single acting engines; two ways of ensuring lubrication in these circumstances are :—

(1) Supplying the oil at a pressure (200 to 300 lb./in.^{2}) in excess of the minimum pressure on the bearing.

(2) Providing longitudinal grooves with a pitch equal to the movement, so that a film is drawn over each element of surface.

Lubricating Oils.—Builders of Diesel Engines frequently issue specifications giving the viscosities at various temperatures, the flash points, and other physical properties to be expected of oils suitable for the various services of their engines. Guided by such specifications Diesel users find by experience the grades of oil which give the best all round results, taking cost into consideration. The following notes give merely a rough indication of some of the properties of oil which are found suitable.

The following notes refer to straight mineral oils.

Specific gravity is an indication of origin ; paraffin base oils (Pennsylvanian) have a specific gravity at 60° F. from about ·87 to ·90 ; asphaltic base oils have a S.G. of about ·90 to ·95 at the same temperature and viscosity ; S.G. increases with viscosity in each case.

The specific gravity decreases with temperature at the rate of about 3% per 100° F.

The specific heat at 60° F. varies from about 0·45 to 0·50 for different oils. It increases with temperature at the rate of about 13% per 100° F. ; the value 0·5 is near enough for most ordinary calculations in the absence of more exact data.

Viscosities of lubricating oil vary over a wide range. For cylinder lubrication of large double acting engines, oils of high viscosity are required. Single acting engines even of large diameter up to 30″ or so can use less viscous cylinder oils. For trunk engines up to about 15″ diameter a single oil is frequently used for cylinders and bearings.

Fig. 346 shews viscosity curves for six representative lubricating oils, and the table below is a rough guide to their application :—

Type of Engine.	R.P.M. × $\sqrt{\text{B.H.P.}}$ per cyl. up to	Cylinders.	Bearings.	Double Purpose.
Double acting . .	6000	A, B	D	—
Crosshead single acting ..	3000	C, D	D	D
,, ,, ,, .	2000	D	F	D
Trunk engines . .	7500	—	—	D
,, ,, . .	4000	—	—	E
,, ,, . .	2000	—	—	F

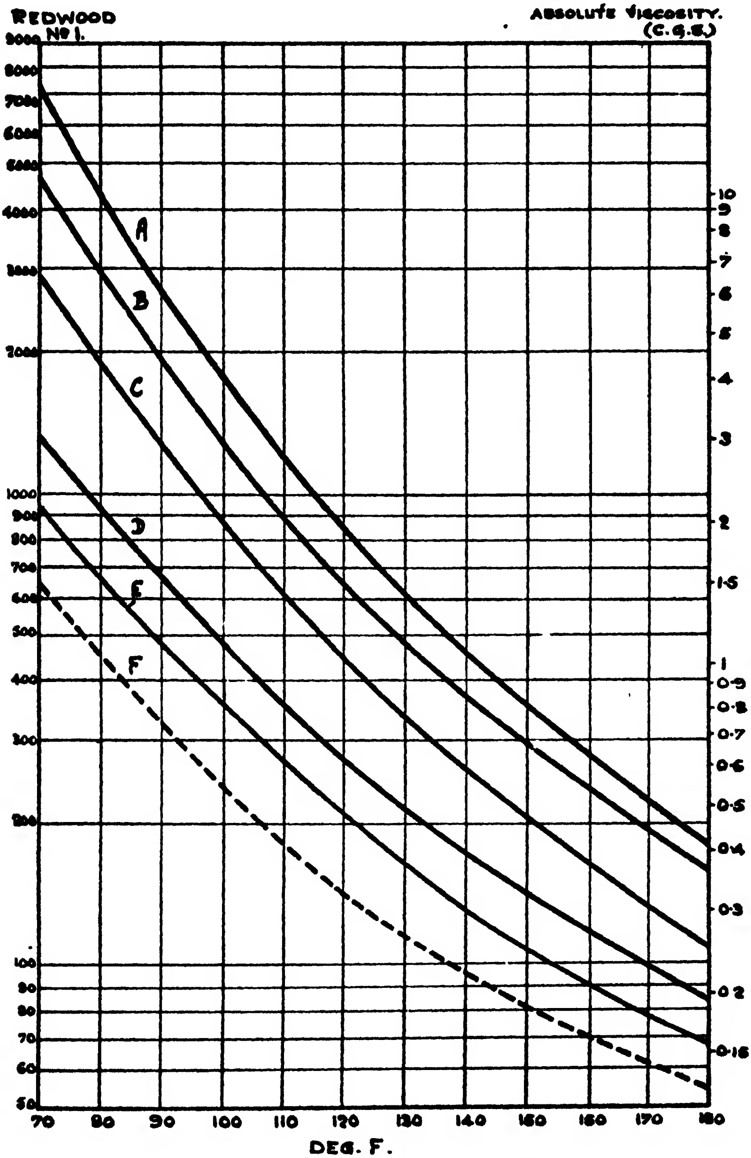

REDWOOD No 1.
ABSOLUTE VISCOSITY. (C.G.S.)
9000
8000
7000
6000
5000
4000
3000
2000
1000
900
800
700
600
500
400
300
200
100
90
80
70
60
50
10
9
8
7
6
5
4
3
2
1·5
1
0·9
0·8
0·7
0·6
0·5
0·4
0·3
0·2
0·15
A
B
C
D
E
F
70
80
90
100
110
120
130
140
150
160
170
180
DEG. F.

FIG. 346.

If the temperature of the bearing oil leaving the cooler exceeds about 130° F., oils of higher viscosity may be used.

Piston cooling oil is usually the same as the bearing oil.

Literature.—Smith, P. H., "Diesel Engines, excessive Lubricating Oil Consumption."—Constable.

Boswall, R. D., "The Theory of Film Lubrication."—Longmans, Green.

CHAPTER XXV

THE COOLING WATER SYSTEM

Heat Flow to Cooling Water.—The amount of heat transferred to the cooling water of a Diesel Engine amounts at full load to about 18 to 25% of the total heat supplied by the fuel. If the exhaust manifold is water cooled the latter figure may be increased to about 30%. Adopting the figure 25% and a fuel consumption of 0·37 lb./B.H.P. hour the amount of heat carried away by the cooling water at full load is

$$0{\cdot}25 \times 0{\cdot}37 \times 18{,}000 = 1660 \text{ B.T.U./B.H.P. hour.}$$

At no load fuel is consumed at about $\frac{1}{6}$ of the full load rate and about $\frac{2}{3}$ of the heat value ultimately finds its way into the cooling water. The heat rejected to cooling water at no load therefore amounts to about

$$\frac{1660}{6} \times \frac{0{\cdot}67}{0{\cdot}25} = 740 \text{ B.T.U./hr. per full load B.H.P.}$$

This is about 45% of the full load amount. These and approximate figures for other loads are tabulated under :—

Load.		B.T.U./hr. per full load B.H.P.
0	no load	740
$\frac{1}{4}$	quarter	970
$\frac{1}{2}$	half	1200
$\frac{3}{4}$	three-quarters	1430
1	full	1660
$1\frac{1}{10}$	10% overload	1890
$1\frac{1}{5}$	20% overload	2120

The distribution of heat among the cylinders, covers, pistons, etc., has been considered in Chapter XV; here we are chiefly concerned with the total amount.

The Flow of Cooling Water.—The rate of water circulation depends on the temperature rise admissible or desirable.

The following table gives the flow in galls. per minute per full load B.H.P. to deal with 2120 B.T.U./hr. per full load B.H.P. @ 20% overload, for various values of the temperature rise.

Temperature Rise F. deg.	Water Flow in galls. per hr. per full load B.H.P.
10	21·2
15	14·1
20	10·6
25	8·5
30	7·1
35	6·1
40	5·3
50	4·2
60	3·5

In general a large flow of water with a small temperate rise is to be preferred. If the source of supply is too cold for satisfactory and economical operation of the engine this can be corrected by returning part of the warm discharge water to the pump suction. Inlet and outlet temperatures will be discussed under the various systems described below.

Direct Cooling by Salt Water.—In the tropics sea temperatures as high as 95° F. are experienced and it is not advisable to allow the outlet water temperature to exceed about 135° F. This gives a temperature rise of 40° F. The above table gives 6·1 galls. or $\frac{1}{37}$ ton per B.H.P. hour. The salt water enters the ship by way of a ship-side valve provided with a strainer, and passes thence through a weed box to the blast air compressors (if fitted) and the lubricating oil cooler. It is usual to pass the whole quantity of water through the blast air compressor and oil cooler in series in order to secure for these organs the lowest possible temperature conditions. Then the water passes to a main along the engine by which it is distributed by branches to the several cylinders. The discharge from each cylinder is controlled by a valve discharging to a common outlet pipe, which leads the water overboard through a ship-side discharge valve. In some instances the pistons are cooled by salt water, in which case appropriate inlet and outlet branches with control valves are arranged for each piston cooling gear. Thermometer pockets are arranged at inlet and outlet, and permanent thermometers are fitted at the water discharge from each cylinder.

The disadvantages of this system are as follows :—

(1) The unavoidable deposit of salts in the hotter parts of the cylinder jackets, particularly cylinder covers.
(2) Accumulation of mud, sand, etc., in the water spaces.
(3) In cool waters the temperature of water entering the cylinder is undesirably cold.

A better but more expensive system is described below.

Indirect Cooling by Salt Water.—In this system the water jackets and sometimes the pistons are circulated with fresh or distilled water. The fresh water and lubricating oil are cooled in heat exchangers of the condenser type, through which salt water is circulated. If blast air compressors are fitted they are circulated with the same salt water which is used in the heat exchangers. An incidental advantage of this system consists in the ease with which the jacket water can be heated by a steam jet during cold weather in port. The chief advantage is the elimination of incrustation and sludge.

In the fresh water cooler, the salt water passes through the tubes and the fresh water outside. The fresh water may pass over the tubes transversely or longitudinally in several passes.

The heat transmission coefficient from water passing through a $\frac{3}{4}''$ bore tube at 4 ft./sec. to the inside wall of the tube is about 1500 B.T.U./ft.^{2} hr. °F. With a similar longitudinal water speed on the other side of the tube the overall coefficient of heat transfer from fresh to salt water works out at about 700 B.T.U./ft.^{2} hr. °F. Fouling of the tubes after a few voyages may reduce this figure to 200 or less. If we adopt the latter figure to avoid frequent cleaning and assume a mean temperature difference of 20° F. between the salt and fresh water, and a heat flow of 1200 B.T.U./B.H.P. hr. (if the fresh water cooler has only to deal with cylinders and covers and not the piston cooling) the required surface works out at,

$$\frac{1200}{200 \times 20} = 0{\cdot}3 \text{ ft.}^2/\text{B.H.P.}$$

Direct Cooling with Unlimited Supply of Fresh Water.—This system arises in connection with stationary plants adjacent to lakes, rivers, or canals. It is generally similar to the first system mentioned above and suffers from like disadvantages. The water outlets from each cylinder are sometimes led to an open funnel or tundish, at which the flow can be observed.

In addition to the possibility of incrustation and the collection of mud in the jackets there are dangers of corrosion due to dissolved oxygen or chemical refuse, etc.

Care must be exercised in arranging the water pump suction in such a way that only reasonably clean water is drawn into the system. It is usual to provide a reserve pump and/or an overhead tank with a capacity equal to about half an hour's flow. For reasons indicated, there is a growing body of opinion in favour of indirect cooling, using heat exchangers as described above for salt water.

Cooling with Restricted Water Supply.--Where water is scarce or expensive it is necessary to circulate the cooling water and transfer the heat to atmosphere at some part of the circuit by means of one or other of the following means :—

(1) Cooling tanks.
(2) Cooling pond.
(3) Cooling tower.
(4) Natural draft radiator.
(5) Forced draft radiator.

Methods (1) and (2) depend partly on radiation and natural connection, aided by wind, and partly on the cooling effect of evaporation. In method (3) the actions are intensified by the breaking up of the flow into drops falling through the air with considerable speed under gravity. In certain types of forced draft radiator (item 5) the water is drawn into thin films over the cooling elements and comes into direct contact with the cooling air. In all these cases the make up feed is considerable and may amount to 3% of the flow. If the make up feed is too hard softening may be desirable.

Natural draft radiators may be successfully used at the top of high buildings or other exposed situations.

Forced draft radiators of the enclosed type provide the simplest solution for road and rail traction purposes.

Capacity of Cooling Water Tanks.—Consider a 16 B.H.P. engine rejecting, say, 2000 B.T.U. per B.H.P. hr. to the cooling water provided with circular cooling water tanks 3′ 0″ dia. × 7′ 0″ high. Let the maximum and minimum water temperatures be 150° F. and 100° F. respectively, and the air temperature 60° F. For simplicity of calculation, suppose the tanks connected in parallel; in practice they are commonly connected in series.

The area of the surface of water is 7 ft.^{2} and that of the sides 66 ft.^{2} for each tank. The temperature difference between water surface and air$=150-60=90°$ F. The mean temperature difference between tank sides and the air$=\frac{1}{2}(150+100)-60=65°$ F. Neglecting heat lost by evaporation the coefficient of heat transmission is about 1·5 B.T.U. per ft.^{2} hr. °F. The total heat dissipated is therefore

$$1{\cdot}5\,(7\times 90+66\times 65)=7300 \text{ B.T.U./hr. per tank.}$$

The required number of tanks is therefore

$$\frac{16\times 2000}{7300}=4{\cdot}3$$

Probably three tanks would be sufficient in most instances. Their total capacity would be about 136 ft.^{3}=8500 lb. of water, and the initial rate of temperature rise would be

$$16\times 2000\div 8500=3{\cdot}8 \text{ °F. per hour.}$$

This rate of temperature rise would diminish progressively as the water becomes heated above the air temperature.

If the engine runs 8 hours a day on full load and shuts down with water temperatures of 150° F. and 100° F. (mean 125° F.) the initial rate of fall of temperature will be

$$3\times 7300\div 8500=2{\cdot}6 \text{ ° F. per hour.}$$

A few trials shew that the mean rate of fall during the next 16 hours will be about 2·0° F. per hour, so that the mean water temperature on starting next day will be $125-2\times 16=93°$ F.

The rate of heat emission at starting will be $3\times 1{\cdot}5\times (7+66)(93-60)=10{,}800$ B.T.U./hr.

At shutting down $3\times 7300=21{,}900$ B.T.U. hr.
Mean $=16{,}350$ „ „

Heat accumulated in water per hour of running time

$$=\frac{8500\,(125-93)}{8}=34{,}000$$

Total heat flow accounted for,

$$34{,}000+16{,}350=50{,}350$$

which is in excess of the required amount, viz., 32,000 B.T.U./hr. This means that the mean water temperature of 125° F. will not be attained.

Head Available for circulation.—If,

a=height of water jacket measured from water inlet to water outlet.

b = height of water tank measured in the same way.
c = height from water outlet from jacket to bottom of tank.
s_1 = specific gravity of water at inlet to engine.
s_2 = ditto at outlet from engine.

Then the head available for circulation is approximately,

$$(s_1 - s_2)\left(c + \frac{a+b}{2}\right)$$

For example,	s_1 for	100° F. =	0·993
	s_2 ,,	150° F. =	0·980
	$s_1 - s_2$	=	0·013

Let a = 1, b = 6·5, c = 2 ft.
Then head available for circulation is,

$$0{\cdot}013\left(2 + \frac{1+6{\cdot}5}{2}\right) = 0{\cdot}075 \text{ ft.}$$

The piping and fitting must be large enough in bore to give the required flow with this head.

The required flow is,

$$\frac{16 \times 2000}{(150-100)} = 640 \text{ lb./hr.} = 64 \text{ galls./hr.}$$
$$= 10{\cdot}2 \text{ ft.}^3\text{/hr.} = 4{\cdot}9 \text{ in.}^3\text{/sec.}$$

If the piping is $1\frac{1}{2}''$ bore (1·77 in.^{2} area) the velocity is 0·23 ft./sec., and the velocity head is 0·00082 ft., so the available head is 92 times the velocity head.

To find Reynolds number for the water flow,

v = about 0·0052, d = 3·8 cms. V = 12·4 cms./sec.

$$\frac{Vd}{v} = \frac{12{\cdot}4 \times 3{\cdot}8}{0{\cdot}0052} = 9{,}000,$$ which is above the critical value, so the resistance must be calculated for turbulent flow. For this value of Reynolds number the velocity head is lost about every 20 diameters (rough pipe). If the total length of pipe is, say, 40 ft. $= 40 \times 12 \div 1\frac{1}{2} = 320$ diameters, the total loss of head will be made up about as follows :—

Item		Times velocity head
Pipe friction $320 \div 20$	=	16
10 sharp bends	=	10
2 valves 2×20	=	40
		66

which is less than the available head of 92 times the velocity head.

This very rough calculation is merely indicative of the method of dealing with such a problem.

Cooling Ponds.—Where space is limited a cooling pond may sometimes be conveniently arranged on the roof of the engine house. If the outlet temperature is 150° F. and the air temperature 60° F., and the coefficient of transmission of heat from water to air 1·5 B.T.U./ft.^{2} hr. °F., the required surface area amounts to about

$$\frac{1660}{1 \cdot 5 \times 90} = 12 \text{ ft.}^2 \text{ per B.H.P.}$$

Such an allowance would probably be excessive unless the engines were required to work at full load night and day. If the engines only work, say, 16 hours a day at an average load factor of 60%, the required surface would be considerably less. In temperate climates the rate of heat transmission will usually be increased by evaporation and wind. Experience of similar conditions in the same kind of climate is the best guide, but account should be taken of the worst condition of air temperature and humidity likely to arise.

Utilisation of Jacket Heat.—In communities where central heating is favoured the jacket heat of Diesel Engines at the central station can be profitably used for this purpose. A plan which has been successfully carried out is as follows:—

The jacket water of Diesel Engines aggregating several thousands of B.H.P. is circulated through the town a distance of a mile or more, and returns to the engine. The water is circulated at such a rate that the temperature rise in the engine is about 30° F., and it is heated a further 20° F., or so, in an exhaust fired boiler. The heat so distributed is sold to consumers and the revenue is an important item in the working account. A cooler is installed at the central station for temperature adjustment, particularly for use in summer.

The same plan can be adopted in hotels, factories, or other buildings in which a use can be found for a supply of low temperature heat. With such systems hot water supplies are obtainable by means of heat exchangers without bleeding the circulating system.

CHAPTER XXVI

NOISE

In the extreme forms of artillery and the like, the internal combustion engine is the noisiest invention of man. The results of shell shock in war have drawn attention to the injurious effects of excessive noise in civil life, and there is an increasing consciousness of the desirability of reducing mechanical noise to some tolerable level. It is remarkable to what an extent the noise consciousness of an individual is heightened by a few observations with a noise measuring instrument. It may be predicted that when the use of such instruments becomes more general and results are expressed in universally accepted numerical form, the general demand for quieter operation of machinery will become more insistent.

Quietness is an obvious desideratum for the passenger accommodation of ships, railway trains, motor vehicles and aircraft, but there are other aspects of the subject of mechanical noise which deserve consideration.

Excessive noise from an engine, even if prevented from reaching the ears of people outside the engine compartment, may have effects on the engine attendants not only injurious to health but prejudicial to effective attention to their duties. Long-continued exposure to excessive noise causes partial deafness and indifference to noise qualities which may prevent the detection of less intense sounds which might otherwise give early warning of the development of an abnormal condition of the machinery. For example: a shortage of oil to the gudgeon pin of an oil engine causes a characteristic top end rattle or knock which may continue undetected if the engine is noisy in other respects, due to valve clatter, for instance.

This suggests a third aspect, viz. noise as a symptom of maladjustment, whether inherent in design, introduced by faulty handling, or due to wear.

Exhaust Noise.—If sufficient space is available for chambers of large volume in relation to the size of the cylinders, there is no difficulty in effectively silencing the exhaust outlet to atmosphere (see p. 424). With restricted weight and space on board ship, the use of lagged steel exhaust pipes and lagged steel silencers of moderate volume, gives a satisfactory result with practically complete inaudibility in the passenger accommodation. Waste heat boilers are commonly used, and these have a good silencing effect, but it is usual to fit silencers in case the boilers have to be bye-passed.

In road and rail vehicles the limitations of weight and space are such that complete silencing with full throttle is seldom achieved. With such vehicles periods of full throttle are usually of short duration, and no great inconvenience is felt if the exhaust noise is barely perceptible against the background of other noises, many of which are extraneous to the engine.

The silencing of aircraft engines is still more difficult on account of the severe limitations of weight and space.

Further developments of exhaust silencing, particularly for high speed engines, may be expected along the lines of :—

(1) The interference principle, whereby definite notes are eliminated by cancellation of waves. In principle the exhaust is divided into two streams along separate passages which unite later. If the lengths of the two paths differ by half the wave length corresponding to the note to be eliminated, every wave crest traversing one path meets a wave trough which has traversed the alternative route, and cancellation results. In this way one or more notes can be eliminated or greatly reduced in intensity. It is not essential for equal quantities of gas to traverse both routes, but pressure waves of equal frontal area × amplitude.

(2) The diffusion principle in which the propagation of waves is hindered by perforating the exhaust pipe and surrounding it with an annular chamber which may be filled (in part or whole) with divided material with a high percentage of voids.

Silencers of the older conventional types appear to dissipate the energy of the sound waves by repeated throttlings and expansions which break up the acoustic pattern. They are liable to cause excessive back pressure unless the pressure drop at each restriction is calculated in advance of manufacture.

Air Suction Noise.—The air suction noises of a four stroke engine appear to originate at the head of the valve. The old slotted strainers were just tolerable with slow speed engines, but are unsuitable for most present requirements. The noise can be reduced to a negligible amount by fitting felt lined manifolds with trumpet entrances, also felt lined. Similar arrangements can be used with reciprocating and other types of blowers.

Positive rotary and centrifugal blowers tend to emit a definite note depending on the revolutions and the number of impeller blades. Interference silencers of the type referred to above can be used with good effect.

Valve Clatter.—This source of noise increases with weight of the moving parts concerned, which should therefore be as light as other considerations will allow, but so long as definite clearances are required between cams and followers, tappets and push rods, levers and valve stems, considerable noise, producing shock, is inevitable at high speeds. Cam noise can be moderated by flooding with oil and largely suppressed by total enclosure of the cam-shaft. It is important for cam-shafts to be of sufficient diameter, otherwise noise may be accentuated by torsional recoil from one cam to another. This point deserves special notice in connection with long engines (for example, 10 or 12 cylinders in line).

Tappet noise between push rods and levers, levers and valves, may likewise be muffled by air tight covers enclosing the whole gear. Such covers are more effective if felt lined. It appears that fuel pumps and fuel valves of airless injection engines contribute an appreciable amount of noise which may be mitigated on the above lines.

For special purposes a sheet steel felt lined "overcoat" applied to the noise-emitting surfaces of an enclosed engine has been found to make a marked reduction in the emission of noise.

Piston Slap.—This has been referred to in Chapter XVI, p. 311. Compared with other sources of noise piston slap (unless acute) is a relatively dull deep noise, providing an inconspicuous background for the livelier noise of valve clatter. For this reason it easily escapes notice and in its milder degrees, it is probably a good deal more prevalent than is commonly supposed. When other sources of noise have been reduced to a low level, piston slap may become conspicuous among the sources of noise requiring treatment.

Bearing Knocks.—A stoppage of oil to the top end of a four stroke engine is followed by a characteristic knock, giving what is often quite inadequate warning of piston or gudgeon pin seizure. A general shortage of oil, short of complete starvation, can produce noisy running without definite failure. In such instances there appears to be insufficient oil in the bearing clearances to cushion the reversals of pressure.

The general noise level of an engine with bearings set or worn to clearances which are too coarse, can be materially reduced by adjusting the bearings to finer clearances. This source of noise may escape early notice if the engine is noisy in other respects.

Two stroke single acting engines running at moderate speeds are exceptional in this respect; bearings may be run out without knock being noticed; this is due to the absence of pressure reversal. With higher speeds the pressure at the big ends may be reversed on the upper half of the compression stroke by the inertia of the reciprocating parts and the centrifugal force of the revolving part of the rod.

Roller and Ball Bearings give out a characteristic noise, and engines fitted with these in positions of main duty may be noticeably noiser than similar engines fitted with plain bearings.

Automatic Valves are used for compressors, reciprocating blowers, reciprocating water, oil and bilge pumps, and in some systems of two stroke engine scavenging. As used for air such valves are usually of a light plate type, with carefully restricted lift to prevent shock, avoid cracking, and minimise noise. As suction valves of compressors and blowers they contribute a characteristic deep throb which may dominate the general noise background unless the suction inlet is carefully silenced.

The valves of reciprocating water pumps appear to work best at high speed when the valves are of large diameter and low lift. An air snifting valve is usually provided. In cases where heavy valves of this kind work quite quietly at high speed it seems probable that the valve oscillates between limits without actually seating metal to metal.

Gearing.—Spiral gears and high efficiency worm drives are fairly quiet; well designed roller-chain drives with wide centres between wheels run sweetly, but if the centres are short in comparison with the chain pitch, running may be distinctly harsh.

Trains of spur or bevel wheels may be decidedly noisy if the loading is alternating or the peripheral speed high, particularly if the tooth form, finish, centering or meshing are imperfect. The use of helical teeth and accurate finish by grinding contribute towards quiet running.

Literature.—Hart, M. D., "The Aeroplane as a Source of Sound."—Aero. Res. Comm. R. & M. 1310 N. 26, H.M. Stationery Office.

Davis, A. H., "Modern Acoustics."—Bell.

Davis, A. H., "Noise."—*Engineer*, 17.8.34.

CHAPTER XXVII

GENERAL SURVEY

Heat Engine Cycles.—Recent developments of internal combustion turbines and closed cycle turbine systems, as well as the compounding of reciprocating internal combustion engines with gas turbines and the use of regenerative heat exchangers to improve the efficiency, have focussed attention on heat engine cycles well known in thermodynamic theory, but hardly practicable with the reciprocating engine alone. Of these theoretical cycles a unique position is occupied by the constant temperature cycle, the theoretical efficiency of which was shown by Carnot to be unsurpassed by any other cycle having the same upper and lower limits of temperature. Amongst Dr. Diesel's original proposals for a " Rational Heat Motor " was included a scheme for operating a reciprocating internal combustion engine on a near approach to Carnot's cycle in accordance with the following operations.

(1) Approximately isothermal compression during the first part of the compression stroke, facilitated by water injection.
(2) Adiabatic compression for the remainder of the compression stroke in an uncooled top part of the cylinder.
(3) Expansion at a temperature maintained constant by a controlled admission of fuel, followed by
(4) Adiabatic expansion.

Experimental work soon showed the necessity for departing from this programme by dropping all attempts at isothermal compression and arranging for a period of combustion at nearly constant pressure; but even after the engine had reached commercial production, Dr. Diesel was able to exhibit remarkable indicator diagrams showing a period of combustion at nearly constant temperature sandwiched between a period of combustion at constant pressure and a near approach to adiabatic expansion. The curtailing of the expansion stroke with release of exhaust gas at a high temperature and a pressure of 3 or 4 atmospheres was obviously a cause of discontent to Diesel, who constructed a compound engine with a low pressure cylinder to complete the toe of the indicator diagram. This, however, failed for reasons which appear more obvious now than they did then, viz., difficulties with the transfer valves and excessive friction of the L.P. cylinder in comparison with the additional work obtainable. The development of efficient rotary blowers of axial and centrifugal type and the availability of heat resisting steels suitable for gas turbine blades working at temperatures of 1000° F. and over, have brought the compounding of internal combustion engines within the practical sphere.

The later developments of steam turbine practice with interstage reheating and regenerative feed heating may be interpreted as efforts to modify the Rankine cycle into closer conformity to the Carnot cycle. Recent developments of the gas turbine with regenerative heat exchangers and interstage reheaters may be interpreted in much the same way.

Next in historical importance comes the constant volume cycle, on which most petrol and gas engines operate ; with the airless injection oil engine there has been a tendency to admit higher and higher maximum pressures, particularly in high speed engines so that they now operate on a cycle which approximates more to the constant volume cycle than the "Diesel cycle." Both cycles suffer from the premature release of exhaust to which allusion has already been made. Attempts to overcome this defect by making the expansion stroke longer than the compression stroke (Atkinson cycle) have been technically successful in at least one instance but have not achieved commercial production.

The constant pressure cycle has never achieved much success with the internal combustion reciprocating engine, but its use in conjunction with a high speed rotary blower of axial or other type and a gas turbine with a combustion chamber between the outlet from the blower and the inlet to the turbine, leads to a very simple and compact prime mover. Since the nett power is only a fraction of the power absorbed by the blower, the efficiency of the combined arrangement depends very much on the efficiency of blower and turbine separately. The efficiency may be much improved by the introduction of a regenerative heat exchanger whereby the heat rejected at the exhaust end of the turbine is made available to the air before entering the combustion chamber. Further improvements are possible by approximating to isothermal compression by interstage cooling between separate compressor units and by exhaust reheating between successive turbine stages. The cycle then begins to approximate to the Carnot cycle as already indicated.

A great impetus has been given to these developments by the application of jet propulsion to aircraft which has involved much research on the conditions for greatest efficiency of centrifugal and axial compressors and axial flow gas turbines.

Compounding.—Dr. Diesel's abortive experiments on compounding, carried out half a century ago, have been mentioned. The difficulties he experienced have been removed by the development of successful gas turbines. The turbo-charging units of Rateau and Büchi may be regarded as the first step. The next step was to gear the exhaust turbine to the engine and arrange for a fairly early release of exhaust so that the turbine output was more than sufficient to drive the supercharger and the excess was available for the final output drive. As a further step, the supercharger was increased to give a large excess of air and to absorb the whole output of the reciprocating engine, the useful work being derived from the exhaust turbine only. The reciprocating engine and blower then took the place of the boiler or steam generator in a steam turbine plant. These developments have been associated with the names of Gotaverken and Sulzer. A scheme of this sort has been proposed for an aircraft engine. It has been suggested that a comparatively simple type of loop scavenge radial engine with gear driven supercharger would be suitable for the gas generating unit since a very high combustion efficiency would not be necessary. These ideas have opened up new possibilities for the use of reciprocating oil engines.

It is instructive to compare these ideas with the historical development of the steam engine through successive stages from the low pressure single stage non-condensing engine to the multistage reciprocating engine with condenser, the combination arrangement of reciprocating engine and exhaust turbine, to the multistage turbine installation with heat exchangers for air heating, feed heating and interstage reheating.

Reciprocating Internal Combustion Engines.—The compression ignition oil engine is one of many forms of the internal combustion engine, which may be classified in various ways, e.g. :—

Fuel . .	*Gaseous.*—Town gas, producer gas, natural gas, sewage gas, etc. *Vapour.*—Petrol, paraffin. *Liquid.*—Petroleum distillate, tar oil, etc. *Solid.*—Coal dust.
Compression	*Low Compression.*—Below ignition point of fuel. *High Compression.*—Above ignition point of fuel.
Ignition .	*Spark Ignition.* *Surface Ignition* (Hot bulb, etc.). *Compression Ignition.*

The possible combinations are numerous and many have been used successfully.

For example, many engines have been built with compression ratios of about 11 to 16, running on town gas, natural gas, sewage gas, or producer gas and ignited with a small pilot injection of liquid fuel. Such engines run very smoothly at mean pressures little below those appropriate to oil engines of the same size and general type. The pilot injection of liquid fuel accounts for about 3 to 15% of the total heat supplied. Such engines are readily arranged to run alternatively as gas engines or oil engines and the changeover from one fuel to the other can be carried out practically instantaneously on load without any misfiring or observable fluctuation of speed.

The pilot injection of liquid fuel gives rise to a small but noticeable hump on the combustion line of a draw card (indicator card taken by pulling the drum cord by hand during the rise of the pencil), and appears to have a very useful effect in preventing excessive rate of pressure rise (detonation). High compression gas engines can also work successfully with spark ignition in small cylinders. With large cylinders, say 20″ diameter, bumping is liable to occur if the mean pressure exceeds a rather moderate value. Whether this can be avoided by more intensive cooling or other precautions does not yet appear.

A quite different combination of qualities is found in the Hesselmann low compression spark ignition engine which with little modification can run on petrol or gas oil. With gas oil the customary injection equipment is used to introduce the fuel in a fine spray which is ignited by spark.

Much work has been put into the compression ignition engine burning coal dust, but it appears still to be in the experimental stage.

The remaining sections of this chapter will deal with the reciprocating compression ignition heavy oil engine.

Range of Size, Speed and Output.—The compression ignition oil engine now covers a range of sizes, speeds, duties and outputs not paralleled by any other type of prime mover.

The highest outputs are associated with marine propulsion and low revolutions. The highest revolution speeds have been reached in the endeavour to meet the requirement of aircraft. The adjoining table gives an approximate survey of the present position (1947).

Service.	B.H.P. per Cyl.	Cyl. dia. ins.	R.P.M.	Piston Speed Ft./min.
Marine Propulsion (Merchant)				
Ocean-going vessels	200/3000	16/33	150/85	1000 to 1300
Channel vessels	100/400	14/25	250/150	1000 to 1300
Coasters, tugs, etc.	50/200	10/20	300/200	1000 to 1300
Smaller craft	5/30	3/8	1200/400	1000 to 1300
Marine Propulsion (Naval)				
Submarines	100/500	10/20	500/400	1500/1000
High-speed craft	30/150	6/8	2500/1000	2600/1600
Ships' Auxiliaries				
Merchant	40/200	8/16	600/300	1300/1000
Naval	50/150	6/10	1200/700	1600/1200
Industrial Engines				
Large	150/3000	14/33	300/120	1000 to 1500
Medium	50/150	8/14	600/300	1000 to 1500
Small	5/50	3/8	2000/600	1000 to 1500
Land Traction				
Locomotives	50/150	7/14	1500/700	1600/1200
Army tanks	20/50	4/6	2000/1500	2500/2000
Road vehicles and small auxiliaries	5/35	3/6	4000/1500	2000/1500

Breathing Capacity.—Of the various factors which limit the output of an internal combustion engine the most important is breathing capacity. In discussing questions of output it is convenient to refer to the B.S.I. Standard of rating, which may be interpreted as follows :—

Twelve Hour Rating.—Power developed by the engine for the purpose of an acceptance test not exceeding twelve hours in duration.

One Hour Rating.—Power developed during a one hour overload test; it is 10% greater than the twelve hour rating.

Twenty-four Hour Rating.—Power developed over a test period exceeding twenty-four hours in duration; it is 90% of the twelve hour rating, (1935 specification).

The twenty-four hour rating may also be interpreted as the service power for night and day running, but in view of the variations of service loads and the wisdom of over-estimating rather than underestimating service requirements, the *average* load in service is usually between 60% and 80% of the twelve hour rating in the case of industrial engines and about 70% to 85% for marine engines. In either case the 10% overload rating is seldom used in service.

Locomotive engines may average as much as 70% of the twelve hour rating on main line service or as little as 10% in shunting duty. The load factors of road vehicle engines probably lie somewhere between these extreme figures, and the outputs of these engines are commonly expressed by performance graphs on a basis of revolutions and without reference to the B.S.I. basis of rating for stationary engines. The outputs at the various speeds may be close to the smoke limit without mention of duration. This should be borne in mind when comparing the high brake mean pressures developed by such engines with the more conservative figures in use for industrial engines.

An average airless injection industrial engine of the normally aspirated four stroke type develops on its twelve hour rating a brake mean pressure of about 75 lb./in.^{2} and the volumetric efficiency based on the ambient air conditions ("air supply ratio") is about 85%. It is an interesting fact that there is little difference in this ratio as between an engine with one exhaust valve per cylinder, running at a piston speed of 1000 ft./min. and an engine with two exhaust valves per cylinder, running at a piston speed of 2000 ft./min., the respective nominal air speeds being about 150 and 230 ft./sec. respectively. The higher speed engine would usually be given a slightly later closing of the air inlet valve, say 40° late as against 30° for the slower.

With a brake mean pressure of 75 lb./in.^{2} and an indicated mean pressure of about 93 lb./in.^{2}, about 50% of the induced air has its content of oxygen converted to CO_2 and H_2O vapour and the exhaust contains about 7½ of CO_2.

With a normally aspirated engine it is reasonable to assume that the whole of the induced air is retained in the cylinder on the compression stroke and that a volume of exhaust gas equal to the clearance volume is retained also. With special valve timings and tuned inlet and exhaust systems it may be possible to scavenge the exhaust gas out of the clearance space and thereby increase the air supply ratio about 8% to 10% with a corresponding increase of potential output. Furthermore, by special "ramming" inlet pipes it has been found possible to obtain a substantial supercharging effect at one particular speed and thereby to increase the output still further, but this arrangement has not been widely used on account of the unwieldly pipes required.

When supercharging by means of an engine driven, or separately driven blower, the first advantage obtained is the scavenging of the clearance space. With a suitable overlap of air and exhaust valve openings at the idle dead centre this advantage can be obtained with a low air pressure of about 1 lb./in.^{2} With higher pressure air the advantage is further increased by virtue of the greater density of the air trapped in the cylinder. It must be borne in mind, however, that the density of the induced air depends not only on the pressure but also on its temperature and this depends on the adiabatic efficiency of the blower. With a blower having an adiabatic efficiency of 75% the resulting density of the air as compared with air at atmospheric pressure and 60° F. is approximately as follows :—

Pressures lb./in.^{2}	2	4	6	8	10	12	16
Pressure ratio .	1·13	1·27	1·41	1·54	1·68	1·82	1·96
Density ratio .	1·08	1·16	1·24	1·31	1·39	1·46	1·53

These figures show the important gain in potential output which is obtainable, with higher pressure, by cooling the air after compression. Apart from the direct gain by scavenging the clearance volume there is an additional gain (difficult to assess accurately) by reason of the cooling effect of passing an excess of air during the overlap period. This cooling effect mitigates to some extent the reduction of density caused by the convection of heat from the walls of the combustion chamber on the suction stroke.

With a turbo-charged engine the average exhaust pressure at full load is less than the air pressure so that there is a small additional gain on this account. With a direct driven blower the work absorbed in driving the blower is not fully compensated by the excess of suction pressure over exhaust pressure, so that a certain loss is sustained in this respect. With an electrically driven blower there is the additional loss of two transformations, viz., from mechanical work to electrical energy and vice versa.

An important point is the extent to which power may be augmented by supercharging without increasing the intensity of heat flux to the cylinder cover, cylinder liner and piston as compared with a normally aspirated engine, bearing in mind that the cooling effect during valve overlap benefits just those parts which are most in need of it. It seems to be established that the power of a normal engine can be increased 30% by supercharging without increase of heat flux at the critical spots, but any increase much beyond this figure may necessitate modifications, depending on whether the original design has or has not a margin of security against heat stressing. Engines of small size running at moderate piston speed frequently have a wide margin in this respect. In the larger sizes of engine, oil-cooled pistons frequently have a large reserve capacity for dealing with heat flux. It seems reasonable to suppose that cylinder covers fitted with two smaller exhaust valves have a wider margin than cylinder covers with a single exhaust valve of larger diameter.

With two cycle engines it is necessary to discriminate between the air supply ratio as defined above and the volumetric efficiency defined as the volume of air (referred to ambient conditions of pressure and temperature) retained in the cylinder during compression, divided by the swept volume of the cylinder.

Most forms of uniflow two cycle engines must be regarded as doing well if they achieve a volumetric efficiency of 0·85 with an air supply ratio of about 1·20. Such engines can develop about the same mean pressure as a normally aspirated four stroke engine but with a somewhat lower exhaust temperature. To obtain an equal volumetric efficiency with loop scavenge usually requires a higher air supply ratio of 1·4 to 1·5 and valve control of the air ports so that the latter may be closed by the piston after or simultaneously with the exhaust ports. The loop scavenge engine without valve control of the air ports has been improved, as compared with earlier models, by the direction of the entering air streams against the cylinder wall opposite the exhaust ports and giving the streams an upward inclination at about 45°. This is known as "back flow" scavenge and helps to prevent short circuiting of air from the air ports to the exhaust ports.

There is no obvious limit to the extent to which a two stroke engine may be supercharged by increasing the output of the blower and throttling the exhaust. A very early Junker engine had its indicated output doubled by this simple expedient; but a large blower absorbs power which is increased by throttling the exhaust so the two stroke designer aims at narrowing the margin between air supply ratio and volumetric efficiency without throttling the exhaust unless the exhaust gases are to be used to drive a turbine.

A certain amount of deliberate throttling of the exhaust has from time to time been found to improve the performance of two cycle engines by diminishing the loss of air through the exhaust ports before they are covered by the piston. A similar effect is obtainable to greater advantage if pressure pulses in the exhaust manifold can be arranged to synchronise with the closing of the exhaust ports whilst pulses of negative pressure prevail during the middle part of the scavenge period, thereby reducing scavenge resistance.

Remarkable results have been obtained, particularly with single cylinder engines, by utilising the ejector effect of the exhaust ports and pipes to induce the air charge without using a blower. Volumetric efficiencies of about 1·4 have been obtained in this way. In such cases it appears that the cylinder dimensions, porting and revolutions are such that the scavenge resistance is small, of the order 1½ to 2 lb./in.^{2}. By scavenge resistance is meant here the static pressure required to pass air through the inlet ports and exhaust through the exhaust ports against atmospheric pressure at the required rate. Nevertheless the fact that an effective urge of 1½ to 2 lb./in.^{2} is obtainable in this way is significant and shows the importance of correct timing, porting and streamlining of the whole air induction and exhaust system of a two cycle engine.

When air enters the cylinder by way of a complete ring of tangential ports it appears that a vortex is set up with a tendency for a core of exhaust to be left at the centre. With a view to removing this core, the last part of the ports to be uncovered on the outstroke and first to be covered on the instroke are sometimes made radial.

The whole sequence of exhaust, scavenge and charging is a complicated aerodynamic process calling for much experimentation and theoretical analysis. The simplest view which can be taken of it in the first instance is to regard it as a straightforward chain of variable orifice discharges, and then to consider to what extent effective use can be made of the inertia of the gas streams to facilitate entry of air through the ports, the creation of a partial vacuum in the cylinder during the middle of the scavenge period, and the final charging of the cylinder prior to the closing of the exhaust ports. The problem is naturally complicated by the angular settings of the various cylinders exhausting into a common manifold, and the acoustic properties of the exhaust piping system.

The Compression Space.—A compression ratio of 11 is about the lowest to give satisfactory starting with an air temperature of 50° F. without heating the jackets or the incoming air or using glow plugs or squibs, etc. In practice the smaller sizes of cylinders are given higher compression ratios approximately as under.

Cylinder dia., in.	3	6	9	12	15	18	21
Compression ratio	18	16	14½	13½	13	12½	12

Increasing compression ratio increases the theoretical upper limit of attainable efficiency and potential output. It also increases the temperature and density of the charge before fuel injection and thereby reduces delay of combustion and leads to a smoother and more controllable combustion process. There is, however, a serious limitation, viz., the effect of "dead air" sometimes also referred to as "parasitic volume." A dead air space is a part of the clearance volume, e.g. a valve pocket or indicator hole or bumping clearance above the piston rim, which is inaccessible to the injected fuel. In general the effect of turbulence in distributing fuel through the air charge, although important, is very limited in action. With rotary turbulence the process is much more like spreading butter on a slice of bread than vigorous stirring of liquid in a cup with a spoon. The available time is too short for turbulence to carry fuel into recesses at the boundary, and the pattern developed by the relative motions of air and fuel seem to persist not only throughout the expansion stroke but the exhaust stroke also.

Perhaps an exception may be made in favour of engines fitted with precombustion chambers, giving rise to very intense turbulence. It is noteworthy that such engines develop high mean pressures and succeed in consuming a high proportion of the induced charge of oxygen.

With direct injection of fuel into a single chamber the air contained in the "dead air" spaces is practically inert; it follows therefore that the potential output of a cylinder is limited by the "live air" ratio, viz.:—

$$\frac{\text{Clearance Volume} - \text{Parasitic Volume}}{\text{Clearance Volume.}}$$

It appears to be near the truth to say that potential output is proportional to the product of volumetric efficiency and live air ratio. The high performance given by small four stroke direct injection engines for road traction is partly due to the high live air ratio obtained by cutting down parasitic volume to a minimum. With loop scavenge two cycle engines a comparatively low volumetric efficiency may be to some extent compensated by a high live air ratio made possible by the absence of valves (other than fuel, starting and relief valves) in the head.

Unavoidable bumping clearance between the cylinder cover and the piston top, side clearance around the top land of the piston, valve pockets, etc., together make up a certain parasitic volume. If this volume is fixed, an increase of compression ratio results in a reduction of live air ratio, and so there is an upper limit of compression ratio beyond which potential output tends to decrease. The following table gives calculated figures for the optimum compression ratio and relative potential outputs corresponding to various ratios of parasitic volume to stroke volume, on the assumption that the potential output is proportional to live air ratio multiplied by the theoretical efficiency (constant volume cycle) corresponding to the compression ratio.

Parasitic volume % of stroke volume	1	1½	2	2½
Optimum compression ratio	13	11	9	7
Relative output	1	·94	·90	·85

In view of the reasons already indicated for preferring high compression ratios to low, the above figures show the importance of keeping the parasitic volume down to a very small value.

The shape of the combustion space with piston on top dead centre, and ignoring dead air spaces, is determined by the top surface of the piston and the under surface of the cover. With a flat cover and central injector good results have been obtained with a piston cavity of hemispheroidal shape or ellipsoidal with depth equal to about 0·7 times the radius. With such a space five equally spaced injector holes with an included angle of about 110° to 130° are often successful. With engines of small stroke bore ratio a relatively shallow pan shape of nearly constant depth can be used in conjunction with a larger number of fuel nozzle holes with an included angle of 140° to 160°. A compromise between the two above shapes may require a central injector hole of short range, in addition to the six or more inclined holes. The short range of the centre hole is obtained by giving the hole a length/dia. ratio of 1·5 or less. Alternatively the central hole may be dispensed with if a central cone or hump is raised in the centre of the piston to fill up the space which the inclined fuel valve holes cannot provide with fuel.

With a two stroke engine * there is more freedom than with a four stroke engine in shaping the underside of the cylinder cover. Accordingly with such engines successful combustion chambers may be made of conical shape with the tip of the fuel injector near the apex and with a flat or nearly flat piston top, or with a piston top shaped at an angle of about 45° to guide the inlet air and retreating exhaust.

The so-called "squish" effect, i.e. the promotion of turbulence by the near approach of relatively large opposed surfaces of piston and cylinder cover at the top dead centre has been much used in small engines, and there is a tendency to use this effect also in larger engines. With supercharged four stroke engines this effect is to some extent limited by the necessary clearance spaces for the air and exhaust valves during the overlap period.

Besides the above mentioned many other shapes of combustion chamber have been used successfully, but apart from pre-combustion chamber designs the same principles apply, viz., high live air ratio and high ratio of,

$$\frac{\text{Live air volume}}{\text{Live air surface}} \div \text{cylinder diameter}$$

It appears that in general, cylinders of large diameter are more

* Of the loop scavenge type.

tolerant in respect of shape of combustion chamber than small ones and this may be attributable to the longer period of time available for diffusion and combustion.

Mean Pressure and Fuel Consumption.—One pound of diesel fuel having a lower calorific value of 18,000 B.T.U. per lb. requires for complete combustion about 14·2 lb. of air, each lb. occupying 13·1 ft.^{3} at 60° F. and atmospheric pressure (14·7 lb./in.^{2} abs.). If the indicated thermal efficiency relative to the lower calorific value were 100%, the maximum M.I.P. corresponding to 100% utilisation of the air would be,

$$\frac{18{,}000}{14{\cdot}2 \times 13{\cdot}1} \div \frac{778}{144} = 520 \text{ lb./in.}^2$$

and the fuel consumption $\dfrac{33{,}000 \times 60}{18{,}000 \times 778} = 0{\cdot}142$ lb./I.H.P. hr.

The indicated thermal efficiency is limited on theoretical grounds by the compression ratio, the maximum pressure and the % of air utilisation. With a compression ratio of 13, a maximum pressure limited to 700 lb./in.^{2}, the ideal indicated thermal efficiencies taking into account the variation of specific heats are approximately as tabulated below for various percentages of air utilisation. The same table gives the corresponding mean pressures and fuel consumptions per I.H.P./hour.

% Air Utilisation	0	20	40	60	80	100
Efficiency thermal %	63	61	57½	55	52	50
Mean pressure, lb./in.^{2}	0	63	109	168	210	249
Fuel consumption lb./I.H.P. hour .	·221	·231	·243	·261	·280	·295

To allow for incomplete combustion, radiation and conduction loss and late burning it is reasonable to increase the consumption by about 14% and reduce the mean pressures inversely *pro rata* obtaining the following :—

% Air Utilisation	0	20	40	60	80	100
Mean pressure lb./in.^{2}	0	55	96	147	184	217
Fuel consumption lb./I.H.P. hour .	·251	·263	·276	·298	·320	·335

These fuel consumptions are in fair agreement with values obtained at various fractional loads in practice but the mean pressures would require for their attainment a fairly high degree of supercharge.

Bearing in mind that dead air is practically ineffective for combustion and that the temperature is far from uniform over the volume of the air charge during combustion, it is not unreasonable to regard the dead air as a heat insulator. On this basis the output of a naturally aspirated four stroke engine will be obtained by multiplying the above mean pressures by the product of volumetric efficiency and live air ratio. If these are both 0·85, their product = 0·72, and the following figures result :—

Effective air utilisation %	0	20	40	60	80	100
Mean pressure, lb./in.^{2}	0	39·5	68·5	105	132	156
Consumption, lb./I.H.P. hour	·251	·263	·276	·298	·320	·335
Lost mean pressure	15	15·5	16	16·5	17	17·5
Brake mean pressure, lb./in.^{2}	−15	24	52·5	88·5	115	138·5
Mech. effy.	—	·61	·77	·845	·870	·890
Fuel consumption lb./B.H.P. hour	—	·433	·360	·353	·367	·378

The last four rows show the resulting figures for brake mean pressure, mechanical efficiency and fuel per B.H.P. hour on the assumption that the lost mean pressure varies from 15 lb./in.^{2} at no mean pressure to 17·5 lb./in.^{2} at the 100% live air utilisation.

Up to a brake mean pressure of 80 to 90 lb./in.^{2} the results are in good agreement with test results from medium speed industrial engines, running at, say, 400 R.P.M. Such engines have as a rule an overload capacity which is limited by considerations of thermal effects and durability generally. Smaller engines of generally similar type may give higher mean pressures of 120 lb./in.^{2} or over for short periods but usually with higher fuel consumptions than those tabulated and maximum pressures in excess of 700 lb./in.^{2}.

Other types of engine may be examined in a similar manner. Small cylinders from, say, 4″ to 7″ diameter usually benefit by a compression ratio of 14 to 16 and care must be exercised in design to see that the live air ratio is high.

With fixed compression ratio the efficiency increases with increasing maximum pressure. With fixed maximum pressure the efficiency increases with increasing compression ratio, apart from the dead air effect.

In practice the best compression ratio seem to be somewhat higher than the optimum indicated from dead air considerations and it is scarcely worth while exceeding a maximum pressure 1·6 times the compression pressure.

For example, one of the Junker series of opposed piston two cycle aircraft engines had a compression of 850 lb./in. and a maximum pressure of 1400 lb./in. These figures correspond to a fairly high theoretical efficiency. The output of this engine is further assisted by a very large live air ratio due to the favourable shape of the combustion chamber and the use of four special injectors. Accordingly the engine was able to develop a high brake mean pressure with a fairly good fuel consumption per B.H.P. in spite of a high lost mean pressure of perhaps 35 lb./in.^{2}.

The highest total air utilisation so far claimed for a four stroke engine is about 85%. In a B.S. twelve hour rating 50% is nearer the mark for an average industrial engine, and less for two cycle engines.

The figures given above show that the performances of actual engines can be fairly well accounted for on known physical principles. Complete utilisation of available air is hardly to be expected in a heavy oil engine, although actually attained in the petrol engine. If too high a percentage of the available oxygen is consumed the combustion of the later portions of fuel is liable to be incomplete, resulting in smoke, deposits and excessive heat flow. For engines which are required to work for long periods at sustained heavy loads a good margin is necessary in the interest of reliability and moderate maintenance.

The necessary conditions for high output and low fuel consumption are identical, viz. :—

(1) High volumetric efficiency.
(2) High live air ratio.
(3) Good distribution of fuel in the live air space.
(4) Compression as high as the dead air considerations permit.

(5) Maximum pressure about 1·6 times the compression pressure.

(6) Low value of lost mean pressure.

A normally aspirated four stroke engine running at 300 to 400 R.P.M. with a piston speed of 1250 ft./min. has a lost mean pressure of about 15 or 16 lb./in.^{2}; for a two stroke engine in the same range 18 lb./in.^{2} is about the minimum. Increase of revolution and/or piston speed tends to increase the lost mean pressure. This increase is kept within bounds by curtailing the surface of the pistons and minimising pumping losses by large valve or port areas. With supercharged engines the efficiency of the blower may have an important influence in minimising the lost mean pressure. In all classes of high speed engines the use of light alloy pistons and light connecting rods of H section is helpful.

Exhaust Temperature.—A thermometer or pyrometer in each exhaust branch is a convenient means of checking the balance of power between the cylinders, and average exhaust temperature is often used as a rough criterion of rating. A popular figure for the twelve hour rating of four stroke engines both normally aspirated and supercharged is about 750° to 800° F., as recording by a pyrometer in the exhaust branch a few inches away from the outlet port in the cylinder head and with an uncooled exhaust manifold. The reading of such an instrument is liable to be influenced by the mere proximity of water-cooled metal and if the pyrometer is inserted through a water jacket the reading may be as much as 100° F. too low, so caution is necessary in making comparisons. Exhaust temperatures of two stroke engines seldom exceed 700° F. and may be much less, depending on the air supply ratio.

Lubricating Oil Consumption.—With cross-head engines in which the cylinders are isolated from the crankcase by a diaphragm, the overall consumption of lubricating oil, including cylinder lubrication, amounts in general to a fraction of 1% of the fuel used. With trunk engines much remains to be learned regarding the exact mechanism whereby lubricating oil passes up the cylinder.

A four stroke engine of 24″ stroke running at 300 R.P.M. in good order may use as little lubricating oil as ¼ of 1% of full load fuel, the only special precautions being splash guards over the cranks and a single vented scraper ring below the gudgeon pin. Higher revolutions and piston speeds make the problem more difficult; and slack bearings, leaks past the gudgeon pin, choked vent holes behind the scraper, worn piston ring grooves, may put up the consumption of lubricating oil to a formidable figure, say 3 or 4% of the fuel in extreme cases.

Starting with the total oil flow delivered by the lubricating oil pump, a certain fraction of this, say 10%, is thrown in the direction of the cylinder mouth. This fraction meets several lines of defence, viz. :—

(1) Splash guards (if fitted).
(2) The scraping action of the bottom of the piston skirt.
(3) Scraper below the gudgeon pin.
(4) Scraper (if fitted) above the gudgeon pin.
(5) The pressure piston rings.

It appears that the failure of any one line of defence may greatly augment the consumption of lubricating oil. In four stroke engines the pressure rings are pressed on the upper and lower faces of their grooves alternately and a pumping action is thereby set up if the clearance is excessive. Accordingly a rising consumption of lubricating oil can sometimes be checked by changing the top two rings. Wear of cylinder liners seems to be less effective in promoting excessive lubricating oil consumption than wear of rings and grooves Scraper rings of the slotted or other type may have their effectiveness greatly reduced if the vent holes behind the grooves are choked with oxidised oil products, so the holes should be ample in size and number.

In two cycle engines the piston rings seem to remain permanently pressed on the lower faces of the grooves and pumping action is less than with four stroke engines. On the other hand, the presence of air ports in the liner provides an alternative means of losing oil up the cylinder. This source of loss is controlled by well-vented scraper rings at the bottom of the piston skirt. The venting of the scrapers is no doubt assisted by the scavenge air pressure.

On the whole there is little difference in the lubricating oil consumptions of trunk engines as between four stroke and two stroke. Each is considered satisfactory in this respect if the consumption of lubricating oil in service does not exceed about 1% of the fuel, on the average.

Thermal Rating.—An engine which has an ample reserve of mechanical strength to resist pressure and inertia stresses when cold may lack durability under full load conditions, when temperature conditions have brought about a readjustment of stresses and some of the material may have suffered a reduction of strength due to high temperature. Furthermore, alternate heatings and coolings may bring about non-reversible changes resulting in ultimate failure by cracking. It is very desirable therefore to have some criterion of thermal rating by which the safety or otherwise of a proposed engine may be gauged in this respect. The general problem of heat stressing is so complicated that direct experience is the only safe guide ; in order to extract as much benefit as possible from all available experience and to enable such experience of any one engine to be applied to another engine of different size and revolutions, it is useful to consider a scale of values of a criterion C say, such that geometrically similar engines of different sizes and speeds having the same C value will have the same temperature stresses. It is convenient also to adapt the formula for C to include departures from strict geometric similarity by way of variation of stroke/bore ratio.

The following is suggested :—

$$C = \eta pi\,(C_e - 0{\cdot}14)\,(B^2n)^{\frac{3}{4}}\left(\frac{S}{B}\right)^{\frac{1}{4}}$$

where ηpi = brake mean pressure, lb./in.^{2}
C_e = fuel consumption — lb./B.H.P. hr.
B = bore in inches, S = stroke in inches.
n = revolution per minute.

For two stroke engines multiply by factor 2. The figure 0·14 represents the fraction of a lb. of fuel (lower cal. value 18,000 B.T.U./lb.) giving Joules equivalent of 1 B.H.P. hour.

For example a four stroke engine 12″ × 15″ running at 500 R.P.M. with a fuel consumption of 0·38 lb./B.H.P. hr. and a brake mean pressure of 75 lb./in.^{2} would have a C value of $75 \times 0{\cdot}24\,(12^2 \times 500)^{\frac{3}{4}}\,(1{\cdot}25)^{\frac{1}{4}} = 94{,}000$.

Naturally the admissible value of C will depend on materials used and details of design. If the piston is the limiting part, the admissible value of C will in general be higher if the piston is cooled than if it is not cooled and so on for other parts.

The engine referred to above would probably have uncooled pistons whereas a similar engine, double the size, i.e. 24″ × 30″, running at half the revolution, with the same mean pressure and fuel consumption, would have a C value of 264,000 and piston cooling would be required. The cooling and thermal stressing of cylinder [illegible] perhaps other parts would require special con[illegible]esulting in constructions not necessary for the sm[illegible].

Conv[illegible]ine 6″ × 7½″ again with the same mean pressu[illegible]onsumption as the 12″ × 15″ engine and based [illegible] detailed design could run up to 2000 R.P.M. befo[illegible] C value of 94,000.

[illegible] Experiments with a plain cylindrical bearing (w[illegible]mferential oil groove) supplied with oil under [illegible]w that if oil is introduced at 180° from the load [illegible]tributed by an axial groove, then the flow through [illegible]g is given by an equation of the approximate form :

$$Q = Q_o + Ap$$

[illegible] p = pressure and Q_o and A are constants.

[illegible]r a wide range of loading Q_o is given approximately by the [illegible]duct :—

Radial clearance × bearing width × peripheral speed. The constant "A" depends on the diameter and clearance of the bearing and the extent to which the axial groove facilitates the escape of oil.

The quantity Q_o represents the flow of oil involved in the formation of a load supporting film and eventually extruded from the ends of the bearing mainly in the neighbourhood of the maximum pressure. It represents the oil flow from a ring lubricated bearing in which the oil is simply deposited at the tops of the shaft under no pressure. In earlier types of Diesel engines with wide bearings, such bearings worked admirably without oil coolers, and bearing wear and shaft wear was very small since abrasive grit from the cylinders did not reach the bearings.

With forced lubricated bearings the second term Ap represents an additional flow the chief function of which is to cool the bearing. This additional flow may become excessive if the bearing clearance is unnecessarily large or if axial grooves or pockets are carried too near the edge.

The behaviour of journal bearings under constant loading has been widely studied both experimentally and theoretically and operating film thicknesses are readily calculable for all ratios of diameter to width, but less is known about the behaviour of bearings under the variable condition of pressure which exist in the bearings of internal combustion engines.

One fact stands out; examination [illegible] and crank pins seems to show that the position [illegible] n rubbing is invariably in line with, or nearly in line [illegible] ection of resultant centrifugal action. It seems prob[illegible] e mean inertia force due to reciprocating parts be ad[illegible] tant centrifugal effect of rotating parts, and trea[illegible] tant revolving load, then the disturbing effect of g[illegible] may be safely ignored and may even be beneficial. T[illegible] nt effect of centrifugal and inertia loading is seen [illegible] clearly in a journal between two cranks spaced [illegible] angle less than, say, 70°. Such bearings have [illegible] suffered from persistent "wiping" and the suggestion [illegible] put forward that the line of nearest approach of the jo[illegible] the bearing revolves with the shaft and that failure of [illegible] film occurs when the line of load passes across an axial g[illegible] Such axial grooves or such pockets are therefore under susp[illegible] and there is a tendency to reduce them to a minimum in suc[illegible] way that there is sufficient surface to support the load in wha[illegible] ever direction the resultant load may point.

The conventional provision of a central circumferential groove divides a bearing into two narrows bands and this reduces the minimum film thickness as compared with an ungrooved bearing. To mitigate this disadvantage the groove is sometimes interrupted in the regions of maximum pressure or omitted altogether in favour of a tapered fan-shaped recess in way of the oil entry hole. This involves supplying the connecting rod with oil by another path, e.g. via the cross-head.

There is ample scope for research on bearings subject to loads varying in amount and direction as well as for simplification of the construction of bearings.

The lubricating oil piping system includes a distribution main of generous diameter to avoid excessive pressure drop along its length, with comparatively small bore branches to the main and subsidiary bearings. A pressure drop of, say, 10 lb./in.^{2} through these branches helps to equalise distribution and maintain a reasonable pressure on the gauge even if the resistance of the bearings themselves is small.

For small subsidiary bearings the use of laminated strip or sintered material is worth consideration. Ball and roller bearings should be generous in relation to the load carried since vibration unavoidable in the structure of an internal combustion s liable to have a peening action between the balls or nd the races.

tion.—Questions of engine balance and torsional of crank-shafts are now well understood and have been ed in great detail in special treatises. Apart from in securing dynamic balance and adjusting frequency, e weights may serve a useful purpose in reducing the loading of bearings or at least that persistent part of the loading which is due to centrifugal force.

Axial vibration of crank-shafts has occasionally come into prominence; the shape of a crank-shaft is such that torsional vibration and axial vibration must inevitably occur together in some measure. It seems doubtful if axial vibration can occur as a major problem with torsional vibration as merely a minor accompaniment, unless the crank webs are unusually thin or otherwise unusually flexible in respect of bending. For most ordinary proportions of cranks the customary precautions against torsional vibration appear to be a sufficient safeguard against axial vibration.

The transverse vibration of the framework of an engine presents a more difficult problem for precalculation particularly in relation to engines installed on board ship. Excessive transverse vibration is avoided in practice by adopting a wide spread of frames and bedplate, the use of engine columns designed for great stiffness in relation to transverse bending, and the provision of strong seatings. The principle of similitude provides a powerful means of utilising the experience of one successful installation in designing another of a different size.

Other forms of resonance are possible among the working parts of an engine and may arise occasionally, e.g. whipping of long tie rods, surging of springs, etc. The stresses in the connecting rods and pistons rods of engines running at piston speeds of about 1200 ft./min. have been measured whilst the engine is running by means of strain gauges and an oscillograph and the general results are in agreement with static calculations based on the indicator diagram. With ultra high-speed engines the possibilities of resonant vibrations amongst the running parts would be greater. A general criterion as between geometrically similar engines would be inertia stress.

Materials.—The journal pieces of large marine engine crank-shafts are almost always made of low carbon Siemens-

Martin steel having a tensile strength of 28 to 32 t
The webs and crank pins may be made of the same mat
there is an increasing tendency to use mild steel cas
these parts and to cast balance weights and crank
eccentrics (if used) in one with the webs. Wear of c
and journals in service is almost negligibly small (of t
1/1000" per annum).

For medium sized engines the crank-shafts are commonly forged solid of low carbon steel or semi-built with crank webs and crank pins in one piece of forged or cast steel. Resistance to abrasive wear is increased by using medium carbon steel (about 0·4% carbon).

Small high speed engines, particularly those fitted with lead bronze bearings, may require crank-shafts having a higher degree of surface hardness conferred by heat treatment, flame or induction hardening, or nitralloy hardening.

For connecting rods mild steel is commonly used, but the use of medium carbon steel permits a reduction of section width and weight.

When weight reduction is important the use of alloy steel may be advantageous, so long as it is confined to parts in which the major stresses occur at sections free from stress raising characteristics or at which there is little or no reversal of stress.

Where there is a large extent of stress reversal combined with unavoidable stress raisers in the form of holes, threads, shoulders, etc., it seems at present doubtful if alloy steels have so great actual endurance as plain low carbon steel. In most of the applications in which alloy steels have proved their usefulness this unfavourable combination has been absent. For example, in a locomotive coupling rod the inertia bending stress at the middle section is completely reversed every revolution but the section is quite uniform and devoid of stress raising holes, grooves, etc. In bolts and studs where the threads, and other sudden variations of section provide considerable stress raising characteristics, the variation of stress during working is usually a small fraction of the stress due to tightening.

The endurance and wearing properties of cast iron have been greatly augmented by improved foundry technique giving closer control over composition and constitution, as well as by the use of alloying constituents such as nickel, molybdenum and vanadium. This progress has enabled rising cylinder pressures

...arried without corresponding increases in the weights of ...ates and frames, and has increased the life of parts subject ...gh temperatures, e.g. pistons and cylinder covers. The ...parative insensibility of cast iron to notch effects and its high damping capacity makes it possible to consider the use of high grade cast irons for the smaller sizes of crank-shaft. The comparative toughness of some of the newer cast irons enables them to be used for a variety of subsidiary parts previously made of steel or bronze.

For light weight Diesel engines for road and rail traction, for high speed motor boats, etc., aluminium and magnesium alloys are used for structural and other parts as in established petrol engine practice. In addition, aluminium alloy pistons are in use up to 20″ in diameter or more, with advantage over cast iron in respect of general weight reduction, reduction of inertia stresses, quieter running in some instances, and reduction of the hazards of seizure.

Some aluminium alloys are satisfactory bearing metals in conjunction with hardened steel, e.g. gudgeon pin bushes.

For large engines tin-base whitemetal is still the favourite bearing metal for main bearings and big ends. For smaller engines lead base bearing metals of higher melting points are also used. In many high speed engines for road traction, etc., the cracking of bearing metal has been overcome by the use of steel bearing shells lined with lead bronze in conjunction with a hardened crank-shaft.

Manufacture and Design.—The Central European tradition of machine design was characterised superficially by a suavity of outline in mass and detail and a high degree of standardisation of recurrent problems of detail design. The earliest Diesel engines were built under the full sway of this tradition which had many notable achievements to its credit. This tradition was never fully accepted in Great Britain and Scandinavia, still less in the U.S.A., and its force is now spent.

The structure and details of a Diesel engine are now designed from considerations of function (including space requirements, accessibility and durability) and cost of manufacture. It may justly be argued that these considerations have always guided the course of design, but the force of competition and the proved success of original ideas in all departments of mechanical engineering during the past thirty years or so, have compelled designers to adopt a receptive attitude towards any pr idea however unconventional.

It has sometimes happened that modifications in carried out primarily to reduce cost have resulted in performance; it has also happened that improved workshop techniques devised to improve the accuracy of machine work have reduced costs. Such experiments have led to a continual search for methods of reducing costs by attention to details of design.

A draughtsman making the drawing of an engine detail should have a clear mental picture of the operations required in manufacture including the moulding of any castings. His aim (or one of them) should be to keep the number of operations to a minimum and to avoid slow or expensive operations.

Castings should be shaped for easy moulding with no unnecessary cores, drawbacks or loose pieces on the pattern, and no superfluous ribbing. Ribs are frequently overdone, though some ribs, functionally redundant, may facilitate flow of metal and thus contribute to sound casting.

Much unnecessary machining of steel parts can be cut out by use of drop forgings if the numbers justify the cost of dies.

In small details effective use can be made of bright bars of various sections, extruded metal strip, and tubes of various sections, etc.

Many engine builders incorporate in their designs, accessories and fittings made by specialist firms, thus reaping some of the benefits of large scale production.

In arranging drilled and tapped holes some economy may be effected by avoiding where possible oblique holes and a needless range of sizes. Operations of chamfering, etc., if indicated on the drawing, and carried out on the machine, may save time in fitting and trouble later on.

In the design of large frames, bedplates, etc., cost may be saved by keeping planed or milled surfaces to as few planes as possible, but in such cases consultation with the shop management is essential as everything depends on the machine available and the method used. What appears to the designer an obvious saving may turn out to be the opposite.

These few examples may serve as pointers towards a subject of primary importance to all designers of machinery.

Mechanical Science.—Machine design is an art, which enlists in the service of its own purposes the available results of all the sciences which it can use. These experimental and mathematical sciences have been greatly extended in scope and applicability during the last few decades. Such studies as Fluid Mechanics, Film Lubrication, Heat Transfer, Acoustics, Mechanical Vibration, Strength and Elasticity of Materials, have been transformed; partly by the accumulation of new experimental data, partly by the theoretical investigation of new problems, and partly by a reorganisation of the forms in which the subjects are expounded in books and papers. Engineers have contributed much towards these results. Also the essential underlying apparatus of mathematics has been clarified for the benefit of engineers.

It would not be difficult to point out gaps—things which an engine designer would like to know, and which no scientist can tell him—nevertheless it is probably true to say that for every problem in engine design there is a developed branch of science which can be of help and point the way to its solution by systematic research.

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