No. 8

HIGH SPEED HEAVY OIL ENGINES

In a previous paper (No. 12, page 65) a series of lectures by Mr. H. R. Ricardo was published in which the problems involved by speeding up the Diesel cycle to a high rate of engine revolution were discussed, and the lines upon which these engines were developing were indicated. More than 100 engines of this type are now on service or under construction for use in motor boats and over 100 for electric generators and it is proposed to give some details of the types that are being fitted, together with some general notes on their operation and upkeep.

High speed heavy oil engines may be classified by the design of the combustion chamber into the following categories:—

(1) Ante-chamber type.
(2) Direct injection type with masked inlet valve.
(3) Direct injection type with sleeve valve.
(4) Comet head type.
(5) Clere Story type.
(6) Air cell type.

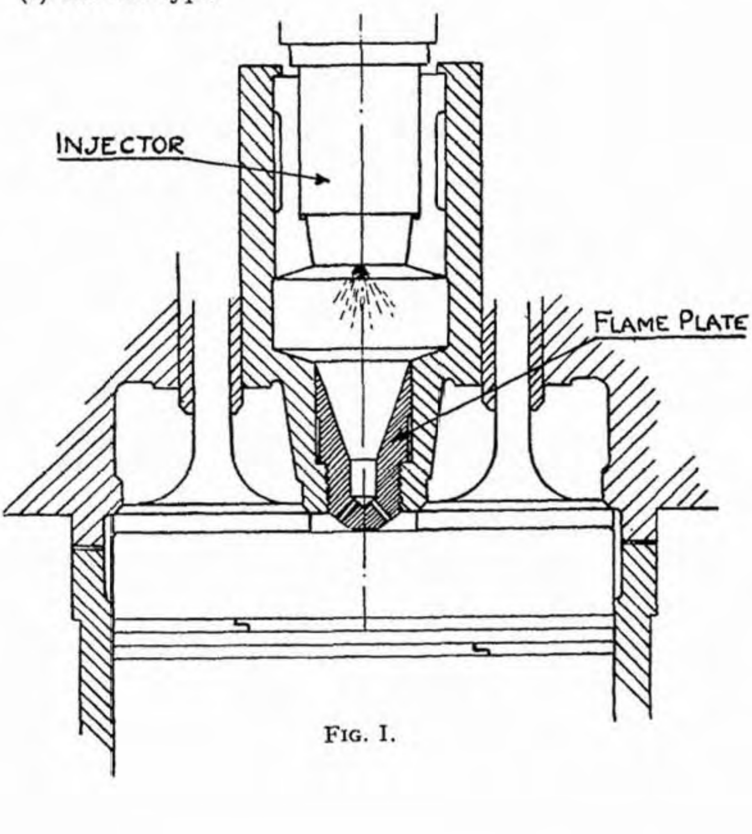

FIG. I.

(1) The only representative of this type in the Service is the McLaren engine made under Mercedes Benz patents. This engine is of $5\frac{5}{16}$ in. bore by $7\frac{7}{8}$ in. stroke developing 15 B.H.P. per cylinder at 800 r.p.m., though recently the engine has been speeded up to 1,000 r.p.m.

Fig. I shows the arrangement of the combustion chamber of this engine. Air is forced into the antechamber during compression and then fuel is injected at a comparatively low pressure (about 1,000 lb./sq. in.) through a single orifice of the order of $\frac{1}{8}$-in. diameter. As the piston starts its downward stroke the partially burning fuel and air in the antechamber are intimately mixed in their passage through the holes in the flame plate which separates the antechamber from the engine cylinder. The engine is arranged for air starting, air being supplied direct to the cylinders; hand starting is out of the question for an engine of this size, but even if a smaller engine was produced on these lines it would not be an easy starter due to the loss of heat incurred by forcing the air through the holes of the flame plate into the antechamber.

The heat loss incurred at the flame plate on both the compression and the working strokes, and the rough and tumble mixing of fuel and air renders the engine incapable of very high ouptuts. A brake mean pressure of 80 lb./sq. in. is about the highest that can be expected with a clean exhaust. The heat losses also affect the fuel consumption, a good figure for this type being ·45 lb./B.H.P.-hour.

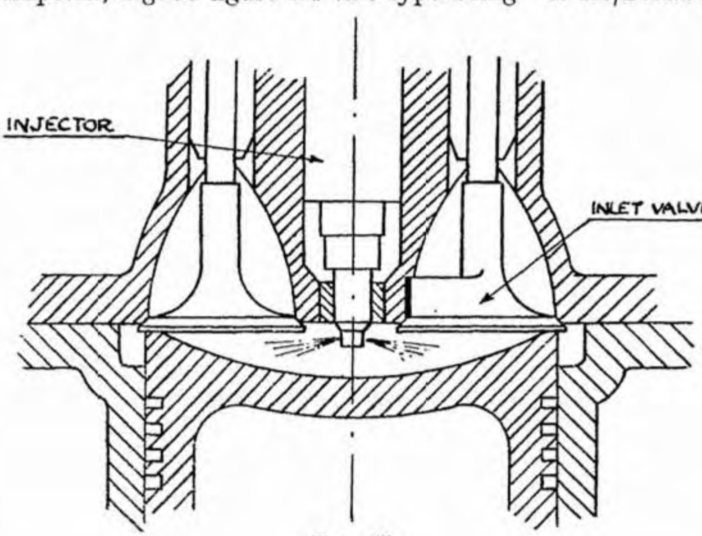

FIG. II.

(2) Fig. II shows the combustion chamber of a direct injection engine where the masked inlet valve ensures tangential admission of air during the induction stroke; the resultant air swirl continues

on the compression stroke so that when fuel is injected the air assists in seeking out the fuel rather than the fuel having to find all the necessary air for combustion.

There are two engines of this type at present in the Service, the Gardner and the Ferry F.D., each of which has a bore of 4¼ in. and stroke of 6 in. and develops 9½ B.H.P. per cylinder at 1,000 r.p.m.

The comparatively low rate of swirl produced by a masked inlet valve results in comparatively little heat being given up to the cylinder walls on compression and therefore this type of engine is a very ready starter from cold. At the same time, however, the complete mixing of fuel and air cannot be effected rapidly enough by the slow air swirl, and the fuel must therefore be well atomised and given a high velocity at the injector nozzle. This is effected by the employment of a high injection pressure and small nozzle holes.

The small diameter of these holes has in some cases led to difficulties. Care is necessary to see that the correct fuel is employed and that it is properly strained. Frequent cleaning of the injectors with paraffin and with the cleaning needles supplied for the purpose will greatly reduce the prevalence of choked nozzles and fractured fuel pump drives. It should not be necessary to point out that the cleaning needles require careful handling, but cases have come to light where the engine has been run with one or more nozzles blocked by a piece of broken needle.

Well designed engines of this type are capable of high outputs, brake mean pressures of 95 lb./sq. in. being recorded with a clean exhaust.

A satisfactory feature of the type is the comparatively small cooling effect of the cylinder walls on the air during compression, by virtue of which a compression ratio as low as 13 : 1 can be employed

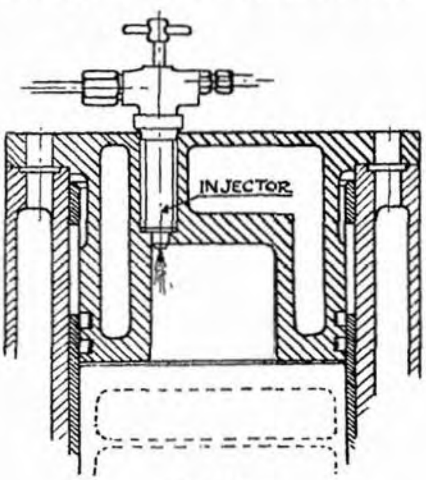

FIG. III.

while maintaining easy starting. The fuel consumption is particularly good, a figure of ·39 lb./B.H.P.-hour being readily obtainable.

(3) Fig. III shows the combustion head of the Ricardo sleeve valve engine, which is the only engine of this type at present on the market. They are made in three sizes, $5\frac{5}{16}$ in. by $8\frac{1}{2}$ in. developing 20 B.H.P./cyl. at 1200 r.p.m., $7\frac{1}{2}$ in. by 12 in. developing 50 B.H.P./cyl. at 900 r.p.m., and $8\frac{7}{16}$ in. by $13\frac{1}{2}$ in. developing 60 B.H.P./cyl. at 800 r.p.m. The engines are manufactured under licence by Brotherhood, Vickers, and Mirrlees, though the latter firm appears to have dropped this type in favour of the Comet head design.

In the sleeve valve type of engine the air is admitted tangentially to the cylinder bore, through ports in the sleeve. In this way an orderly air swirl is promoted, and since a considerably higher rate of swirl is possible than with engines of type (2) the very small diameter nozzles are unnecessary. Many of these engines operate with an injector consisting solely of a large diameter nozzle with a simple non-return valve, instead of the spring loaded needle valve usual to high speed engines; in these cases fuel is injected at a

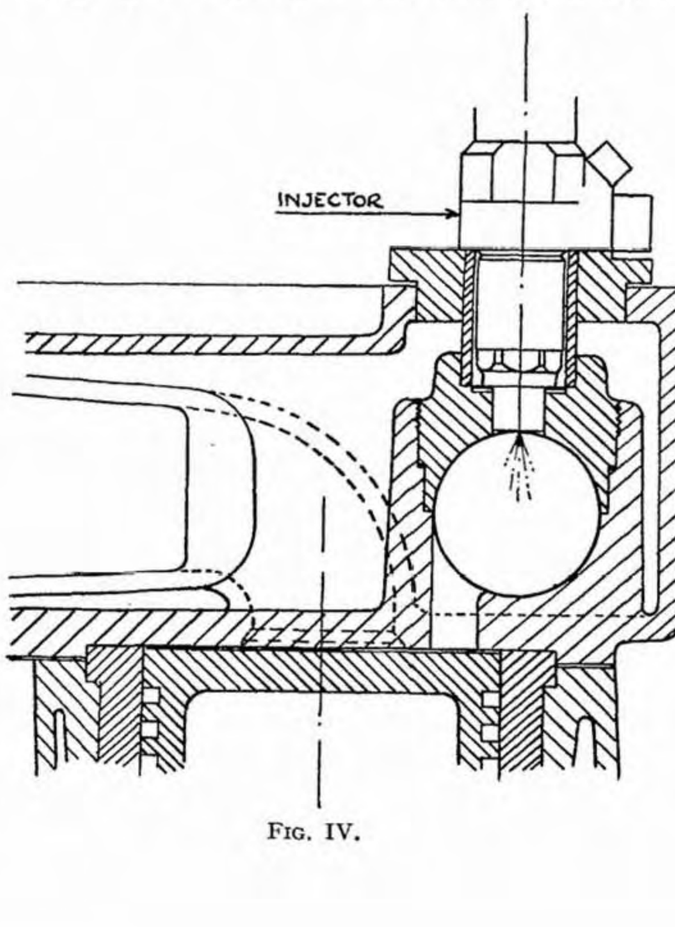

FIG. IV.

pressure only slightly in excess of the cylinder pressure, and little or no atomisation of the fuel is obtained at the nozzle. The freedom from trouble experienced with the normal type of spring loaded injector has however lead manufacturers to employ this type in Ricardo sleeve valve engines, a modification which should lead to a slightly improved performance and would also appear to render the engine suitable for a wider range of fuels.

Fuel consumptions below ·39 lb./B.H.P.-hour have been recorded from this type of engine, and the exhaust remains clear at 90 lb./sq. in. brake mean pressure. The engines are too large to permit of hand starting, air starting being generally provided, those of Messrs. Brotherhoods' design being provided with a separate air motor driving on to the flywheel through a Bendix pinion.

(4) Fig. IV is typical of the combustion chamber design of the fourth type of engine, though different makes vary as to the angle at which the communicating passage enters the cylinder, the direction of the fuel spray (against, across, or behind the air swirl), and the degree of cooling applied to the walls of the combustion chamber, while in some engines a cylindrical chamber is used.

The engines of this type that are at present on service or on trial are :—

Maker.	Bore.	Stroke.	B.H.P./cyl. at 1000 r.p.m.
	In.	In.	
Ferry S.D.	3¾	5¼	5½
Lister	4½	5½	9½
Thornycroft	4	6	7½
White	3¾	5	5½
Paxman	5⅝	5⅞	10

The essential feature of this type of engine is that the flow of air in the combustion chamber shall be so controlled both in direction and velocity that all the fuel injected is impinged upon by sufficient air for its complete combustion and as a consequence low fuel injection pressures and comparatively large diameter nozzles can be employed.

The high speed of the air over the surface of the combustion chamber facilitates the transfer of heat from the compressed air, and, despite the use of a compression ratio of the order of 15 : 1, starting from cold is not easy. This difficulty increases as the cylinder size is reduced due to the increased ratio of surface to volume of the combustion chamber, and in consequence a 16 : 1 compression ratio is usual for the smaller sizes of engine. In order to ensure hand starting a tinder cartridge is generally arranged in the combustion chamber to supply additional heat to the compressed air ; such a fitting is acceptable for an electric generator engine but is not desirable for motor boat engines due to the difficulty of lighting and inserting the cartridges simultaneously in a high wind with motion on the boat, electric heating plugs are therefore supplied to engines requiring such aids when fitted in motor boats.

Two of the above engines make different provision to overcome the starting difficulty. The Thornycroft engine runs on a normal compression ratio of 19 : 1 which is found to give easy starting without additional heat being supplied, but this involves greater bearing loads and the engine must be designed accordingly. In the Lister engine the combustion chamber consists of two spheres connected by a small passage, the outer sphere being shut off for starting by a hand controlled valve and so raising the compression ratio to 19 : 1, the normal running ratio being about 15 : 1.

One of the difficulties of design with this type of engine is to provide sufficient cooling of the combustion chamber to avoid formation of carbon deposits and at the same time to avoid excessive cooling which makes starting difficult and gives rise to a smoky exhaust when running on light load for a considerable period. To overcome this difficulty it will be found that in some engines parts of the combustion chamber are uncooled or heat insulated.

This type of engine is capable of operation at high rates of revolution ; high brake mean pressures are also obtainable, but the fuel consumption is seldom below ·42 lb./B.H.P.-hour while higher figures are to be expected with the smaller engine sizes.

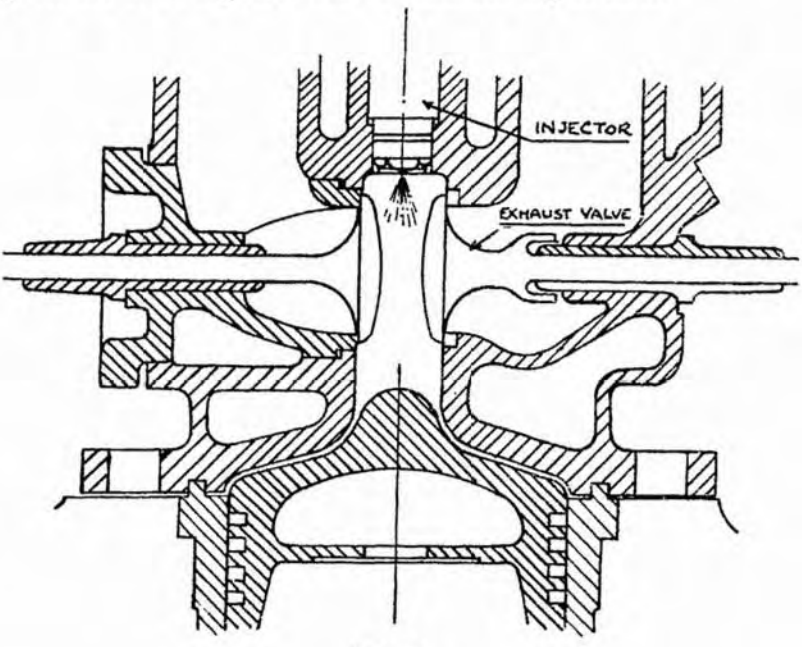

Fig. V.

In general these engines are noisier in operation than those of types 2 and 3. Diesel knock is attributed to a long ignition delay period of the fuel, ignition being delayed until near the end of injection with a consequent sudden and uncontrolled rise of pressure. Two factors which tend to reduce the delay period, namely a rapid supply of oxygen to the fuel and a high air density, are present in engines of type 4 to a greater extent than in types 2 and 3—but despite this the engines are noisier. It is thought that the increased noise is due in part to the higher compression ratio, which raises also the maximum pressure, and also to the presence of an uncooled portion of the combustion chamber which acts as a better sound box than the completely water jacketted heads of the other types of engine.

(5) The following engines of the Clere Story type (*i.e.*, with horizontally opposed overhead valves) are employed in the Service, a typical combustion chamber design being shown in Fig. V.

Make.	Bore.	Stroke.	R.P.M.	B.H.P./cyl.
	In.	In.		
Gleniffer	6	7	900	20
Paxman	6½	10	800	28
Ruston Hornsby ..	5⅜	8	1000	17½

The air in the combustion chamber is in a state of rough and tumble turbulence which is sufficient to make the employment of

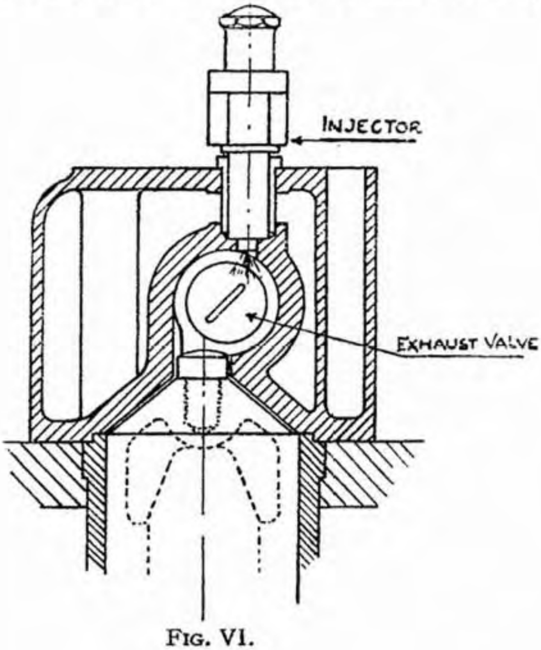

FIG. VI.

high injection pressures or small nozzles unnecessary but the performance is not outstanding in any respect, 80-85 lb./sq. in. being the highest brake mean pressure that can be expected with a clear exhaust, while the fuel consumption is seldom below ·44 lb./B.H.P.-hour except in the slower speed engines. Engines of this type are generally rather quieter in operation than those of type 4.

The great advantage of this type is that the disposition of the valves makes for a very accessible engine.

A modification of this type of engine is the R.N. engine (Fig. VI) of $4\frac{1}{8}$ in. bore by 6-in. stroke, developing 9 B.H.P./cyl. at 1000 r.p.m. The valves of this engine are offset from the centre line and the passage from the engine cylinder is arranged tangentially to the combustion chamber in order to promote an orderly but slow speed swirl.

(6) McLaren L.M. and S.L.M. engines are the only representatives of the air cell type at present employed in the Service, the combustion chamber design of the L.M. type being shown in Fig. VII. Two sizes of engine are in use, namely the L.M., $4\frac{1}{8}$-in. bore

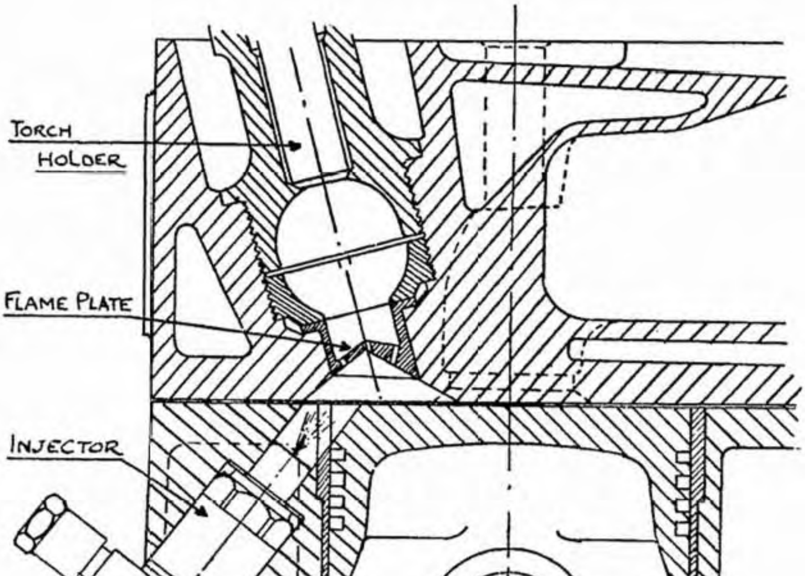

FIG. VII.

by $5\frac{7}{8}$-in. stroke, and the S.L.M., 3·74-in. bore by 4·92-in. stroke, developing $8\frac{1}{2}$ and $5\frac{1}{2}$ B.H.P./cyl. respectively at 1000 r.p.m. The S.L.M. type is arranged with a horizontal injector and specially shaped piston.

Air is compressed into the air cell and fuel is sprayed directly at the flame plate at the base of the air cell; an injection pressure of 2600 lb./sq. in. being necessary to ensure penetration of the fuel to this point. Ignition starts before top dead centre in the space between the nozzle and the flame plate, but as soon as the piston starts its down stroke air rushes from the cell to complete the combustion of fuel.

Due to the loss of heat incurred when the air is forced through the holes in the flame plate, cartridges or heater plugs are required to enable the engine to start by hand from cold, despite the 15 : 1 compression ratio employed (16 : 1 on the smaller engine). This heat loss also affects the fuel consumption, which is seldom below ·44 lb./B.H.P.-hour. Like the engines of type (4) these engines are rather noisy in operation.

Variations of the air cell type include the Acro and the Lanova designs. The former has the cell incorporated in the piston, while the latter has the injection directed towards the entrance to the air cell, producing a delay in the rise of pressure in the cylinder although very high pressures are created in the cell. This is not conducive to efficiency, but the reduction in maximum pressure in the cylinder has great advantages. Messrs. Dennis and Messrs. Meadows have taken up the patent in England.

Mechanical Details.—In their mechanical details high speed heavy oil engines follow generally upon the lines of petrol engines, but with a general stiffening up of all parts due to higher maximum pressures; whereas in a petrol engine the maximum pressure is of the order 300 lb./sq. in. the compression pressure of a high speed heavy oil engine is in the neighbourhood of 500 lb./sq. in. dependant on the compression ratio, and the normal maximum pressure 600-700 lb./sq. in. This latter pressure will be increased if the timing of fuel injection is unduly advanced, and under these conditions the running of the engine will be rough. It should be noted that the use of an unsuitable fuel may, by virtue of ignition delay, cause bad timing and consequent rise of pressure. Cases of maximum pressure rising to 1100 lb./sq. in. have been known.

One of the principal troubles experienced during the development of high speed heavy oil engines has been the failure of crankpin bearings.

In the earliest days one of the causes of big-end failure was undoubtedly whipping of the crankshaft. Crankshaft sizes were increased till the bearing diameters are now seldom less than five-eighths of the cylinder bore, an increase of 25 per cent. on petrol engine practice. Due to the increased crankpin diameter, efforts were made to reduce the thickness of crankpin bearings, so as to avoid a large increase in inertia stresses due to the heavier connecting rod. It is common practice to fit bearings consisting of a $\frac{3}{16}$-in. thick shell with $\frac{1}{16}$-in. thickness of white metal. Bearing failures with thin shelled bearings suggested that such shells were insufficiently stiff and in consequence many makers fit steel shells. In high speed engines employed for road transport work crankpin bearing troubles still occur, and many other materials are being tried in an endeavour to provide a satisfactory bearing. Among these are lead bronze, cadmium bronze, and duralumin.

Service experience with aluminium pistons has not been uniformly satisfactory, especially in small engines, and as a general rule cast-iron pistons are fitted to engines employed in motor boats or for electric generators. Gudgeon pins are generally of the fully floating type and, as a rule, 3 pressure and 2 scraper rings are fitted to the piston.

Cylinder wear is a matter of great importance with this type of engine, because if allowed to proceed too far its adverse affect on the compression pressure will prevent the engine from working. It appears, however, that provided the piston rings are kept in good order, considerable cylinder wear is required to affect adversely the compression pressure, for examination of an engine which had been giving no trouble in hand starting from cold at normal temperatures was found to have the cylinder bores worn 19/1000 in. on $4\frac{1}{4}$-in. diameter.

The high maximum pressures acting behind the piston rings tend to produce more rapid cylinder wear than in petrol engines, and in order to reduce this wear renewable liners with special wear-resisting qualities are usually fitted. Recent theories, which are corroborated by experiment, indicate that cylinder wear may in part be due to condensation on the cylinder walls and can be considerably reduced by maintaining a high circuiting water temperature. With this end in view a hand controlled byepass valve is now usually fitted in the circulating water system so that hot water leaving the jackets may be re-circulated through the system. The circulating water, if drawn from the sea, should not, however, be allowed to exceed a temperature of 140° F., as above this temperature the deposition of scale in the water passages will occur at a rapidly increasing rate.

Generating sets are now being provided with a dual system of water cooling, wherein distilled water only is passed through the jackets. Neither boiler-feed water nor other fresh water which may contain lime should on any account be used. The system is provided with control valves and thermometers in order that the temperature of the jackets can be controlled at will.

The more complete combustion of fuel in the Diesel cycle results in cooler exhaust gas temperatures than occur in petrol engines, and this should result in a considerably longer life for the inlet and exhaust valves. The results obtained so far in this respect indicate that the heat resisting steels, which were found very satisfactory in petrol engine practice, are liable to local pitting in heavy oil engines due to the trapping of hard carbon particles beneath the valve seat. The present trend of design appears to be towards the use of case-hardened steels, which were definitely not satisfactory in petrol engines.

Maintenance.—Apart from the fuel injection equipment (which will be dealt with separately) and the upkeep of valves and bearings, the efficient upkeep of these engines resolves itself into maintaining the necessary compression pressure. In the normal course of events loss of compression is insidious and will at first be more apparent in starting difficulties than in bad running. For this reason any engine which is fitted with alternative hand and power starting arrangements should be started by hand from cold at regular intervals.

A point often ignored in the matter of starting by hand is that, since it is temperature and not pressure which causes the fuel to ignite, a big reduction in effort can be effected by throttling the air inlet. Half a charge of air will require only half the work to compress it, but the ratio of compression and hence the temperature at the end of compression is no less than when a full charge is drawn in. As soon as the engine fires, of course, the full air charge must be admitted.

A useful compression pressure indicator can be made by arranging a pressure gauge with a light plate-type non-return valve and an adaptor to fit in place of the fuel injector. The non-return valve should be fitted so that it is as near the combustion space as possible in order to avoid an undue increase in compression volume, and a relief cock should be fitted on the pressure gauge side of the non-return valve. The engine is then run with the adaptor fitted to each cylinder in turn, and the steady reading of the pressure gauge noted. This record should be for comparison purposes only and must not be taken as an accurate reading of the actual compression pressure.

The thickness of the cylinder head gasket is of vital importance in obtaining the correct compression ratio; when refitting these engines cylinder head gaskets should not be made of normal store materials unless it is certain that the material when compressed is of the correct thickness and in general the gaskets should be included in the special store list and obtained from the makers. When renewing gaskets of small engines care should be taken to check the truth of the faces of cylinder block and head, since it has been found possible to distort these surfaces by excessive screwing up on the cylinder head studs.

The quality of the lubricating oil in high speed heavy oil engines requires careful watching. Unlike the petrol engine there is little dilution of lubricating oil by fuel, and the oil is therefore more apt to become gummy and to form a carbonaceous sludge despite the fitting of adequate filters in the lubricating system. Such deterioration if not kept in check will make the engine hard to turn when cold, and, in the case of motor boat engines with built-in reverse gears, may lead to clutch troubles.

Fuel Injection Equipment.—In large Diesel engines the supply of fuel to the cylinders is arranged on one of two systems. In the older airless injection system, fuel is supplied at high pressure to a common rail which feeds all the fuel valves, the timing of fuel injection and the control of the quantity of fuel injected being carried out by mechanically operated fuel valves. In the blast injection system the metering of fuel is carried out by the fuel pump. The pressure required for fuel injection is provided by blast air, while the timing of fuel injection is still controlled by mechanically operated fuel valves.

Airless injection of fuel is universal in high speed heavy oil engines owing to the noise, complication, and reduced durability anticipated with a small high speed air compressor, although blast injection was employed in one of the earliest experimental engines. Owing to the small cylinder sizes employed with this type of engine a mechanically operated fuel valve is undesirable, and is replaced by an automatically operated spring loaded needle valve, or fuel injector. The fuel pump has therefore to meter out the correct quantity of fuel to each cylinder at any particular loading of the engine, to supply this fuel at the necessary pressure for injection, and also to effect the timing of fuel injection.

There are many different fuel pumps on the market, but in nearly every case they operate on the same principle. A constant stroke glandless plunger pump is employed, and the suction valve or port is kept open until the plunger has attained approximately its maximum velocity on the upward (supply) stroke. With the closing of the suction valve fuel is delivered through a light spring loaded non-return valve, and when the necessary pressure has been generated the spring loaded fuel injector lifts. After a period determined by the fuel pump control setting, a spill valve or port is opened which bye-passes the pump discharge back to suction; the spring loaded discharge valve and the fuel injector then close and injection is complete.

The accuracy to which the fuel pump and injector must be made will be appreciated when it is realised that, at light loads, the quantity of fuel delivered to each cylinder per cycle is about the size of a pin's head, the plunger diameter being about $\frac{1}{4}$ in., and the permissible variation in quantity of fuel supplied being less than 5 per cent. at any setting of the pump control.

The pump in most general use is the C-A-V Bosch, which is

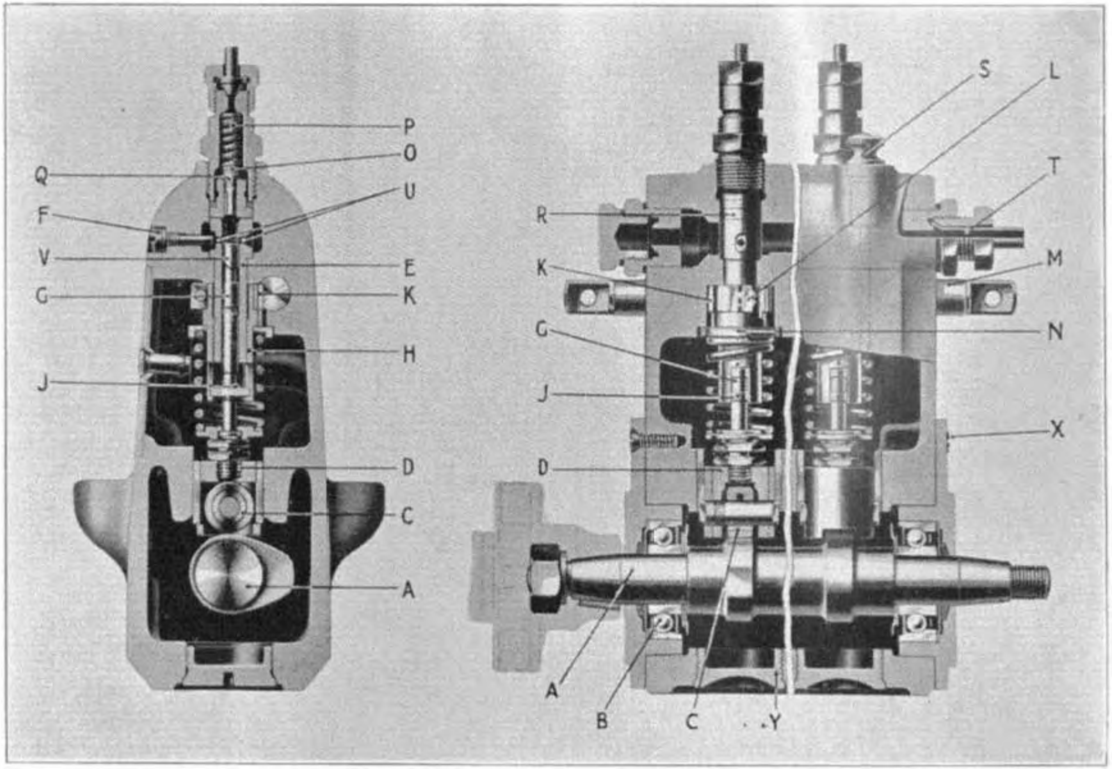

Fig. VIII.

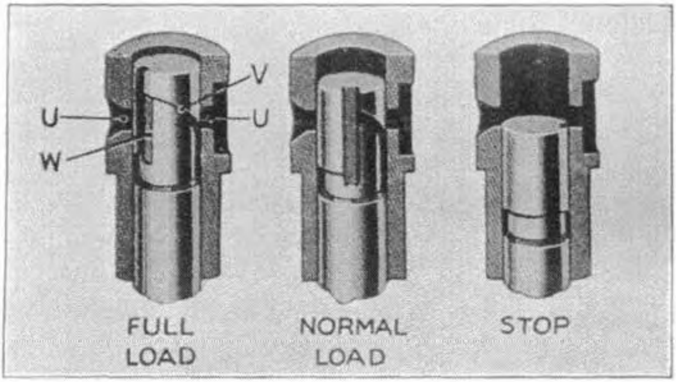

FIG. IX.
Operation of Fuel Pump Plungers.

made in England. Other makes are equally satisfactory in operation, but the majority are produced only in small quantities. The production in large numbers of the very accurately lap fitted parts requires a technique which can only be acquired with experience backed by considerable resources, and in this direction Messrs. C-A-V Bosch have gained a substantial lead over their competitors as a result of their early anticipation of requirements. The following description is therefore devoted chiefly to this type, although it must be remembered that other makes are likely to be introduced during the next few years.

The arrangement of the pump is shown in Fig. VIII. The camshaft (A) is carried on ball bearings (B) and runs in an oil bath, the cams bearing against steel rollers (C) of the adjustable tappets (D). The plunger barrels (E) are fixed in the casing by the set screws (F). The plungers (G) are located in the regulating sleeves (H) by the lugs (J) on the plungers, which operate in the slotted sleeves (H). These sleeves carry toothed arcs (K) secured by the clamping screws (L), a rack on the control rod (M) meshing with the teeth on the arcs (K). The return springs are retained between collars (N) in the housing and on the lower end of the plungers. Held in place by the delivery valve holders or unions are the delivery valve seats (R) with the delivery valves (Q) and their return springs (P). The oil level in the camshaft oil bath can be checked by the dip stick (S) and the bath should be kept filled to the correct level with special service mineral oil; the plungers do not require lubrication.

The operation of the fuel pump can be seen from Figs. VIII and IX. Fuel enters the suction chamber which is common to all the plungers at (T). When the plungers are on their downward stroke two ports (U) are uncovered and fuel is drawn into the pressure chambers above the plungers. On the plungers being moved upwards by the cams, the ports (U) are first closed and fuel is forced past the delivery valves (Q) to the injectors. Immediately the helical edge (V) on the plunger uncovers the spill port on the right, the pressure chamber and suction chambers are put in communication by way of the vertical groove (W), and all pressure being lost, no further fuel passes the delivery valves. When the control rod (M) is moved endwise, the regulating sleeves and therefore the plungers are rotated, and thus the time of uncovering the spill port by the helical edge (V) is varied. This has the effect of varying the effective stroke of the plunger and thus controlling the quantity of fuel delivered to the engine. In the "stop" position the vertical groove (W) is always in communication with the spill port.

Bosch Fuel Injector.—Fig. X shows the Bosch fuel injector which is simply a spring loaded needle valve. Fuel is supplied from the fuel pump to the union connection (A) whence it passes by a drilled passage to an annular space above the seat of the needle valve (D).

The injection pressure is determined by the loading on the spring (E), and a feeler pin (H) is provided to ascertain whether the injector is working when the engine is running. Owing to the high injection pressures employed, the needle valve has to be an accurate lapped fit in its guide (C), which in this case is the nozzle body ; any fuel which does leak past the body of the needle valve is led off from the union connection (G) to a suitable collector.

The rate at which pressure builds up in the fuel system after the suction ports of the fuel pumps are closed is very rapid, due to the high plunger speed at this point, but it is also essential to arrange that the drop in pressure at the end of injection shall also be rapid so as to ensure that the needle valve of the injector closes rapidly, so preventing a dribble of fuel which would carbonise and block up the injector nozzle. To achieve this rapid pressure drop the delivery valve of the Bosch fuel pump is of a special design ; below the mitre face of the valve a piston-like plunger is fitted which is a lapped fit in the valve seat. At the commencement of fuel delivery from the pump the delivery valve has to lift until the piston skirt is above the valve seat before any fuel passes to the injector ; at the end of injection the delivery valve has to fall this distance when re-seating itself and the space in the delivery system is thereby suddenly increased, causing an extremely rapid fall of pressure at the injector.

Solid drawn thick walled steel tubing is fitted for the delivery pipes from the fuel pump to the injector in order to minimise pipe expansion during the period of fuel delivery, this expansion being of considerable importance at low pump outputs. Some manufacturers stress the importance of providing exactly equal lengths of fuel piping to each injector in order to obtain identical injection timing at each cylinder.

Of the other makes of fuel pump that may be encountered in the Service those showing the greatest differences in design to the Bosch pump are the Ricardo and Ruston-Hornsby pumps. In each case a plain pump plunger is employed together with a separate mechanically operated spill valve. Only small differences in fuel injectors are to be found, the principal one being an injector in which the lapped portion of the needle valve body is arranged at some distance from the nozzle. With this design it is claimed that there is less liability of the needle valve becoming gummed up, while it is also possible to grind in the needle valve seat. In many designs of injector no adjustment for injection pressure is provided.

All fuel pumps require to be supplied with fuel under a head of at least 9 in. to ensure that the suction chamber of the pump is always full. Exclusion of air from the system is of the utmost importance since the compressibility of a small air pocket would prevent delivery of fuel. Venting arrangements are provided in the fuel system and these should be used whenever the set has been idle for more than a few weeks.

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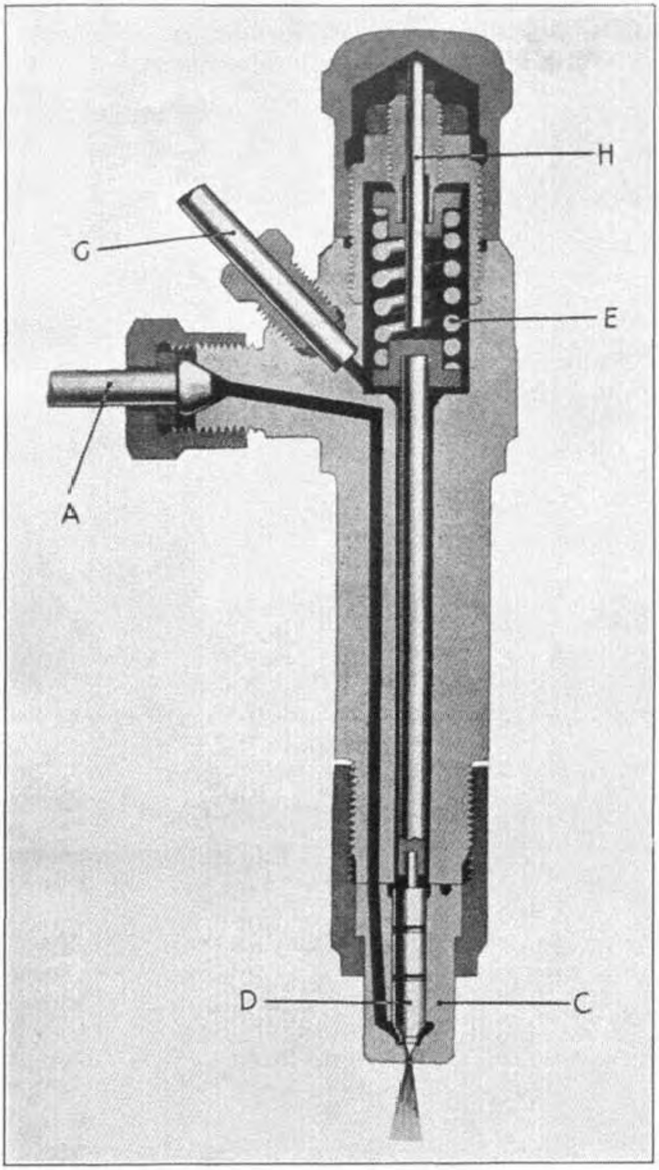

FIG. X.
Bosch Fuel Injector.

Exclusion of dirt from the fuel system is of the utmost importance owing to the damage that may be caused to the lap-fitted surfaces by any solid matter. Fuel filters are always fitted in the suction line of every fuel pump and either fabric screen or Auto-klean type filters will generally be found in the Service. These filters must be examined and cleaned regularly. An additional filter is often found in connection with injectors where very small nozzle orifices are employed (generally fitted on those engines referred to as type 2), a fluted bar being arranged in the union connection of the injector to give edge-filtration of the fuel supplied.

When an engine is shut down for sufficient length of time to justify the work, it is advisable to remove the injectors and clean in paraffin, as others there is a tendency for the small orifices to become carbonised owing to the residual heat of the engine.

Maintenance of Fuel Injection Equipments. (1) *General.*—The performance of high speed heavy oil engines depends largely upon the efficiency of the fuel injection equipment. A smoky exhaust or uneven firing is generally due to a defective fuel injection system or to lack of compression. The fuel injection system should therefore be examined after every 1200 hours running, and whenever the exhaust becomes dirty or uneven.

When dismantling any part of the fuel injection system great care must be taken to exclude any foreign matter, however fine; parts should be washed in clean paraffin and assembled after further rinsing in paraffin or fuel oil.

The component parts of any one injector or of any one element of a fuel pump must be kept together and not associated with similar parts of another unit, as such items are seldom interchangeable. For the same reason, if it is necessary to fit a new spray valve spindle, or a new pump plunger, the spare spray valve spindle and its associated guide, or the spare pump plunger and its associated barrel must always be fitted as a unit.

(2) *Fuel Injectors.*—To test the injectors first remove them from the engine and re-connect them to their fuel pipes, taking care to prime the system thoroughly to ensure that all air is expelled. Turn the engine and observe the fuel sprays; these should be of similar form and equal intensity. Further, there should be no dribble from the nozzle at the end of injection, no matter how slowly the engine is turned.

If the above requirements are met, both the fuel pump and the injectors may be considered satisfactory and the injectors can be replaced without adjustment.

A distorted spray is due as a general rule to a choked nozzle, unequal penetration, to a worn injector valve spindle, an incorrect injector spring setting, or a worn fuel pump plunger; dribbling, to a sticky injector valve spindle or a defective fuel pump discharge valve.

If it is observed that the fuel sprays are not satisfactory, the fuel injectors should be examined first; the nozzles should be cleaned with the tool provided and the injector valve spindles and their guides cleaned and replaced. The injector discharge pressures should then be tested with a suitable pressure testing pump.

If the injection pressure does not agree with the maker's recommended figure, spare injector parts must be fitted, unless means are provided in the design (*e.g.*, Bosch) for making the necessary adjustment.

Grinding in the fuel injector needle valves should only be resorted to when, on test, an injector starts dribbling at or below 75 per cent. of its discharge pressure. Special tools are provided for grinding in the needle valves of Ruston and Gardner injectors. The needle valves of Bosch injectors fitted to engines of less than 10 h.p. per cylinder should never be ground in since this would result in abrasion of the valve spindle and guide and cause poor fuel spray penetration.

(3) *Fuel Pumps.*—Fuel injection pumps are provided with two adjustments, (*a*) the pump plunger tappet, which determines the timing of each element, and (*b*) the pump control which determines the putput of the pump elements at any given setting. In the Bosch, M.L. and Benes pumps this control rotates either the plunger or the barrel and so alters the point of cut-off of fuel discharge. In the Ruston and Ricardo pumps the same effect is obtained by operating a spill valve.

With the exception of examining the discharge valves, and suction and spill valves (where fitted), fuel pumps should not be dismantled on board unless absolutely necessary, *i.e.*, unless it is suspected that a pump plunger or barrel is badly worn. Such a defect would be indicated by an injector which, while producing a normal spray when connected to the test pump, gives a weak spray when connected to the fuel pump.

Fuel pump valves should never be ground in (except in the case of Ricardo pumps, fitted on Vickers-Ricardo engines). If a valve or seat is worn or damaged a spare pair must be fitted.

If a pump plunger or barrel is found to be worn or damaged a spare plunger and barrel must be fitted as a pair; these items are however made to such limits of accuracy as to enable a spare pair to be fitted without upsetting the tappet adjustment. The fuel pump control will, however, require to be re-set.

To set the fuel pump control when reassembling the fuel pump, or to check that the control is providing equal outputs from each pump element, remove the injectors from the engine and connect them up to the fuel pump. Turn the engine and gradually move the fuel control operating lever to the "stop" position; all the injectors should cease to spray at the same position of the operating lever for a constant turning speed ; if they fail to do so the element settings must be adjusted individually until this condition is fulfilled.

The security of the fuel pump control adjustment (item L, Fig. VIII) is of the utmost importance. Should one of these slack back it is possible for that particular pump element to deliver an excessive quantity of fuel, and, as the governor would then have no control over this element, the engine would seriously overspeed as soon as the load is removed.

To check the timing of the fuel pump, the correct relative phasing of the individual elements must first be obtained. Take each element in turn and remove the fuel discharge pipe and the fuel pump discharge valve and spring, afterwards replacing the holder or union ; turn on the fuel and rotate the engine from the bottom dead centre of the compression stroke of the cylinder corresponding to the pump element being tested towards the top dead centre of the same stroke. Near the top of the stroke fuel will cease to flow ; this is the point of commencement of fuel discharge. Mark this position on the engine flywheel, and check that all the elements commence to discharge fuel at the correct phase angles to each other (*i.e.*, 180° for a 4-cylinder engine, 120° for a 6-cylinder, etc.). If the phasing of any one element is not correct the plunger tappet of that element must be adjusted accordingly.

The point of commencement of fuel discharge determined above must not be confused with point of fuel injection, the timing of which is given in some makers' instructions. There is an appreciable time lag between the two points which is dependent among other things, on the quantity of fuel contained in the fuel discharge pipes, and on the discharge pressure to which the fuel injectors are set. Special apparatus is required to determine the actual point of fuel injection and, therefore, the timing of the fuel pump must be set by experiment after the correct phasing of the individual elements has been obtained. Alteration to the timing of the pump is effected by re-phasing the fuel pump camshaft. Too early injection of fuel causes excessive combustion noise, while too late injection causes a smoky exhaust, particularly when idling.

(4) *Periodical Tests by Dockyard.*—The above tests and overhauls may be carried out on board and cover for any normal defects that may arise, but not for detection of the inevitable gradual wear of the accurately fitted fuel pump parts, for which special testing equipment is necessary. From time to time therefore all fuel pumps should be sent to a Dockyard for complete testing. The Yards are being provided with special apparatus with which it is possible to measure accurately the fuel outputs from the individual pump elements at any setting of the pump control, and thus by comparison with standard outputs, to determine the extent to which wear has occurred.

The apparatus consists of a mounting on which the fuel pump can be set up and driven by hand, fuel being discharged to a set of special testing nozzles the output from which is measured in calibrated test tubes. These special nozzles are set for an injection pressure of nearly 5000 lb./in.^{2} thus magnifying the effect of any wear that has taken place in the pump. The hand turning gear of this apparatus is provided with a simple form of dividing head to enable the relative phasing of the individual pump elements to be accurately checked.