In 1952, C.C. Pounder published a book called “Marine Diesel Engines”, and the following is an extract from Chapter V, which dealt with Harland & Wolff engines.

“TWO-STROKE, ECCENTRIC-TYPE, OPPOSED-PISTON ENGINES”

“DOUBLE-ACTING CROSSHEAD ENGINES”

FIG. 54. – DOUBLE-ACTING, TWO-STROKE, ECCENTRIC-TYPE, OPPOSED-PISTON ENGINE
Figs. 54 and 55 show the latest type of double-acting engine of Harland & Wolff design and manufacture. It is a most compact form of direct-coupled marine Diesel engine; it has a very high output density – or power-weight, power-space ratio.

The engine was evolved, in Belfast, from types which had cylinder covers and small exhaust pistons.

**Construction**

Figs. 54 and 55 are self-explanatory. The top and bottom exhaust pistons are a fixed distance apart and reciprocate together. Each exhaust piston is secured to a cast-steel yoke; the top and bottom yokes are coupled by four long rods. To each bottom yoke is attached a pair of spindles, articulated to rods which terminate in the eccentric straps. Loose collars between the bottom faces of the lower yokes and the shoulders of the exhaust piston spindles facilitate accurate assembly. The side thrust at the point of articulation is taken by a cast-steel guide shoe. The valve-spindle fork is so designed that an extra-large-diameter cross-head pin is possible, with benefit to the bearing pressure; see Fig. 56. Four carefully fitted, mild-steel bolts inside the 38 to 42-ton case-hardened carbon-steel cross-head pin hold it in the fork and sustain the shear and bending forces.
Fig. 57 shows an eccentric rod and strap. The oil grooves are in accordance with earlier practice. The latest straps have no longitudinal grooves and a circumferential groove only on the upper half.

The essential function of the exhaust pistons is to control the exhaust periods; but they also contribute to the power output of the engine. The ratio of exhaust-piston stroke to main stroke is about 1:3. The eccentric sheaves are integral with the cast-steel crank webs. Such crank webs have been widely used during the last fifteen years. Some of the webs produced under wartime conditions were distinctly below pre-war standards – which should be borne in mind if, in future, a crankshaft shows web defects, especially at the foremost cranks. The crankshafts of Fig. 55 are fully-built, made in two sections coupled together by mild-steel fitted bolts; the journal-pieces and crank pins are of forged mild steel. In strength the crankshafts are about 10 per cent in excess of Lloyd’s rules. Differences in firing order and crank sequence are responsible for some apparent anomalies in crankshaft and intermediate shaft sizes. Crank sequences and angles for six- and eight-cylinder engines, when looking aft, are shown in Fig. 58. For six cylinders the firing order is: 1T, 6T, 2B, 4B, 3T, 5T, 1B, 6B, 2T, 4T, 3B, 5B; for eight-crank engines two cylinders fire simultaneously, one top and one bottom, i.e., there are eight impulses per revolution. For seven-cylinder engines the crank sequence is: 1, 7, 2, 5, 4, 3, 6, with 51.43° between cranks; the firing order is: 1T, 4B, 7T, 3B, 2T, 6B, 5T, 1B, 4T, 7B, 3T, 2B, 6T, 5B. The angle of advance of the eccentrics is 185.5° ahead, becoming 174.5° astern. The opening and closing periods are shown in Fig. 59.

**Bedplates and Frames**

The bedplates and frames are substantially constructed in cast iron or fabricated steel. All joints and surfaces are accurately faced, metal-to-metal. The bedplate is bolted to strong seatings which are incorporated in the ship’s double-bottom; an oil-tight sump is arranged in the well of the seatings under the bedplate. The underside of the bedplate flanges is machined, and a
sealing bar is welded to the tank top around the bedplate. The holding-down bolts and cast-iron chocks are of standard form. The main bearing keeps and white-metal-lined bushes are of cast steel. The bushes are circular, with an eccentricity of 1 mm., to facilitate removal. All main bearings are the same length, and mild-steel studs hold down the keeps.
FIG. 58. – CRANK SEQUENCES

FIG. 59. – EXHAUST AND SCAVENGE PERIODS, D.A. 2C. ENGINE
Connecting-rod

The forged mild-steel connecting-rods are of normal design, with top- and bottom-end bearing bushes of white-metal-lined cast steel; the bolts are mild steel. Steel of 38 to 42 tons/sq. in. ultimate tensile strength is used for the crossheads; the guide shoes – which may be articulated – are steel castings, with white metal on ahead and astern faces. The guide plates and bars are cast iron. Crossheads and guides are marked to show when the pistons are respectively in top and bottom positions. When running ahead, the crosshead shoes of single-screw sets slide on the guide plates; those of twin-screw sets on the guide bars. Ready access to the crankcase, from the front of the engine, is thereby assured.

The vertical edges of the crosshead shoes are preferably shaped to the equivalent of a very large radius, thus preventing “cape and corner” bending loads on the piston-rod attachment as the crosshead moves up and down in the guides.

Piston-rods

The piston-rods are of forged, heat-treated, mild steel, 28 to 32 tons/sq. in. ultimate tensile strength. Each rod is provided with a stout flange at its upper end for bolting to the piston; at its lower end it is attached to the crosshead by a wrought-iron top nut and a mild-steel bottom nut, both octagonal. The threads are of special form, cut with the greatest accuracy and given an exceedingly fine finish. The threads in the nuts are equally well made, with the correct mating fit. A key prevents the piston-rod from turning in the crosshead. Each rod is enclosed in a cast-iron sleeve, which is screwed into the piston at its top end and slides freely on a gunmetal bush at the crosshead end of the rod. The annular space between sleeve and rod is utilized for conveying oil to the piston, thereby cooling the sleeve. The sleeve passes through a stuffing-box located in the lower exhaust piston and through a gland in the diaphragm plate on the crankcase top.

Thrust Block

The thrust block, which is integral with the bedplate, is normally of the design of Fig. 60, but sometimes the Michell design is used.

A single split-hollow thrust disc is arranged between two collars forged on the shaft. The white-metal bearing surface, on each disc face, is divided by radial oil grooves into sections, the mean width of which is, at most, 0.75 of the radial length. These radial grooves are continuations of helical grooves cut in the white-metal bore of the thrust disc. Oil is supplied to the helical grooves by holes drilled into the hollow casting of the disc, which serves as an oil reservoir. The radial grooves are carefully taper-scraped at one side, the oil being induced up the taper as the thrust collar rotates, thus ensuring a continuous oil film for sustaining the thrust load. The short distance between adjacent oil grooves, in the direction of rotation, limits the oil-temperature rise. The oil grooves are of ample cross-section, so that the cold oil fed to a groove
can take away the warm oil carried over as a film from the contiguous section of thrust surface. The oil-temperature rise is about 5° to 10° C. (say 10° to 20° F.) maximum; usually it is less. The thrust loading is about 15 kg./sq. cm. (say 200 lb./sq. in.), when calculated on the basis of brake thrust multiplied by an experimental coefficient.

In the main bearings there is a circumferential groove in the upper half; a pocket is provided at each side of the bush and there are no longitudinal oil grooves. Lubricating oil from the main system is fed by external pipe to the main-bearing housing, and thus, by way of the circumferential groove in the upper bush and the shaft radial holes, finds its way into the longitudinal oil hole of the crankshaft and so to the crankpin, flowing outward through two radial holes to the bottom-end bearing. Passing through the side pockets in the bottom-end bushes, the oil flows up a central hole in the connecting-rod to the top-end bushes, whence it finds its way out into the crankcase.

**Stress System**

For the eccentric drive, each rod is supplied with oil at a point on the crosshead. This oil passes to the top-end bearing and then continues downward through the eccentric rod to the strap. As exhaust and main pistons have equal cross-sectional areas, and as both systems of gear, comprising piston and connecting-rods for main pistons, side and eccentric rods for exhaust pistons, lead back to the crankshaft, it follows that all vertical forces are balanced within the engine running-gear; this leaves only the forces caused by the differential obliquity effect, the torque-reaction, the inertia forces and the sea-way racking forces to be absorbed by the engine structure. Accordingly, frames and bedplates of engines having full-sized exhaust pistons are lighter than those of engines with cylinder covers. If the top or bottom side of any cylinder is out of commission the stress system is undisturbed. For a six-crank engine the residual primary couple at 120 r.p.m. is 18.4 tonnes-metres (59.3 tons-ft.); for an eight-crank engine, also at 120
r.p.m., the residual primary couple is 5.8 tonnes-metres (18.7 tons-ft.). The secondary couples and all forces are balanced.

Fig. 61 shows the main bearing loads of a six-cylinder engine at 120 r.p.m.

![FIG. 61. – MAIN BEARING LOADS, D.A. 2C. ENGINE](image)

**Scavenging**

One of the characteristics of this engine is the method of scavenging. Scavenging air enters the cylinder through ports arranged at mid-height and scour the cylinder longitudinally, the exhaust gases leaving through ports respectively at the top and bottom extremities of the cylinder liner. The scavenge ports are controlled by the main piston, the exhaust ports by the exhaust pistons. Effective scavenging has its reflection in the mean pressure which can safely be carried, and therefore in the horsepower which can be delivered by the engine. Fig. 62 shows sections through a typical cylinder.

**Cylinder**

The cylinders are carried, at mid-height, on the substantial entablature which serves as a scavenge belt, the cylinders freely expanding above and below this belt. Each cylinder is lubricated at two diametrically opposite points near the end of the top piston path and at two similar points for the bottom piston, at right angles to the first-mentioned pair. For each exhaust piston there are three lubricating points. All points are located about the middle of the group of piston-rings served, when these are at the end of their stroke. The cylinders are fitted with mechanical lubricators.
The cylinder liners are turned all over, the material being vanadium cast iron. The jackets are fresh-water cooled. Cylinder-liner cooling must be as effective as possible, to assist in maintaining the lubricating-oil film. The main and exhaust pistons should be as hot as the materials will permit, to promote good combustion. Oil cooling is very satisfactory for this purpose. The rise in oil temperature between piston inlet and outlet is not more than about 20° C. (say 35° F.) at full power.

FIG. 62. – CYLINDER SECTIONS, DOUBLE-ACTING ENGINE
Pistons

The form and proportions of the pistons are well adapted for withstanding temperature variations. The top and bottom crowns are chrome-molybdenum cast steel, the body pieces special cast iron. Sandwiched between the latter and strongly bolted to them is the flanged end of the piston-rod. The crowns are screwed into the body pieces, the joint faces being oil-tight, metal-to-metal. There are six piston-rings and one scraper ring at each end of the piston. The piston-rings are arranged in the cast-iron body pieces at maximum distance from the hot ends. The piston-body diameter is 1 mm. (0.039 in.) less than the cylinder bore, tapering to 3.5 mm. (0.138 in.) less at the piston crowns. A baffle in the lower piston deflects the entering stream of cooling oil on to the crown. Similarly, in the upper piston a funnel ensures that the returning oil makes contact with the upper crown. The cast-iron sleeve shields the piston-rod from the combustion gases and provides a very suitable rubbing surface for the stuffing-box rings.

The exhaust pistons are of similar construction to the main pistons, a chrome-molybdenum cast-steel crown being screwed, with eight threads per inch, into a body-piece of special cast iron. There are six piston-rings to each exhaust piston. The corresponding scraper ring is arranged at the end of the cylinder liner. The exhaust-piston body is cylindrical, 1.5 mm. (0.059 in.) less in diameter than the cylinder bore; at the innermost piston-ring it begins to taper towards the end of the crown, where it is 3.5 mm. (0.138 in.) less. The exhaust pistons are tap-bolted to the cast-steel yokes; gauges are supplied for registering their positions relative to crankshaft top dead-centres.

Both the main and exhaust pistons are oil-cooled through a telescopic-pipe system. The crosshead shoe is lubricated from the cooling-oil pipe rising to the main piston.

Fuel-surcharging Pump

The fuel-surcharging pump is located near the manoeuvring platform, and is lever-driven from the engine. The fuel oil flows from the service tanks to the surcharging pump, and is delivered through effective filters to the fuel pumps, thence to the cylinder fuel valves. A starting valve of automatic type is arranged at each cylinder end; it is operated by compressed air, the supply of which is controlled by an air distributor rotating at engine revolutions. A safety valve of standard design is mounted on the top and bottom end of each cylinder.

Scavenge Blowers

The scavenge blowers – of which there are two per engine – are of rotary type. The fuel-pump camshaft and the scavenge blowers are chain-driven from the crankshaft. The drive for each half of the blower comprises a duplex chain, 2.5-in. pitch. For the fuel-pump camshaft there is a single chain, 4-in. pitch. On each chain-drive there is an adjustable, spring-loaded jockey wheel, which may be arranged either on the slack side or on the driving side.
of the chain. Only in short chains is it important to have the jockey wheel on the slack side.

**Overhaul**

The overhaul of the engine is simple. The separate end of the lower cylinder liner is disconnected and dropped, the rings of the lower exhaust piston thereby being exposed. To facilitate the upward withdrawal of the main, also the bottom exhaust, piston the cylinder bore may be stepped to three diameters, each differing from the next by 1 mm. on the bore. The crankcase doors are completely unobstructed for their full height.

In general terms, speed, ease and cost of overhauling any kind of engine depend – in the last analysis – upon the mechanical gear provided. A large and powerful engine, having excellent power-driven gear, is easier to overhaul than is a small engine provided with nothing better than hand-blocks.

![FIG. 63. – DOUBLE-ACTING, TWO-STROKE ECCENTRIC-TYPE, OPPOSED-PISTON ENGINE](image-url)
Rating

The engine illustrated in Figs. 54 and 55, and shown on the test bed in Fig. 63, develops 1,000 s.h.p. per cylinder, at 115 to 120 r.p.m. The cylinder is 550 mm. (21.65 in.) bore, 1,200 mm. (47.25 in.) main stroke and 400 mm. (15.75 in.) exhaust stroke. The piston mean speed at 120 r.p.m. is 4.8 metres/sec. (945 ft./min.); the average of the top and bottom mean indicated pressures is 6.5 kg./sq. cm. (92.5 lb./sq. in.). A mean indicated pressure of 7.0 kg./sq. cm. (100 lb./sq. in.) and a mean piston speed of 5.2 metres/sec. (1,025 ft./min.) can continuously be sustained at sea, but 6.5 kg./sq. cm. is more economical in terms of maintenance. These figures are on the basis of a twenty-four hours per day rating.

<table>
<thead>
<tr>
<th>Table III</th>
</tr>
</thead>
<tbody>
<tr>
<td>Engine size: 8.550/1,200 + 400 D.A. 2C., Rated at 8,000 b.h.p. at 125 r.p.m.</td>
</tr>
<tr>
<td>B.h.p.</td>
</tr>
<tr>
<td>M.i.p. (top), kg./sq. cm.</td>
</tr>
<tr>
<td>lb./sq. in.</td>
</tr>
<tr>
<td>M.i.p. (bottom), kg./sq. cm.</td>
</tr>
<tr>
<td>lb./sq. in.</td>
</tr>
<tr>
<td>Exhaust temperature (top) °C.</td>
</tr>
<tr>
<td>°F.</td>
</tr>
<tr>
<td>Exhaust temperature (bottom) °C.</td>
</tr>
<tr>
<td>°F.</td>
</tr>
<tr>
<td>Fuel consumption i.h.p. hour, gm.</td>
</tr>
<tr>
<td>Fuel consumption i.h.p. hour, lb.</td>
</tr>
<tr>
<td>Fuel consumption b.h.p. hour, gm.</td>
</tr>
<tr>
<td>Fuel consumption b.h.p. hour, lb.</td>
</tr>
<tr>
<td>Scavenge-air pressure, mm. Hg</td>
</tr>
<tr>
<td>in. Hg</td>
</tr>
<tr>
<td>Mechanical efficiency, per cent.</td>
</tr>
<tr>
<td>R.p.m.</td>
</tr>
</tbody>
</table>

At 120 r.p.m. the upward acceleration force for the main running gear is 12.1 times the weight. The top compression pressure is normally 33 to 35 kg./sq. cm. (470 to 500 lb./sq. in.), and the bottom compression pressure, 31 to 33 kg./sq. cm. (440 to 470 lb./sq. in.). The initial pressure is not more than 49 kg./sq. cm. (697 lb./sq. in.), and on this figure the crankshaft calculations are based. In the bottom cylinder, the piston-rod sleeve causes a cross-sectional loss of 16 per cent.

The cylinder constant is 1.554 for metric units.

For the computation of this constant, the total stroke is taken as the arithmetical sum of the main and exhaust piston strokes. In strict accuracy
the vector sum should be taken with the eccentric at the angle of lead, but the limitations of accuracy of recording make this difference negligible.

Table III shows test-bed figures for an eight-cylinder engine, rated at 8,000 b.h.p. at 125 r.p.m. Fig. 64 shows another set of results, for a five-cylinder engine rated at 5,000 b.h.p. Three cylinder sizes suffice for powers ranging from 5,000 b.h.p. single-screw to 30,000 s.h.p. twin-screw.

![Test-Bed Results, Double-Acting Engine](image)

FIG. 64. – TEST-BED RESULTS, DOUBLE-ACTING ENGINE

The engine is fitted with an emergency governor of the twin-pawled acceleration type, the cutting-off pawl being capable of adjustment in bad weather. The engine is capable of rapid manoeuvring; complete reversal from full-speed ahead on the engine to full-speed astern requires only a few seconds. The engine can run continuously at 30 r.p.m. The fuel-injection system, manoeuvring gear, etc., are of the firm's standard design.
**Fuel Pumps and Valves**

Fig. 103 shows a typical fuel pump. It is operated by a cam of such shape that a “snappy” cut-off is ensured. The same cam is used for ahead and astern running, there being a lost-motion clutch on the camshaft. The fuel quantity per stroke is varied by rotating the plunger somewhat, thus altering the effective pump stroke and spilling the surplus fuel from the helical edge of the plunger groove. The fuel-regulating gear is operated by hand and by the emergency governor. The plunger of any fuel pump can be cut-out while the engine is running. The
fuel-pump bodies are made of mild steel, the liners of special quality cast iron, the plungers of case-hardened carbon steel. Recently, plungers and sleeves have been made of nitralloy steel. The cam is a case-hardened carbon steel, the roller and spring seat 3 per cent nickel case-hardening steel. For a single-acting two-stroke engine conservatively rated at 1,000 to 1,100 s.h.p. per cylinder, the fuel pump is 42 mm. (1.65 in.) bore, with a full stroke of 65 mm. (2.56 in.). The designed working pressure is 425 kg./sq. cm. (say 6,000 lb./sq. in.).

The corresponding fuel valve is illustrated in Fig. 104. There are normally two fuel valves per combustion chamber. The oil sprays are partly directed towards the piston, but also partly against and partly with the air swirl. The valves are automatic in action; they open considerably below injection pressure, which is 280 to 420 kg./sq. cm. (say 4,000 to 6,000 lb./sq. in.). The nozzle-holes for each fuel valve of a single-acting two-stroke engine, developing 1,000 to 1,100 s.h.p. per cylinder, may be four in number, and 0.041 in. in diameter. The nozzle-hole length is three diameters. The oil speed through the nozzle-holes is about 190 to 210 metres/sec. (say 600 to

![FIG. 106. – FUEL PUMP, DOUBLE-ACTING ENGINE](image-url)
700 ft./sec.). The valve body is made of mild steel, the sleeve of nitralloy steel, the spindle and nozzle end of non-shrink steel, the spring of silicon-manganese steel. All springs are ground and spherodized; they are not to be painted. The introduction into the fuel line of duplicate felt filters, by excluding hard carbon particles from the fuel valves, is of advantage to cylinder-liner life.

Fig. 106 shows typical sections of a pair of fuel pumps for a double-acting, two-stroke engine. One pair serves a cylinder, i.e., one pump delivers to the two top fuel valves, the other to the two bottom valves. The cams are separately adjustable for injection timing. Here, also, as there is a lost-motion clutch on the camshaft, a single cam serves both for ahead and astern running. A spring shock-absorber, for levelling-out pulsations in the surplus fuel spill, is fitted opposite to the suction inlet. The pump-suction side is automatically vented against air-lock.

Each fuel pump discharges to a distribution block, thence to the fuel valves; see Fig. 105. The filter for each valve is of the gauze design. There is no non-return valve between fuel pump and fuel valve. The fuel is injected into the cylinder at high pressure at the appropriate moment. Immediately injection has taken place, the fuel line is relieved of pressure until the next injection. A sharp cut-off of the fuel – to avoid after-burning – is important.

There are many fuel valves in service of the type of Fig. 105. For an engine 620 mm. (24.41 in.) bore, 1,400 mm (55.12 in.) stroke, there are three nozzle-holes per fuel valve, 0.037 in. bore (top valves) and 0.0292 in. bore (bottom valves). The nozzle-hole length is about 3 diameters. Fig. 107 shows the fuel valve for a four-stroke, single-
acting engine. The nozzle holes are six in number and 0.047 in. in diameter. There is one valve per cylinder.

The positiveness of valve opening and closing precludes dribbling and carbon formation on the nozzle-end. The nozzle is cooled, either by circulating incoming fuel oil around it – the nozzle being provided with fins as in Figs. 104 and 105 – or by circulating it with fuel oil from an independent circuit comprising pump and tank; see Fig. 108 for diagrammatic scheme. The latter arrangement allows lower grades of fuel to be used.

In some of the latest single-acting two-stroke engines, the fuel-injection system is operated by the direct pressure of the compressed-air charge in the cylinder. The camshaft-driven fuel pumps are thereby eliminated, but there is a metering pump. For this purpose the type of fuel pump used on the old four-stroke blast-injection engines is utilized.

**Fuel-valve Faults**

The fuel valve is timed and controlled by the injection pump; it is the final delivery valve for the pump, in engines which have no intermediate non-return valves. With this arrangement the compression pressure can pass from a cylinder through the nozzle-holes and force the fuel past a leaky valve. The injection system of that cylinder will then be filled with air or gas and the cylinder will cease to fire. It is therefore necessary, as soon as detected, to open the fuel-valve priming valve, otherwise the compressed air from the affected cylinder will blow through the pump and enter the suction line. Other pumps may then become air-locked and the engine be brought to a standstill.

In small engines a non-return valve is fitted between injection pump and fuel valve to prevent involuntary stoppages. In large two-stroke engines there is no non-return valve, as danger of the engine stopping because of a leaking fuel valve is remote – with the larger volume of oil present. It is also highly important in such engines that a faulty valve should be discovered immediately. This is not always possible when non-return valves are fitted, as they may prevent the gases which have entered defective fuel valves from reaching the injection pumps to interrupt delivery. Thus there may be no outward sign that something is wrong. The exhaust temperature may show little or no change, because reduction in power is accompanied by poor combustion; fuel injection is generally late. Neither firing height on the indicator diagram nor exhaust-temperature changes can be depended upon to reveal a defective fuel valve if non-return valves are fitted in the system. The exhaust will be discoloured, however. The faulty unit can be found by holding a piece of stretched white cloth a few inches above the exhaust test cock of each cylinder in turn.

When an engine which has a defective fuel valve is stopped, fuel may enter the cylinder through the leak, either by gravity or by hand pumping.
The fitting of non-return valves in fuel lines has become standard practice for small engines; difficulties from bad combustion, when running with a leaky fuel valve, are not there as serious as with large engines.

Atomization occurs after the fuel has been forced through the fuel-valve nozzle-holes, and while it is penetrating the compressed air in the combustion space. The oil breaks up into a fine mist suited to instantaneous ignition and complete combustion. Restrictions in the fuel path between injection pump and nozzle end are undesirable. Chattering of the valve spindle on its seat is disadvantageous. With some valves, chattering may be produced during hand-pump testing; this, however, simply shows that the valve spindle is operating freely and that the seat is tight.

The testing of a fuel valve by hand-pump does not reproduce the movement of the fuel-valve spindle as it actually occurs during injection in the engine; e.g., the volume of oil is insufficient to provide the correct lift and to sustain it, and the valve will probably be no more than "cracked" open, the fuel being squeezed out between the two faces. Pressure-drop before the oil is forced through the nozzle-holes is therefore likely, and it might seem that the valve is spraying poorly, when actually it means that a higher rate of oil supply is required.

The closing period of a fuel valve is of first importance. If a valve spindle operates sluggishly, slow closing causes reduced penetration and dripping at the end of injection. After-burning, with its ill effects, will then prevail. To make the closing snappy, where non-return valves are fitted these are so designed that, as they reach their seats, they are displaced into the valve-seat cages, to cause a pressure reduction in the oil trapped in the fuel valves.

**Fuel Valve Pressure-testing**

The sequence of operations when pressure-testing a fuel valve is as follows: (i) the lift stop-screw is released a couple of turns; (ii) while using the hand-pump, the spring adjusting-screw is set and locked at the required opening pressure, this pressure being lower than that of normal injection because the fuel pressure, when the engine is running, will rise higher as the fuel speed increases through the nozzle holes; (iii) a pressure of about 200 lb./sq. in. below normal is applied and maintained and, if the nozzle tip remains perfectly dry for about ten to twenty seconds, the valve-seat is tight, a slight moisture at the nozzle, after this time, being without significance; (iv) the lift stop-screw is adjusted for correct height of opening; (v) half a dozen pump strokes are made in quick succession, and if only one drop of fuel gathers and falls from the nozzle tip the spindle is operating freely enough, but if several drops appear, the spindle is too sluggish and the valve must be taken adrift for polishing of spindle and sleeve – a sheet of cardboard or a bright metal plate being placed about a foot below the fuel valve, for observing the falling drops; (vi) if the fuel valve operates correctly the lift stop-screw is firmly screwed down to prevent the valve from opening, while the test pressure is raised to 2.5 times the normal opening pressure to ensure that the nozzle-cap joint is tight and that no undetected leaks will arise when the valve is in the
engine; it is of no importance if drops of fuel leak past the spindle from the drainage hole above the sleeve, but a continuous stream of oil will show the valve spindle to be too slack in the housing; (vii) the lift stop-screw is re-set and the opening pressure checked. Tests (v) and (iii) are repeated, in that sequence.

When a valve has been stored as a spare it should be re-tested before use, in accordance with (v) above.

**Fuel-valve Assembling**

To assemble a fuel valve when no special clamping device is available, a lathe or drilling-machine is much to be preferred to a vice. The valve body is bolted firmly to the tool-rest or lathe bed, with nozzle-tip in line with, and facing, the back centre. If bolted to a drilling-machine table the nozzle-tip should face the boring bar. A short tube, say an inch or two long – of outside diameter slightly smaller than the bore of the nozzle cap end – is placed to protect the nozzle tip. By means of the lathe back centre – or drilling-machine boring bar – the tube is now forced hard against the nozzle to hold it firmly against its face while the cap-nut is tightened by a spanner. This procedure ensures that the spray direction remains normal.

Carbon accumulations around the nozzle or its thread may make unscrewing of the cap difficult. Immersion in carbon tetrachloride, paraffin or other suitable liquid for twenty-four hours should soften the hard carbon crust. With the fuel valve held as described above, sharp hammer blows on a short, stout spanner should release the cap nut.

A sluggish spindle can often be eased, or leaking valve-seats ground together, without having to unscrew the cap. The spring, with its adjusting screw, is then removed and the spindle ground in place by tube spanner. The carbon from the nozzle holes, also the grinding paste, must be completely rinsed out of the valve bores, say by syringing with tetrachloride or clean paraffin.

Except for small engines – which have conical spindle ends and seats which are integral with the valve bodies – the fuel valves have flat faces, with separate nozzle-ends, as illustrated in Figs. 104, 105 and 107. In large engines experimentation with nozzle-holes is sometimes necessary and, if the nozzle-pieces were not separate, changes would mean discarding complete valves.

**Manoeuvring and Reversing Gear for Double-acting Two-stroke Engines**

The starting and reversing gear for double-acting two-stroke engines is diagrammatically shown in Fig. 109.

Fuel cam 1 is keyed to camshaft 2, on which there are two clutch toes 3 – opposite each other – formed solid with the camshaft. Chainwheel 4, loose on the camshaft, carries a lost-motion clutch ring 5, with two toes 6.
These toes are of such size that, when one side engages camshaft toe 3, as shown, fuel cam 1 is in phase with the crankshaft for ahead running; when chainwheel 4 and clutch ring 5 are rotated until toes 6 engage the opposite side of toes 3, the fuel cam is in phase for astern running. To prevent chatter, the lost-motion coupling 5 has bolted to it a friction plate arrangement. This comprises casing 7, which carries weight levers 8 and alternate friction plates 9 – having projections 10 engaging slots in 7, also plates 11 – which have projections 12 engaging in slots 13 in 2. The movement of 8 is limited by studs 14, also by springs inside 8, so adjusted that no force is exerted upon the clutch plates when the engine is turned slowly. This allows the
chainwheel to move freely on the camshaft, to take up the lost-motion when reversing. As the engine gathers speed, the centrifugal force of 8 overcomes the spring on 14 and exerts pressure on adjusting pins 15, which is transmitted through pins 16 to the clutch plates. The lost-motion rotation of 4 carries with it a nut 18 on the chainwheel sleeve. Rotation of 18 slides the threaded sleeve 17 along the splined camshaft, thereby moving 20, 21 and sleeve 22 – which is free on shaft 23 – and operating fuel-locking cam 19.

The air cylinder, acting through the reverse retaining gear and 24, 23, 25, 26, operates the blower change valves, held firmly against their faces by spring rods, not shown – and the retaining gear provides the reaction. A slot in plate 27 moves shaft 28, rotating at engine speed, which carries ahead and astern distributor cams. By reason of differential pistons on the distributor valves, these make contact with the cams only so long as starting air is on the distributor, the springs holding off the valves at other times. The reversing air and brake-cylinder piston-rod moves locking bar 29, which has two slots so placed that, when full-up or full-down, one is opposite to cam 30, which is operated by starting lever. This prevents starting air being put on the engine before the air-cylinder piston has completed its stroke. The movement of the starting lever which operates cam 30 also operates pin 31; this locks the reversing handle, except when the starting lever is at stop.

Fig. 109 shows the starting lever at “stop”, engine in ahead gear and fuel-locking cam 19 as at (a). The starting-air stop valve is closed. In opening this valve, before pressure can reach the automatic valve by pipe 43, the air passes through 32 to the pilot valve and 33 to the automatic valve top chamber, keeping it closed. Starting air cannot, therefore, reach 34 or 35.

Reversing

To reverse the engine: the starting lever is brought to “stop”, the handle on the reversing air-control valve is moved to “astern” and it is held there. This supplies air to the air-cylinder bottom. When the pointer comes to “astern” on the index plate, the reversing handle is moved to the vertical or neutral position; otherwise when attempting to bring the starting lever to “start”, locking pin 31 will come against the reversing-handle boss instead of entering the hole. The astern cam is now over the air-distributor valve, the blower change valves are reversed, and key 36 on lever 25, acting on lever 21, has extended spring rod 20 somewhat, bringing cam 19 into position shown at (b). The space between regulating lever 37 and cam 19 permits the starting lever being moved to “start”, cam 30 travelling into slot in bar 29 and pin 31 into hole in reversing handle. Draw-bar 38 lifts the air-pilot-valve spindle, releasing the pressure from the automatic valve top chamber, which automatic valve thereupon opens and starting air enters 34 and 35, forcing the air-distributor valves against the cams. The engine crankshaft begins to move astern; chainwheel 4 with its clutch gear, all driven from the crankshaft, takes up the lost motion – about one-third of a revolution – until clutch toes 3 and 6 come into contact for astern running. During this time 18 moves 17 towards 1, relieving the tension in 20 and moving 19 to position shown at (c), releasing 37 and so allowing the starting lever to be moved past “start” to
“fuel”. As soon as the starting lever is moved past “start”, pin 40 engages roller 41 and trips the air pilot valve, thus closing the automatic valve and allowing the air in 34 and 35 to escape. Valve 42 is closed only for testing starting valves.

**Timing of Cylinder Lubricators**

Some engine builders make no attempt to time the cylinder lubricators, but the difference between genuine timing – if this be possible – and indefinite injection is reflected in the rate of cylinder-liner wear, in the carbonizing of ports and in the incidence of scavage fires. The correct instant for injection is when the piston is moving very slowly relative to the angular movement of the crank. This occurs when the piston is approaching and when it is leaving the top and bottom dead-centres. Near the top dead-point the gas pressure is too great for oil injections. Lower down, also, the gas pressure behind the piston-rings can be sufficiently high to prevent the discharge of lubricant by the plunger. There remains the lower end of the stroke, and this is the most practicable region for the purpose.

In four-stroke, single-acting, crosshead engines a timing effect can be obtained. The point to which the crank is set when the lubricator plunger has just reached the end of its pumping stroke is 30° before bottom dead-centre. Experiment shows that, with such an engine running normally at 100 to 110 r.p.m., there is a time-lag between plunger movement and oil injection of approximately 20° of crank angle; *i.e.*, injection has been completed by the time the piston is about 10° of crank angle from the bottom dead-point.

In experiments on lubricating-oil injection settings at 30° after bottom dead-centre, under conditions closely simulating those in a four-stroke trunk engine running at 135 r.p.m., there was at best a continuous dribble, with a minor periodic pulse superimposed upon it.

There is always a non-return valve on the lubricator discharge pipe. This should be placed as near the cylinder as practicable, to assist the timed impulse of oil. There may be carbonization difficulties if the non-return valve is mounted on the cylinder.

The cylinder lubricator, as ordinarily made, has a very short effective stroke and, in ships which travel from cold to hot and hot to cold climates, it is in continuous need of adjustment, because of the oil-viscosity variations which are inseparable from changing ambient temperature. The incorporation of a small, thermostatically controlled, heating element into the lubricator overcomes the need for viscosity adjustment. A speed-reducing gear of say one-third, by which the effective plunger stroke is increased and the number of strokes per minute reduced, can be helpful. The injection lag is, however, thereby increased.

Practice varies in the number of lubricating points per cylinder. Two points are satisfactory for bores up to 550 mm. (21.65 in.), and four points for cylinders 750 mm. (29.53 in.) diameter.
Indicating Gear

Fig. 114 shows the indicating-gear arrangement for a double-acting engine, and Fig. 115 shows the detail. There are two circular cams, one for the top end of the cylinder and one for the bottom end. The roller is changed from one cam to the other by revolving the locating link through 90° and pushing it over from one side of the lever to the other; pushing the roller along its pin, from one cam to the other; and dropping the locating link into the place previously occupied by the roller. The motion of the cam-roller lever is transmitted through links to a vertical rod, the top and bottom extremities of which are attached to cords which drive the respective indicator drums. The spring underneath the lever ensures contact between roller and cam at all times. Similarly, the long spring on the vertical rod ensures its return. The cams are attached by spigoted construction to a carrier, and are each held to it by four tap-bolts arranged in adjustable slots; the carrier grips the camshaft and is secured by a saddle-key; in addition, there are two radial set-screws, thimble-pointed. Cams and rollers are of forged mild steel. The difference in result between the simple circular cam and a truly shaped cam is negligible.

In a single-acting engine wherein the ratio of connecting-rod to crank is 4/1 and eccentric rod to eccentric is 12/1, the cam is 437 mm. (17.20 in.) diameter, the roller 100 mm. (3.94 in.) diameter and the eccentricity of cam 45.5 mm. (1.79 in.). In a double-acting engine in which the ratio of connecting-rod to crank is 4/1 and eccentric rod to eccentric 11/1, the cam is 412 mm. (16.22 in.) diameter, the roller 80 mm. (3.15 in.) diameter and the
eccentricity of cam 41 mm. (1.61 in.). The author prefers to underestimate rather than overestimate mechanical efficiencies.

**Piston-rod Scraper Box**

Fig. 116 shows a typical crankcase-top piston-rod scraper box. The device essentially consists of three cast-iron segmental scraper rings, held against the piston-rod by steel garter-springs. Drainage is most important; accordingly, in the grooves behind the rings, drain holes of ample size are provided. This construction ensures a moist piston-rod, whilst also preventing the lubricating oil from being taken up into the stuffing-box – if a double-acter – there to cause ring-sticking. The vertical distance between the scraper box and the top of the crosshead is important: if it is too small, aerial pumping action may force lubricating oil through the scraper box.

An earlier and fully effective design of piston-rod scraper box – of which there are many in service – is shown in Fig. 117.

**Piston-rod Stuffing-boxes**

The current standard design of piston-rod stuffing-box for double-acting, two-stroke engines is shown at A in Fig. 118. It consists of eight sets of compound sealing rings, carried in distance pieces. The inner or sealing rings are plain, special-quality, cast-iron rings having lap joints. They are initially machined to such an outside diameter that the tips of the half-laps are in line. The rings are then hammer-rolled externally to the ultimate diameter and ground on all faces to the final dimensions. The inner top edge of the two top rings, and the inner bottom edge of the two bottom rings, are rounded to a radius of 1 mm. The intermediate rings have square edges. Behind the sealing rings are plain, special-quality, cast-iron spring-rings. These spring-rings are also ground on all faces. The combination of rings makes an effective seal. Spring steel may, perhaps, be preferable to cast iron for the two top spring-rings; but cast iron is very satisfactory. The perlitic cast-
iron distance-pieces which carry the piston-rings are ground on their faces. If, during the grinding operation, they are held by magnetic chuck, the magnets should be numerous and not too strong; otherwise there may be distortion of the faces. It is imperative that, when the distance-pieces leave the machine, they should be truly plane. For this reason a jig is to be preferred to a magnetic table.

As shown in Fig. 118 the distance-pieces are held together by four sunk straps of mild steel, with tee-end and countersunk screw at the top, and screwed end with nut at the bottom. The assembled box is held to the bottom exhaust-piston by ten 0.75-in. studs, the top face of the box being spigoted and ground; see Fig. 117. The use of positioning pins in the distance-pieces is avoided, as these are liable to be burned through.

From its nature, the piston-rod stuffing-box is a detail upon which users tend to imprint their individuality. Accordingly, a number of variants and sub-variants are current. Thus, instead of one plain lap-jointed inner ring, there can be two rings each split into three segments, the joints in the upper ring
being non-coincidental with the joints in the lower ring. This is shown at C, Fig. 118. At D the inner rings are solid, of lap-jointed type, but each in two segments; the rings behind them are spring steel. The upper four inner rings are made of special cast iron, the lower four of lead bronze.

Another current type of stuffing-box is that shown at B. This is a solid block, with two loose distance-pieces at the bottom. The block houses four rings of the twin type; these rings do not provide a complete seal, as the gas can blow around the circumferential annulus and through the openings between adjacent rings. The three lowest rings, however, are of double seal type, with joints in adjacent rings arranged to be non-coincidental. All rings are of special cast iron. At E another variant is shown, in which the uppermost ring is a solid ring and the remaining seven are of double-seal type.

Sometimes, stuffing-boxes are made with six grooves, each of the two topmost grooves being occupied by a pair of plain Ramsbottom rings, one superimposed upon the other. The amount of inward springiness of these rings is small. In each of the four remaining grooves there are also two plain rings; each of these rings is in three segments, with a steel garter-spring to hold them on to the piston-rod.

Fig. 117 is an earlier design, of which there are many in current use. There are six twin-rings, and a bottom ring of the compound sealing type, i.e., a ring of square section inside a gnomon.

**Piston-rings**

Liner surface should be considered in conjunction with piston-ring problems. It is often believed – quite wrongly – that the best finish is that imparted by honing or grinding. If a liner is turned to a reasonably good finish, the innumerable tiny hollows in the surface may serve as lubricating-oil receptacles tending to reinforce the oil film which is so important. An initially smooth surface is not conducive to long liner life. The limit of permissible wear on liners, before replacement, is about 1 mm./100 mm. of diameter.

It is easy to state that the materials, design and manufacture of piston-rings should be such that there is neither blow-past nor sticking, that the friction and wear are the least possible and so on; it is much less easy to satisfy these desiderata. Non-marine men tend to criticize the number of piston-rings which are fitted to propelling engines. They frequently suggest that the number could, with advantage, be halved. Experience, however, has demonstrated that the customary numbers of rings yield the most satisfactory results.

It is important that the maximum amount of heat from the combustion space should be conveyed away before the top piston-ring is reached, otherwise this ring is likely to stick from heat expansion. For this reason the first groove should be as far away as practicable from the piston crown. The first and second rings should always be made to float more easily in their grooves than those farther down. Some superintendents require a specific hardness for
ring materials, but conflicting results can be obtained when rings of equal hardness are applied to different liners.

The edges of piston-ring working faces should be bevelled, or rounded, to assist in retaining the lubricating-oil film on the cylinder wall until the rings and liners are worn smooth. Thus, for the 620-mm. (24.41-in.) piston, a 2-mm. (0.08-in.) radius would be suitable.

![FIG. 119. – PISTON-RINGS](image)

Regarding the ends, some engineers favour overlapping ends as at (a) and (b), Fig. 119, to obtain a seal; others prefer the diagonal cut of (c). The ring at (d) is a compound sealing ring, as used in four-stroke engines.

End-butting of rings can be troublesome, especially in two-stroke engines, if the ring gap is too small. It is possible for end-butting measurably to lower the mechanical efficiency; the effect can sometimes be felt at the turning gear, if used when the engine is warm. An adequate amount of axial clearance is necessary for all exhaust piston-rings, also for the two top rings of main pistons. Adequacy may imply twice normal clearance. With proper fitting of piston-rings in their grooves, with minimum end gaps, and with a suitable amount of lubricating oil applied to liners at the right moment, scavenge fires are unlikely. In this connection the piston-ring joint design is important. Fig. 120 shows an overlapping end-joint, sufficiently robust to avoid fracture, which prevents the passage of gas. A generous amount of end-clearance, without blow-past, is obtainable with this design. The use of positioning pins for piston-rings is unsatisfactory. A much more robust alternative, obtainable at slight additional cost, is shown in Figs. 119 and 120. It minimizes vertical wear and enables the rings to mate perfectly with the liner. No cylinder completely retains its form cold and hot, and an unanchored ring is liable to rotate somewhat. The ring ends coincide with the broad bar at the scavenge ports.

A correctly made ring, when closed-in the pre-determined amount, should present a truly circular working surface. To reduce costs, makers’ processes are sometimes such that rings are only approximately circular; such rings should be avoided. Groove finish is important. It avails little if excellently
finished rings have to work in roughly finished grooves; blow-through with new rings is often attributable to this cause. To reduce likelihood of seizures during the running-in period of a new engine, lead-bronze rings fitted to machined grooves in the piston circumference are very useful, the face of the rings being about 0.1 mm. (0.004 in.) above the piston surface.

**Care of Scavenge Belts**

Cleanliness of scavenge belts is indispensable. Its assurance implies frequent and thorough attention to all places where oily carbon can accumulate. So long as lubricating oil can be drained from the scavenge belt there will be no fires, but if, due to blow-past, the lubricating oil becomes carbonized, then the walls around the scavenge ports will gradually become dry, and danger of fire will arise.

**Main-bolt Tightening**

All four-stroke, and some two-stroke, engines are braced vertically by long bolts. There is a nut on the bedplate bottom, one on the cylinder top and one on the framing top. The last-named is pulled hand-tight — after being sledged and slackened, to bring all faces together — and then rotated by tup the amount prescribed on the engine-adjustment sheet. The top nut is then pulled hand-tight and rotated the prescribed amount, which just lifts the intermediate nut clear. Thus, for a 500-mm. (19.69-in.) bore, 900-mm. (35.43-in.) stroke engine the intermediate nut is rotated 226° from hand-tight and the top nut 49°. The intermediate nut holds the engine parts together when the cylinder cover is dismantled; it also simplifies the tightening-up of the top nut. Tie-bolt nuts, unlike nuts on running gear, should not require periodical attention but only verification.
For double-acting two-stroke engines the tie bolts are in two pieces – the junction being at the middle nut. The latter is pulled hand-tight and the upper bolt is pulled into the middle nut; then all the main bolts are simultaneously tightened hydraulically by a system consisting of a hand-pump coupled to portable rams on the top nuts. The pump pressure ensures a definite and equal tensile stress in all bolts, usually about 650 kg./sq. cm. (9,250 lb./sq. in.). The top nuts are then run-down, by means of a toggle bar, until they are
hand-tight on the cylinder cover. The hydraulic pressure is released and later reapplied for a check reading; the portable rams are then taken away. The space under the middle nut is filled by a steel split-washer, which slides into place. Fig. 121 shows the arrangement; the screwed part of the bolt is made longer to take the hydraulic ram. In existing engines a tap-bolt can be screwed into the end of the tie bolt, holding the ram, as in Fig. 122. The ram diameter varies with the engine size, the hydraulic pressure being constant – usually about 211 kg./sq. cm. (3,000 lb./sq. in.) – except in special circumstances.

When a long tie-bolt is tightened, the bolt extends and the frame is compressed. The thread protrusion through the nut is the sum of bolt stretch and frame compression. The firing load reaction, on the cylinder cover, further stretches the bolt, but relaxes the framing. The tie-bolt tightening must ensure that, under piston load, the surfaces do not separate.

The present tendency is to extend the principle of hydraulic or pneumatic tightening for important nuts. Fig. 123 shows a method of hydraulically tightening or untightening crosshead nuts. A screw-jack can be substituted for the hydraulic cylinder, as in Fig. 124.