The Future Doxford Marine Oil Engine

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The paper will describe the ideas and objects in the development of the new Doxford engine—to reduce length and weight, to remedy the defects and difficulties which had been experienced with the earlier Doxford Opposed Piston Engine.

The features of the new engine are described and compared where expedient with the earlier engine—the crankshaft and its design features, the bedplate and main bearings, the connecting rods, piston rods, crossheads, piston heads, transverse beams, cylinder liners, and exhaust belts. The fuel injection and starting air systems are described and also the scavenging pump of the normally aspirated engine.

The turbocharging of the engine is also described and the arrangements adopted on the various multi-cylinder engines. The design features of the turbocharged engine are shown and finally a summary of the test results on the first engine is given.

INTRODUCTION

An attempt is made in this paper to describe the latest type of Doxford engine which it is hoped will form the basis of the production at the Pallion works for the next two decades. It is now over ten years since the author began to consider the type of engine which would be required to meet the services and competition of the future for the type of ship constructed by Doxford's and their various engine customers. It was considered that an engine for these future requirements should be of reduced length and weight relative to the previous engine which had been designed by Mr. K. O. Keller and Mr. W. H. Purdie in 1934, although there had been various modifications in the intervening years, e.g. the diaphragm engine was introduced in 1953. It was also considered that some of the difficulties encountered with the earlier engine could be eliminated by careful attention to re-design of the offending parts, so that maintenance would be reduced to a minimum. The Doxford engine has always been noted for its easy accessibility and it was hoped to retain this very important feature. With regard to the fuel consumption, it was considered that very little improvement could be made in this respect relative to the consumption of the earlier engine, since average figures on Diesel fuel of 0·345lb./b.h.p./hr. could be obtained and occasional engines gave a consumption of below 0·33lb./b.h.p./hr. With a normally aspirated engine, very little improvement could be made on these figures which were due to the high mechanical efficiency and good combustion of the Doxford engine. The high mechanical efficiency of the engine was primarily due to the good balance of the rotating parts and to the very efficient scavenging air pumps of the double acting, reciprocating type which were employed. These pumps had to deliver against a low scavenging air pressure and had a small clearance volume and large suction and delivery valves which were relatively lightly loaded. This low scavenging air pressure was, in turn, due to the through scavenge of the opposed piston engine and to the large ports of low resistance for both air inlet and exhaust which this type of engine permitted (see Fig. 1). The exceptionally good combustion was, in its turn, due to the good scavenging of the opposed piston engine, to the rotary swirl which was obtained from the tangential deflexion of the row of scavenge ports extending round the low end of the liner. The combustion was further assisted by the spherical form of the combustion chamber which is the most efficient form and has the least surface area for the volume of combustion space enclosed. In addition, the fuel injection took place through two fuel valves located on opposite sides of the periphery of the combustion space and the fuel sprays were adequately distributed through the rotating air.

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BALANCING AND TORSIONAL VIBRATION CHARACTERISTICS

The previous Doxford engine designed in 1934 had the ratio of the throws of the centre and side cranks such that true rotary balance was obtained and primary balance of the upper and lower reciprocating weights was obtained by adding weight to the lower piston in the form of a heavy skirt or a large diameter piston rod. The reason for this was because the weight of the upper piston, transverse beam, two side connecting rods and crossheads was greater than the weight of the corresponding reciprocating parts of the centre piston and rods. In the new engine these characteristics have been reversed (by a reduction in the stroke of the upper piston), and the ratio of the strokes is such as to give correct reciprocating balance of the primary forces, and rotary balance is achieved by adding weight to the main crankwebs opposite to the centre crankpin. On the prototype single cylinder engine (Figs. 2 and 3) built to prove the new features of this future engine, the scavenge pump was arranged above the transverse beam of the upper piston and the reduced stroke of the upper piston also suited the displacement of the scavenge air pump to give the correct amount of air for the normal aspiration of this type of engine. With these arrangements, each line of the new engine has correct rotary balance and also primary balance of the reciprocating parts so that only the secondary components of the reciprocating forces have to be considered in the multi-cylinder engines. Good balance has always been a feature of the Doxford opposed piston engine and this is the reason for the freedom from vibration and smooth running of the engine.

Torsional Vibration

The crankshaft of the new engine has been made as rigid as possible with large diameter short length bearings and this, together with the reduction of the weights of the rotating and reciprocating parts gives a high natural frequency of torsional vibration of the engine, in fact, double that of the 1934 engine having a corresponding

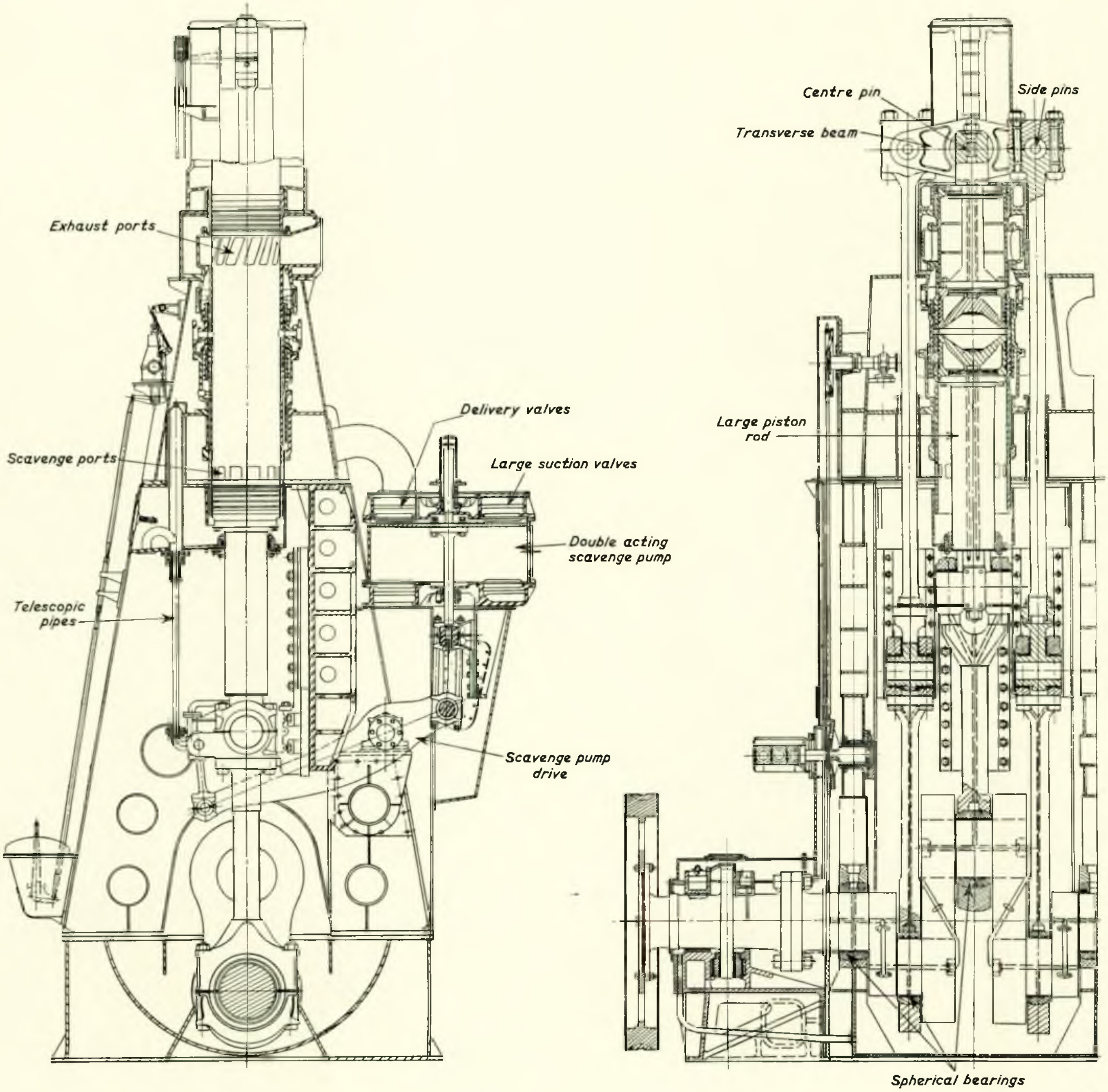


FIG. 1—*Sections through the diaphragm engine*

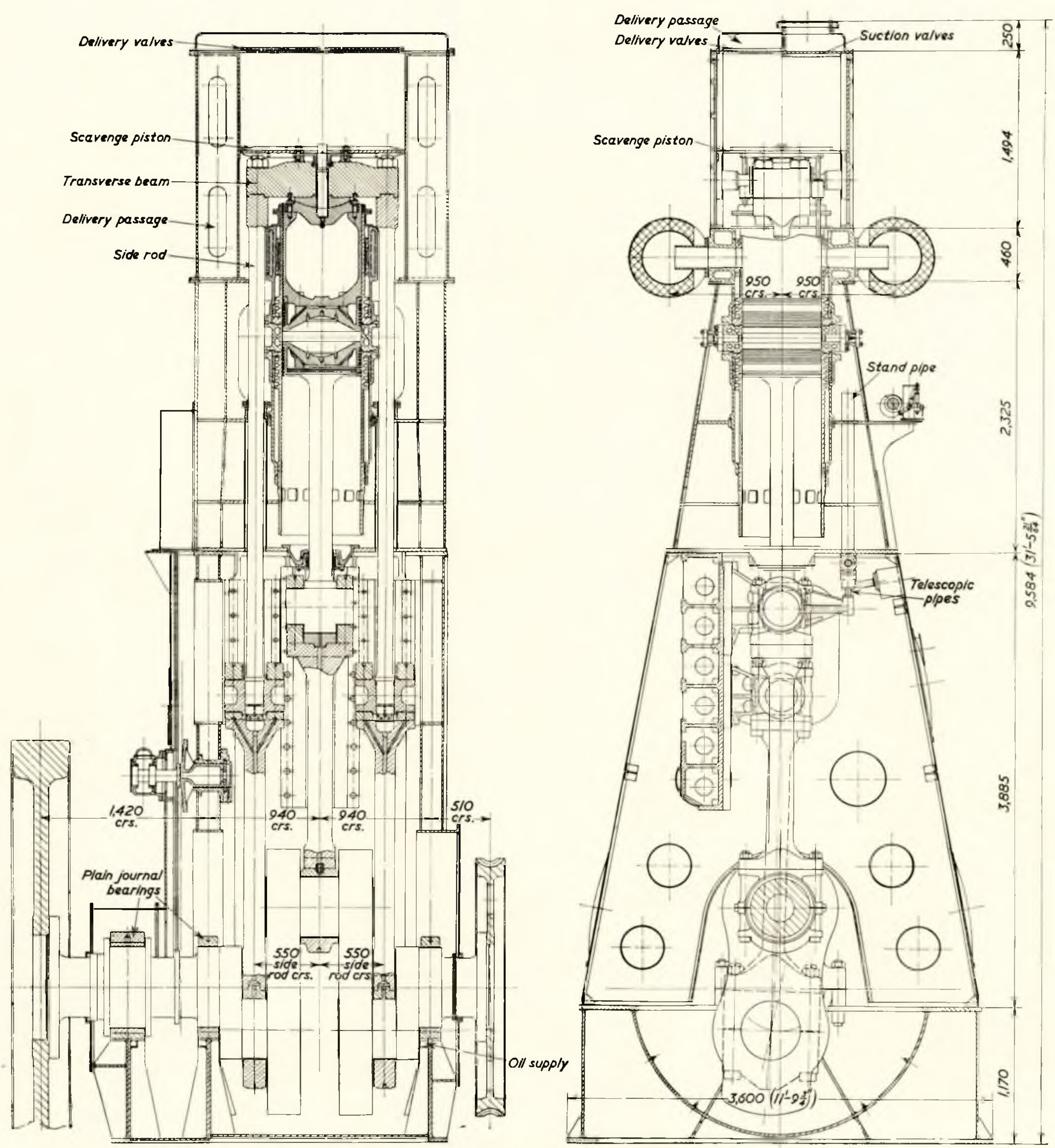


FIG. 2—*Sections through the single cylinder engine*

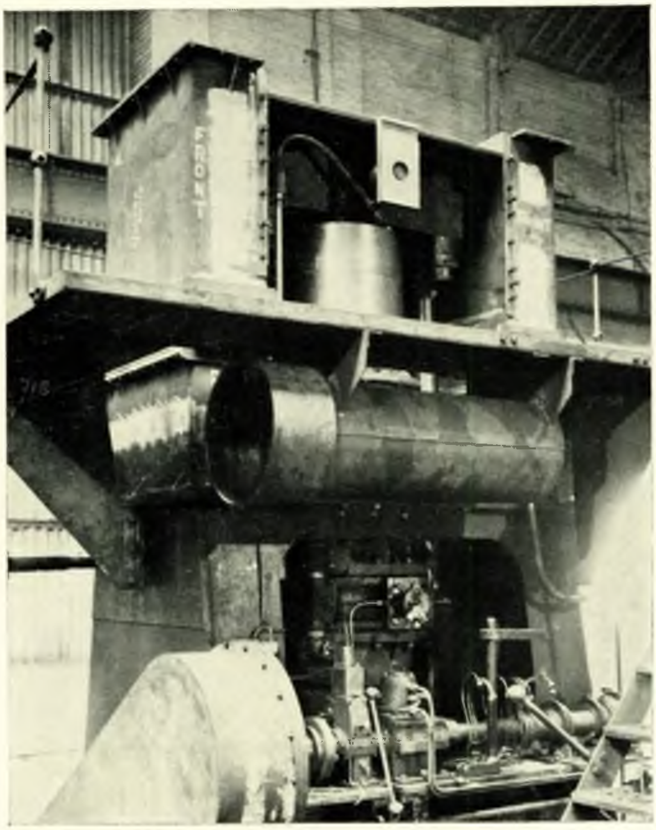


FIG. 3—*Single cylinder engine above the camshaft*

number of cylinders. Thus the vibration conditions become much easier and there should be no necessity to adopt detuning or damping mechanisms on installations of engines having up to five cylinders and it was hoped that even on the six cylinder engine the normal Doxford Bibby Detuner would only be required under special circumstances and would have a relatively easy duty to perform. The stresses in the crankshaft would be reduced by the large diameter of the pins and journals and the large overlap of these reduced the stresses in the side crankwebs.

SINGLE CYLINDER ENGINE

The prototype single cylinder engine which was built some five years ago had a cylinder bore of 670 mm. and a combined stroke of 2,100 mm., the stroke of the centre crank being 1,370 mm., and that of the side cranks 730 mm. This prototype engine is shown in Figs. 2 and 3 and the special feature will be described under the various headings for the multi-cylinder engine. One special feature of this engine was the scavenge air pump with its piston mounted on top of the transverse beam and to enable this to deliver the correct quantity of air for the engine, it was made of rectangular form. The reason for this was that its longitudinal dimensions had to surround the transverse beam (see Fig. 4) and a circular piston of this diameter would have been too big giving far too much air for the engine. The rectangular cylinder was built up of two side plates and two end trunks which conveyed the air from the delivery passage to the entablature. The suction and delivery valves were placed in the upper cover of the pump. Although this scavenge pump worked perfectly and gave no trouble there was a considerable scepticism on the part of many engineers who came to inspect the engine, regarding the feasibility of a rectangular scavenge pump and the effectiveness of the piston sealing arrangements, and therefore, for the multi-cylinder engines of the normally aspirated type, it was decided to change the scavenge pump drive and arrangement as will be described later.

It was decided that this future engine should be built in both normally aspirated and turbocharged forms with engines having 3, 4, 5 and 6 cylinders to give a range of powers from 3,000 to 10,800 b.h.p.

RATING OF THE ENGINE

The 1934 engine had been rated originally at 80lb./sq. in. mean indicated pressure, and this had been increased over the years to 88lb./sq. in. with normal aspiration. This engine had also been turbocharged in recent years to 115lb./sq. in. It was realized that the new engine would require to have a life of 20/25 years and that during this period both speeds and mean pressures would probably increase. The 1934 engine had increased in speed from 100 r.p.m. to 115 r.p.m. and it was thought that while, at the present time, a propeller speed of up to 120 r.p.m. would be considered a sufficiently high speed of rotation, during the next twenty years speeds would probably increase to 130/135 r.p.m., while still achieving a high efficiency of propulsion. Similarly, mean pressures would increase but particularly so with turbocharging, so that the new engine was designed for a rotary speed of 135 r.p.m. and a mean effective pressure of 135lb./sq. in. with the possibility of this rising to 150lb./sq. in. in the next decade. The maximum pressure which would result from such mean effective pressures would probably be in the region of 1,000lb./sq. in. The engine was therefore designed with these ratings and maximum pressure characteristics in view.

With regard to the normally aspirated engine it has been stated that the 1934 engine had been developed to a rating of nearly 90lb./sq. in. mean indicated pressure but during research work a larger quantity of air had been delivered to the engine by the simple expedient of putting a copper sealing ring around the periphery of the scavenge pump to improve the volumetric efficiency and on various endurance tests of 1,000 hours duration 110lb./sq. in. mean indicated pressure had been carried, while still maintaining good combustion with a good fuel consumption and a perfectly clear exhaust. It was decided therefore that the new engine should be rated at a mean indicated pressure of 100lb./sq. in. under normally aspirated conditions. Being lightly loaded this scavenge piston ring did not require any lubrication and later a normal type of cast iron spring ring was fitted.

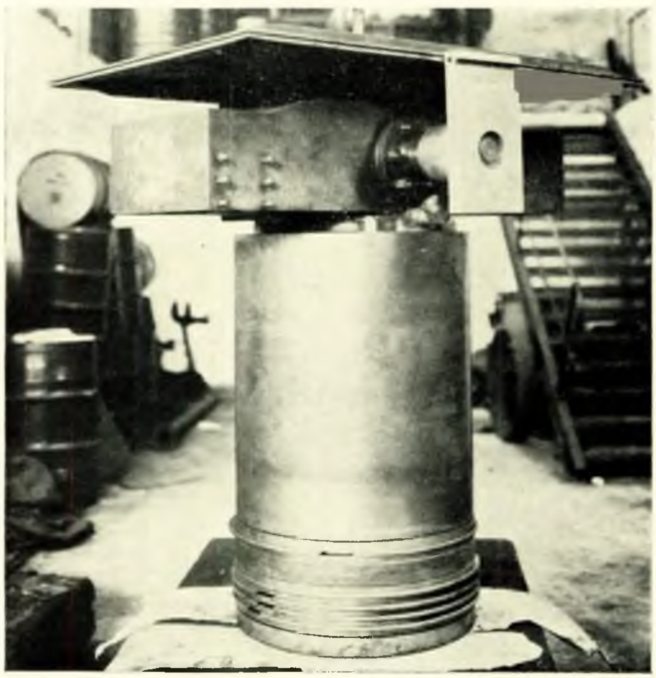


FIG. 4—*Upper piston, transverse beam and scavenge piston of the single cylinder engine*

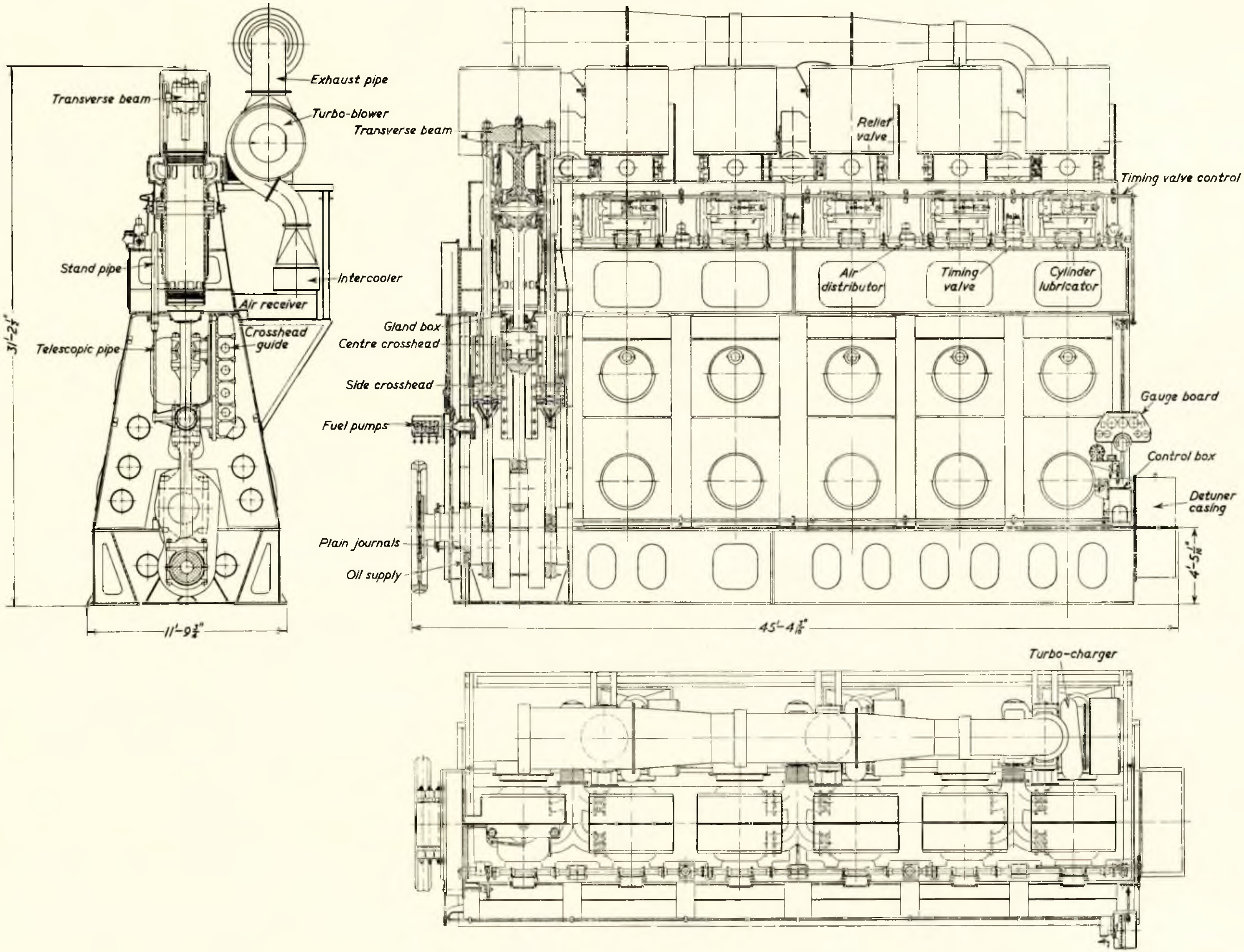


FIG. 5—*Sectional arrangement of the 67PT6 engine*

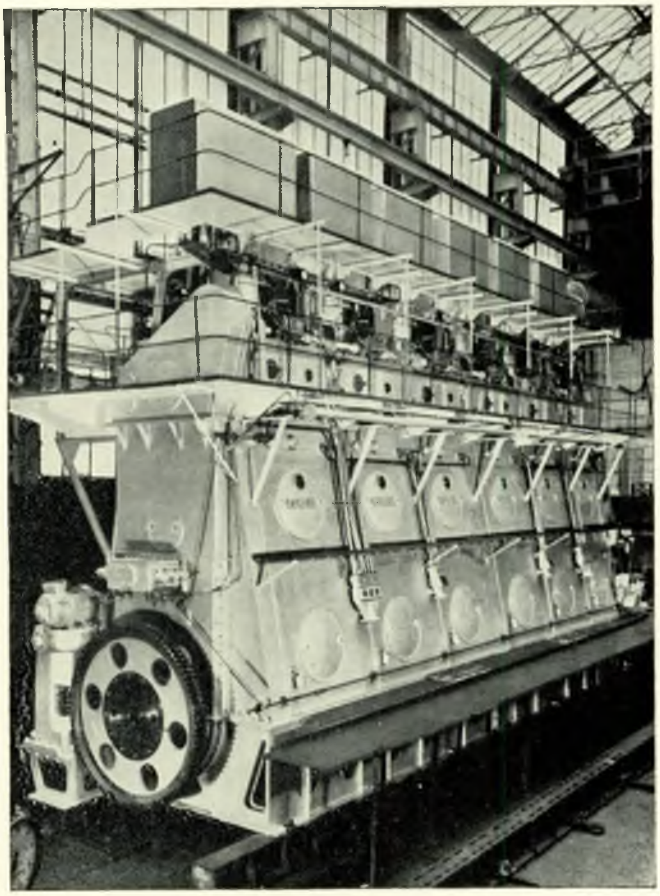


FIG. 6—*The 67PT6 engine*

THE NEW ENGINE

The various features of the new engine will now be described:—sectional arrangements of the engine are shown in Fig. 5 which illustrate the arrangement of the working parts and their compactness and simplicity. Fig. 6 shows the neat appearance of the engine.

The crankshaft of the six cylinder engine is illustrated in Fig. 7 and it was decided to mount this crankshaft in plain bearings and not in spherically mounted bearings as in the 1934 engine. The Doxford engines designed in the 1920's had long bearings of relatively small diameter and the crankshafts were in sections of three throws for each cylinder, coupled together by bobbin pieces with flanged ends. These crankshafts were therefore very long and flexible and, in addition, the standard of manufacture in those days was much inferior to the precision manufacture and finish of the crankshafts of today. In view of this, it was found essential to mount the main bearings on spherical seats so that they could accommodate themselves to the alignment of the crankshaft (see Fig. 1). This practice had persisted throughout the life of this engine. With the much stiffer crankshaft of the new engine with short length large diameter main bearings the author considered that there was no longer any necessity for spherical mounting and the subsequent operation of the engine has confirmed this view. The doglegs between each set of cylinders are therefore short and stiff and, due to the large diameter main journals, there is a considerable overlap between this and the adjacent side crankpins (Fig. 7) to carry the bending moments, as described in the recent paper—"Some Crankshaft Failures—Investigations, Causes and Remedies" read before the Institute early last year. A large diameter rigid coupling is employed in the centre of the engine for coupling the two halves of the crankshaft together and the flexible type of coupling used in the earlier Doxford engines has been eliminated. This has enabled a further reduction to be made in the length of the engine and the total reduction in length on the six cylinder 670 mm. engine, relative to the earlier engine, is more than 13ft. and, in addition, an increase in power of nearly 2,000 h.p. has been obtained. The torsional

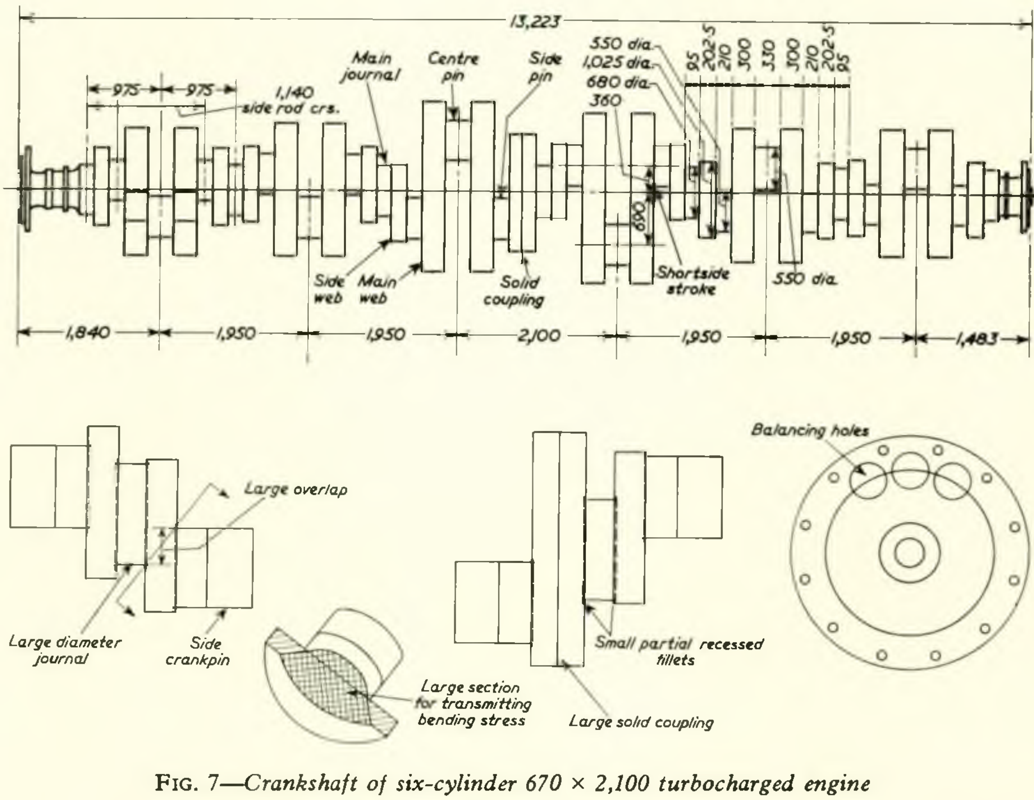


FIG. 7—*Crankshaft of six-cylinder 670 × 2,100 turbocharged engine*

and axial vibration characteristics of the engine have been calculated and considered and where necessary the Doxford Bibby Detuner is used to reduce the stresses of vibration. The several parts of the crankshaft are shrunk together with a shrink interference of 0·002in. per inch of diameter. To provide rotary balance the centre crankwebs are extended and the centre crankpins are bored to reduce the degree of unbalance and the amount of extention required on the centre webs. During the many years of experience in the shrinking of crankshafts, it has been found that this centre lightening hole in the centre crankpin reduces in diameter by some 0·008in. during the shrinking operation, thus relieving the degree of grip and, in addition, whenever a crankweb has been cut away from the crankpin, it has been found that the grip has been much heavier in the centre of the webs than at the ends. This is a well known characteristic. To minimize the reduction in the size of the hole in the centre crankpin and to ensure adequate grip along the full width of the centre webs, bobbin pieces have been pressed into the lightening hole, as illustrated in Fig. 8.

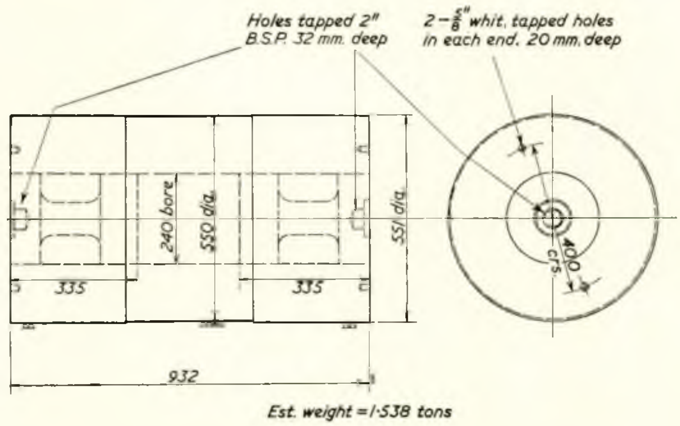


FIG. 8—*Drawing of the centre crankpin with bobbins*

The large dogleg forgings are of 28/32 ton steel, heat treated and annealed and the main webs are machined from forged slabs of 34 ton steel.

The lubrication of the crankpins and journals has been very carefully considered in the light of the experience with the earlier type of engine. On this engine the lubricating oil was supplied through the top of the main bearings by quills and when the crankshaft wore down in service there could be excessive leakage of the hot oil through the clearances and it became very difficult to maintain the oil pressure. It was suspected that this was one of the causes of the difficulties with centre top end bearings which were the last bearings to be lubricated in this circuit. On the new engine the lubricating oil is supplied through the bedplate to the bottom of the main bearing housings and then through a peripheral groove which connects with a diagonally drilled hole through the crankwebs to the side crankpins (see Fig. 9).

The centre of the lower half of the main bearing shell is not grooved so that the white metal on this heavily loaded part is unbroken.

Lubricating oil is supplied up the side connecting rods from the side crankpins to the side connecting rod top end bearings. The lubrication of the centre connecting rod top and bottom end bearings is not through the crankshaft so that there is no drilling of the crankshaft and no cross pipes for this purpose, but the centre connecting rod bearings are supplied with lubricating oil from a telescopic pipe which conveys oil firstly to the top end bearings and the centre crosshead shoe and then down the connecting rod to the centre bottom end bearing (see Fig. 10). This telescopic pipe is situated between the two telescopic pipes which convey the cooling oil to and from the lower piston.

By these various means the engine has been reduced in

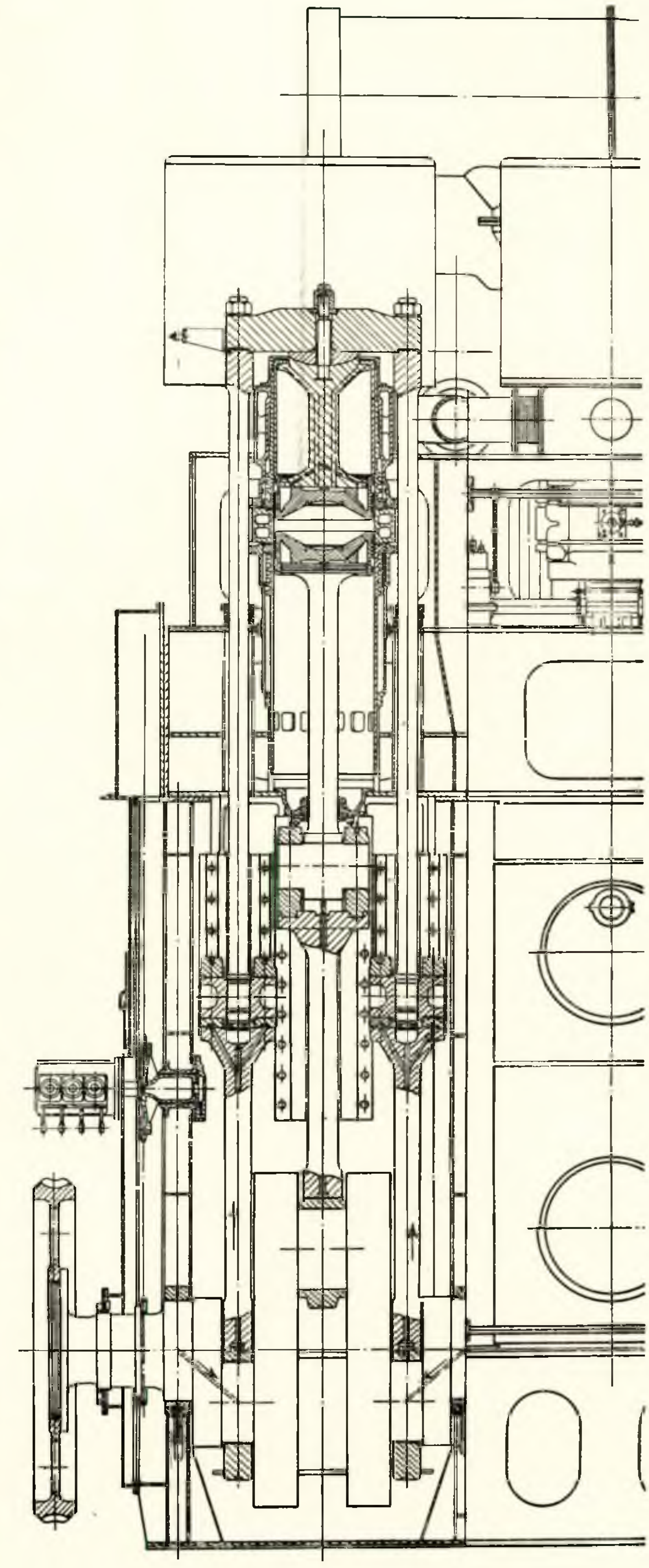


FIG. 9—*Lubricating oil circuit to the main journals, side rod bottom and top ends*

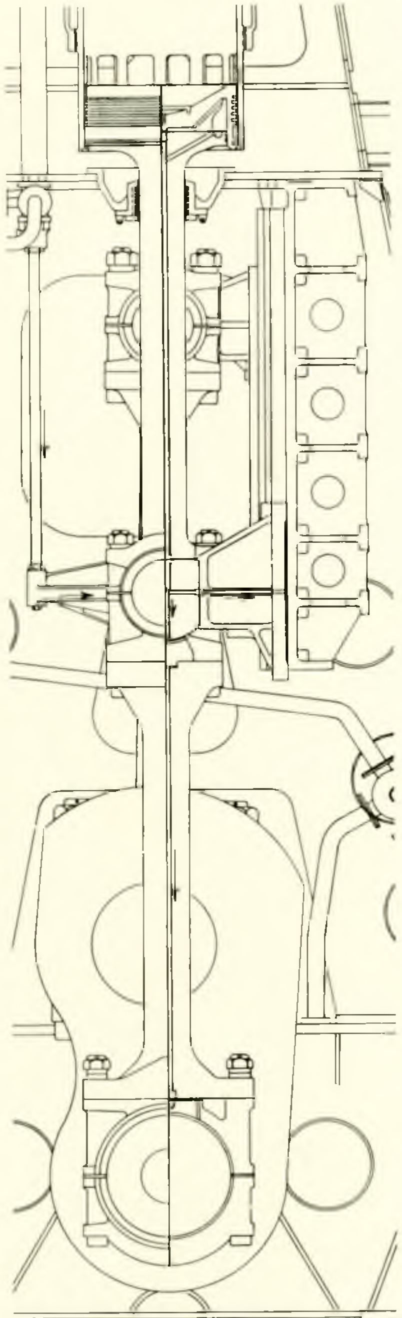


FIG. 10—*Lubricating oil circuit to the top and bottom end bearings*

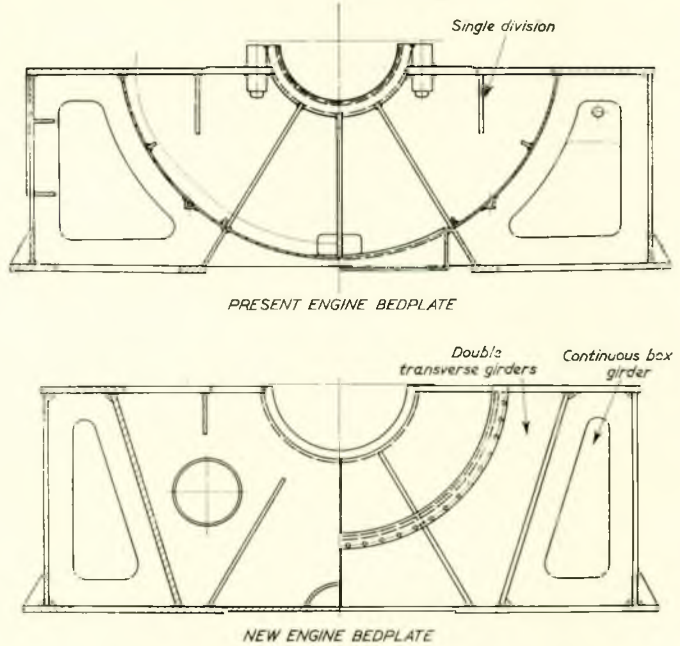


FIG. 12—*Comparison of old and new engine bedplates*

length so that the cylinder centres of the 670 mm. engine are 1,950 mm. giving a ratio of 2·9 whereas the lowest ratio of the older engine was 3·33. This has been accomplished without any increase in the bearing pressures due to the increase in diameter of the crankpins and journals.

Bedplate (See Fig. 12)

The bedplate is of more rigid form than the bedplate of the earlier engines. It is built up of two longitudinal box girders which extend over the full length of the engine, the transverse girders which incorporate the main bearing housings are welded to these longitudinal girders and on the six cylinder engine there are double transverse girders carrying these housings. The bedplate has also been increased in height to the crankshaft centre line Fig. 12, so that there is a considerable increase in rigidity. The main bearing housings are bored parallel, this having been made possible by the increased rigidity of the crankshaft and the short length of the main bearing journals. The complete bedplate may now be stress relieved in a new furnace installed by the author's company, though this is a somewhat costly operation and Lloyd's and other classification societies only require the transverse girders, which are carrying the bearing housings, to be stress relieved. The lower half main bearings are plain steel shells lined with white metal of relatively thin wall type, the white metal being 5 mm. thick. Such bearings require the minimum of fitting and they are held in place by the upper bearing keeps which are cast steel straps again with thin white metal linings.

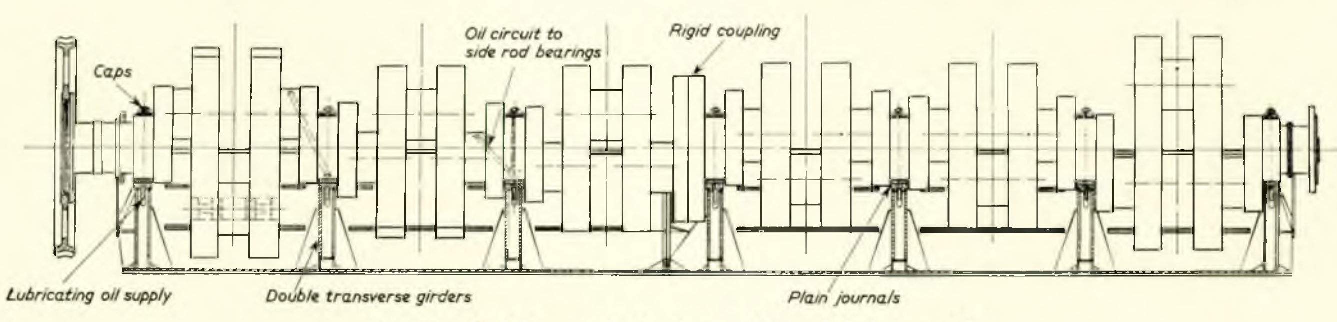


FIG. 11—*Bedplate, main bearings and crankshaft assembly*

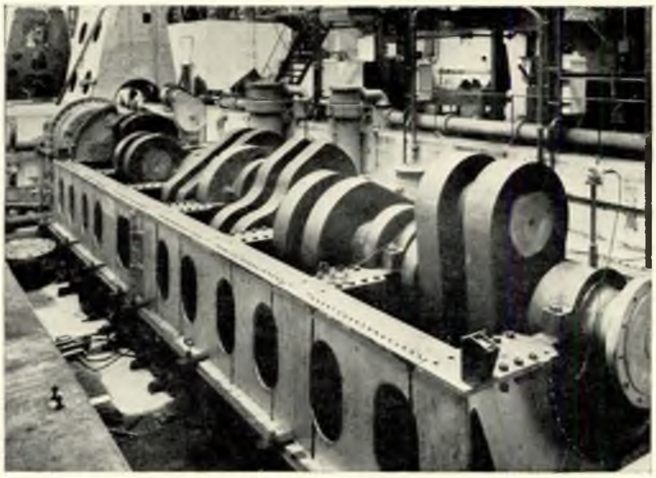
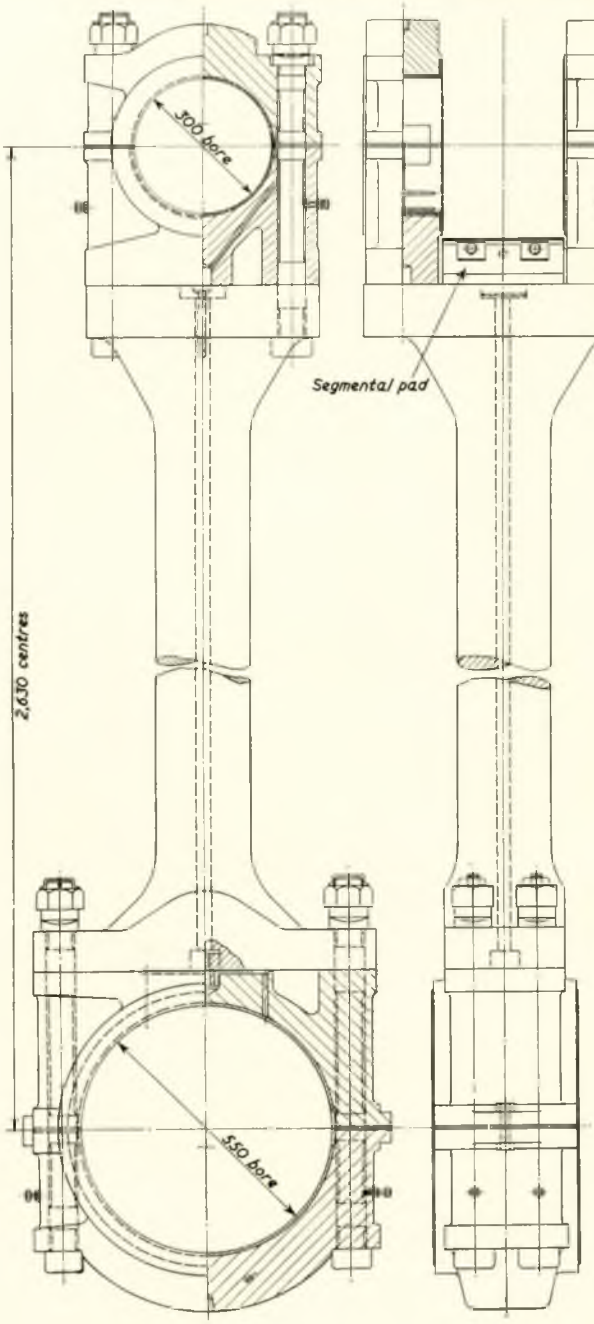
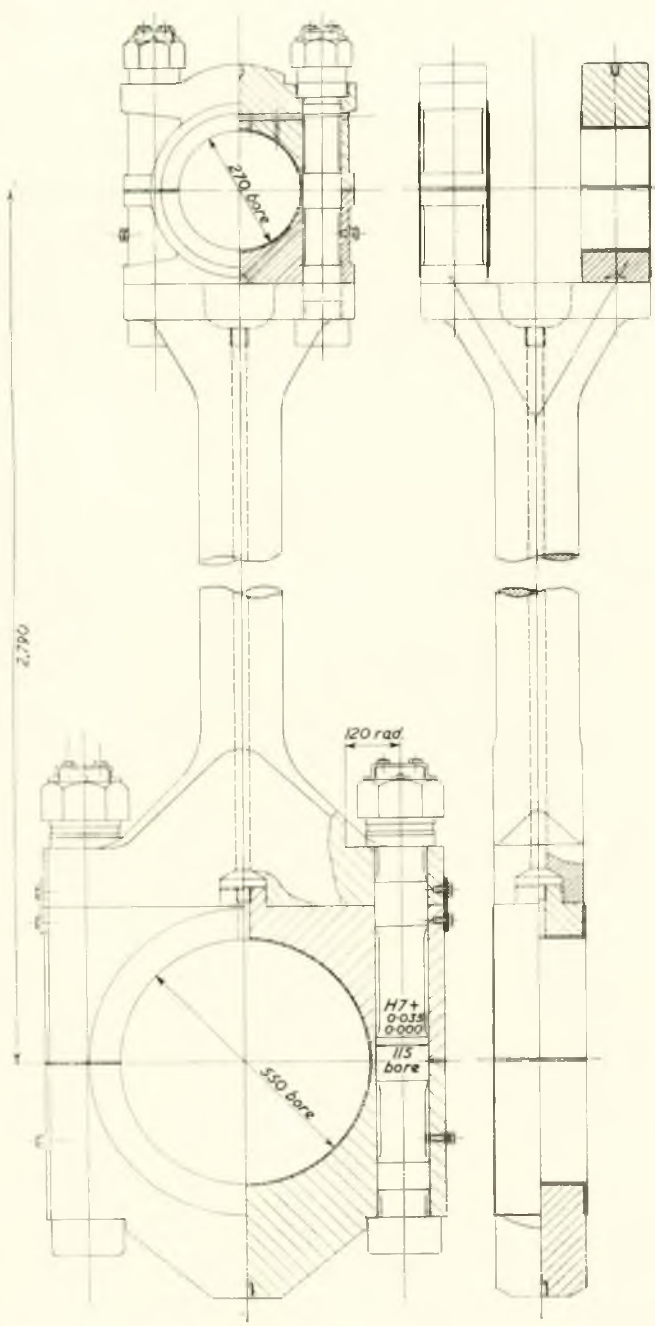


FIG. 13—(left) *The crankshaft in the bedplate*

FIG. 14—(bottom left) *Drawing of the centre connecting rod*

FIG. 15—(bottom right) *Side connecting rod*





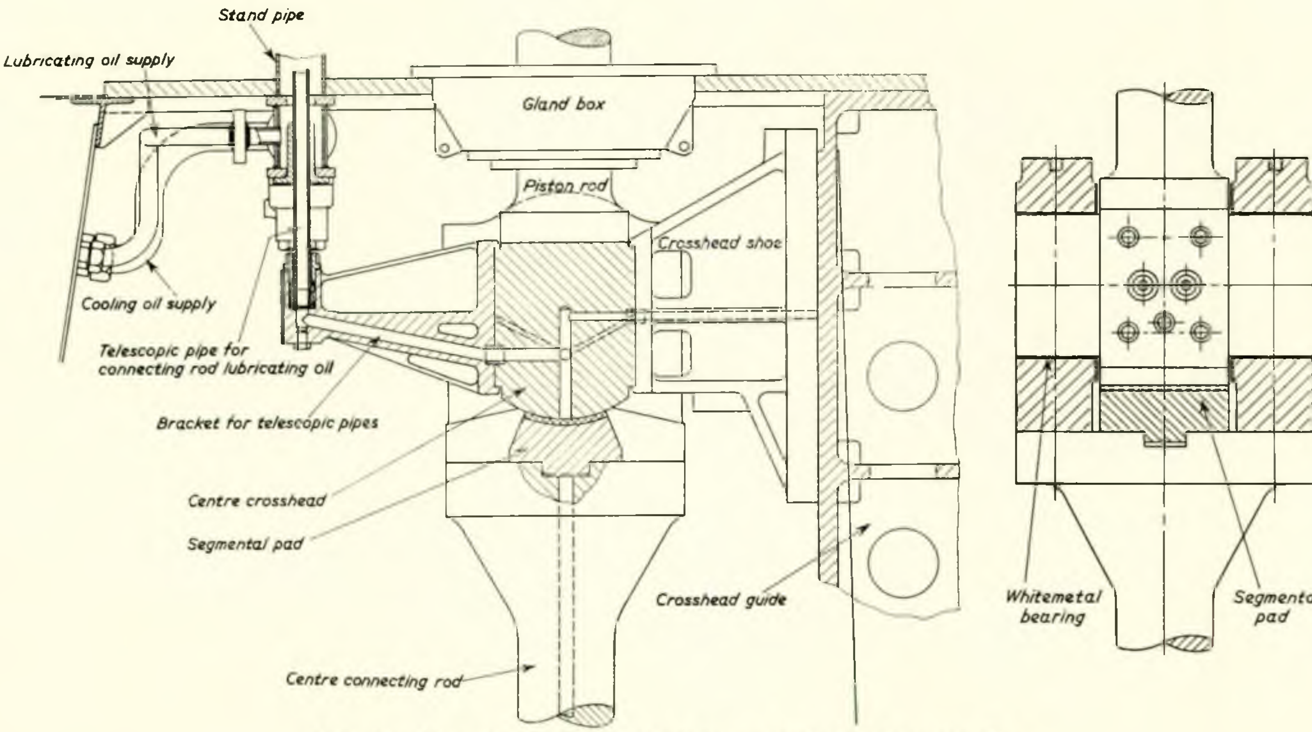


FIG. 16—*Centre top end with its pad, crosshead and shoe assembly*

The bedplate, main bearings and crankshaft are illustrated in Fig. 11, and Fig. 13 shows a photograph of the crankshaft in place in the bedplate.

Connecting Rods (See Figs. 14 and 15)

Both the centre and side connecting rods are simple forgings with palm ends at both ends to which the top and bottom half bearings are bolted. The centre connecting rod top end also has a bearing pad in the centre which takes some of the load transmitted by the piston rod directly on the underside of the centre crosshead. The ratio of the length of the centre connecting rod to the centre crank throw is 3·85 and the ratio of the length of the side connecting rods to the side crank throws is 7·85. The side connecting rod bottom end bearings are machined from steel slab forgings and are thin white metal lined and supplied with lubricating oil from the main bearings as described previously. The top end bearings are of cast steel thin white metal lined and supplied with oil up the side connecting rods. The centre connecting rod has a cast steel bottom end bearing thin white metal lined and the centre top end bearings are also of cast steel with again thin white metal linings but in addition there is the centre pad which is a steel segment lined with copper lead and this has proved very successful in operation. The centre top end with its pad, crosshead and shoe assembly is shown in Fig. 16.

Centre Crossheads (Fig. 16)

The centre crossheads are machined from simple forgings of flame hardening 0·45 carbon steel and have cylindrical bearing surfaces on the ends flame hardened and super finished for the top end bearings. The centre portion of this crosshead is turned cylindrical but afterwards three flat faces are machined on it two being vertical and one horizontal. The lower piston rod is bolted to the upper horizontal surface and the crosshead shoe is bolted to the back vertical face while the front vertical face carries the bracket for the telescopic pipes. There are three telescopic pipes, two being for cooling oil and the centre one for the lubrication of the top end bearings. The underside of the centre crosshead forms a cylindrical surface which bears onto the top end bearing pad as previously described. All the bearing surfaces are flame hardened and super finished. The cooling oil supplied to and returned from the lower piston is transferred through the centre crosshead which is also drilled for the lubricating oil to its various bearing surfaces.

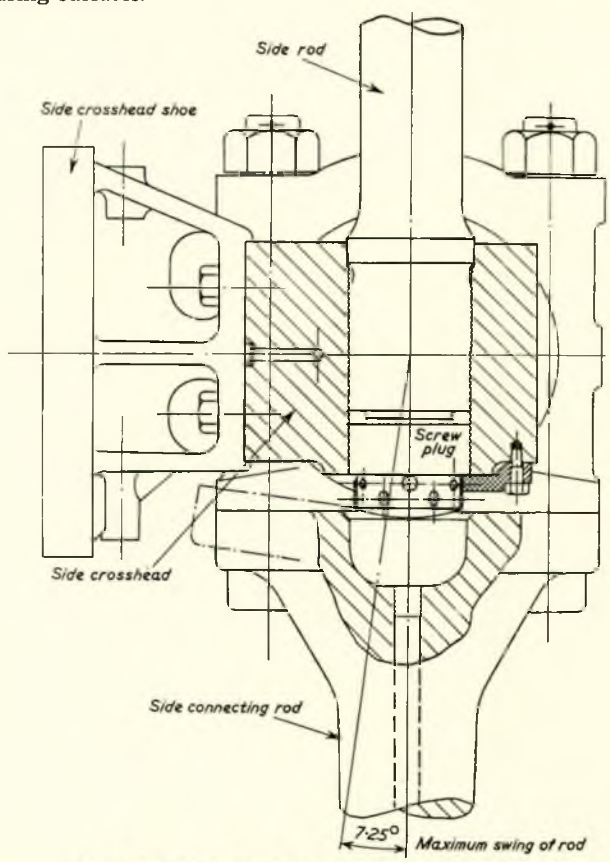


FIG. 17—*Side rod and crosshead assembly*

Side Crossheads (Fig. 17)

The side crossheads are machined from simple forgings with cylindrical bearing surfaces on the ends. The side rods are screwed directly into the side crossheads and locked by screwed plugs on the underside, and in addition to locking the side rods and preventing them from turning, these plugs pre-load the threads of the side rods to minimize the dynamic loads to which they are subjected. The arrangement of the side crosshead with side rod attachment, locking plug, and crosshead shoe is shown in Fig. 17. This arrangement has been entirely successful and has enabled the differential nut construction employed on the earlier engines to be eliminated since there was some considerable trouble with these differential nuts due to crossed threads and to fretting or shearing of the threads. The crosshead forgings for the earlier engine were also of very large size due to the projection into which the side rod and differential nut were screwed.

Lower Piston Rods (Fig. 18)

The lower piston rods are simple forgings, the lower ends being square and palm ended for bolting to the crossheads. The upper ends are cylindrical and form the faces to which the piston heads are bolted. Both the palm end and the cylindrical end have large radii sweeping into the body of the rod thus avoiding concentration of stresses. The oil for cooling the lower piston is transmitted through the crosshead and up the piston rod to the piston head through a drilled hole and is then returned through a similar drilled hole. Adjustment of the lower piston stroke relative to the upper piston stroke has achieved balance without the necessity of enlarged piston rods. The body of the piston rod works in a gland attached to the underside of the entablature. This gland separates the crankcase from the scavenging air space and consists of a number of ring segments held to the body of the piston rod by garter springs. These rings are so arranged that the lower ones scrape oil off the rod back into the crankcase, whereas the upper ones prevent any sludge from the combustion of heavy boiler fuel from falling into the crankcase. This sludge is collected and drained away from the upper well of the gland. The arrangement of the lower piston rod assembly to the lower piston and to the crosshead with the gland is shown in Fig. 18 and Fig. 19 shows a photograph of the assembly.

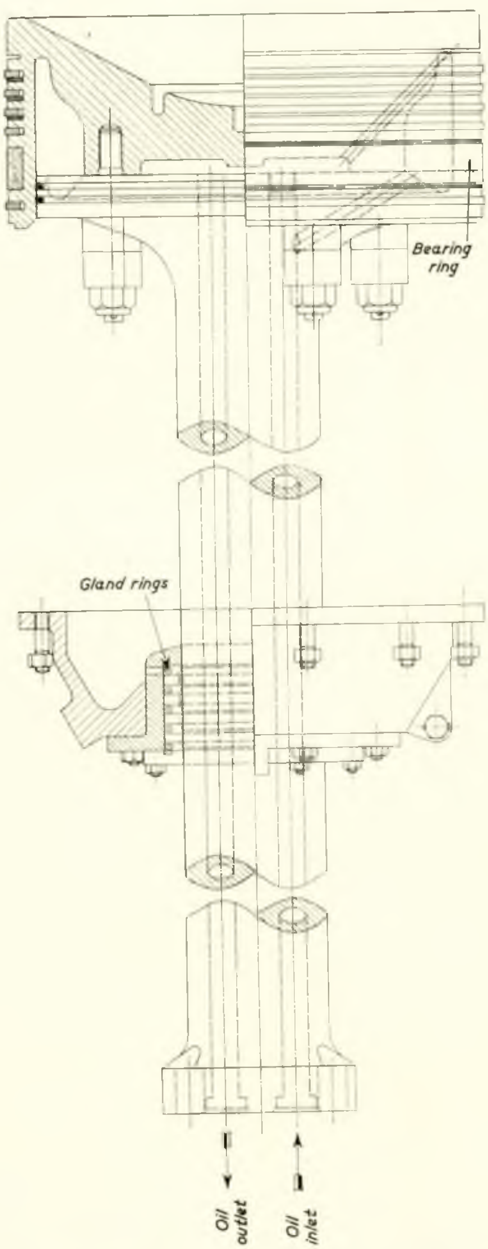


FIG. 18—*Assembly of lower piston rod, piston head and gland*

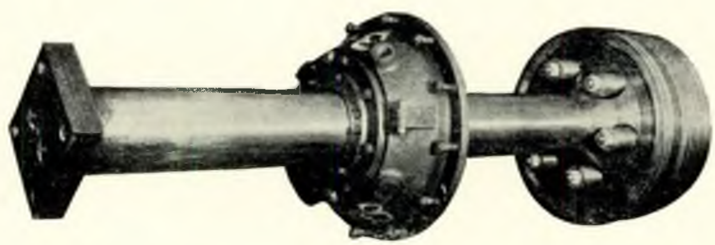


FIG. 19—*Assembly of lower piston rod, piston head and gland*

Piston Heads (Fig. 20)

The upper and lower piston heads are identical and are machined from steel forgings of a relatively ductile steel. The crowns are dished to give a spherical form to the combustion space and the piston heads are free to expand without causing undue stress. The piston crowns are attached to the upper cylindrical face of the piston rod and are supported in the centre so that they are free to expand towards the periphery. The under face of the piston head is machined to form a cooling space between the piston head and the upper face of the piston rod and the cooling oil is supplied up the rod as described previously. A cast iron ring is fitted around the piston head to form a bearing surface; four compression rings are fitted into the grooves above this bearing ring and there is one ring below it to act both as a compression ring and as a lubricating oil spreader ring. Inward springing bearing rings are fitted to the underside of the piston ring grooves onto which the main piston rings bear thus preventing undue wear of the grooves in the piston head. Fig. 20 shows a drawing of the piston head.

Upper Piston Rods and Skirts (Fig. 20)

The upper piston rod is a simple forging with cylindrical ends one being bolted to the upper piston head and the other to the transverse beam which carries the load from the piston to the side rods. This upper piston rod is a short length forging drilled with two holes for conveying the cooling water to and from the upper piston head. The upper piston is attached to the piston rod in the centre of the crown so that it is free to expand in a like manner to the lower piston head. A light piston skirt is provided around the upper piston rod which reciprocates in the upper cylinder liner to shield the exhaust ports and prevent the exhaust gases passing back into the open end of the cylinder and contaminat-

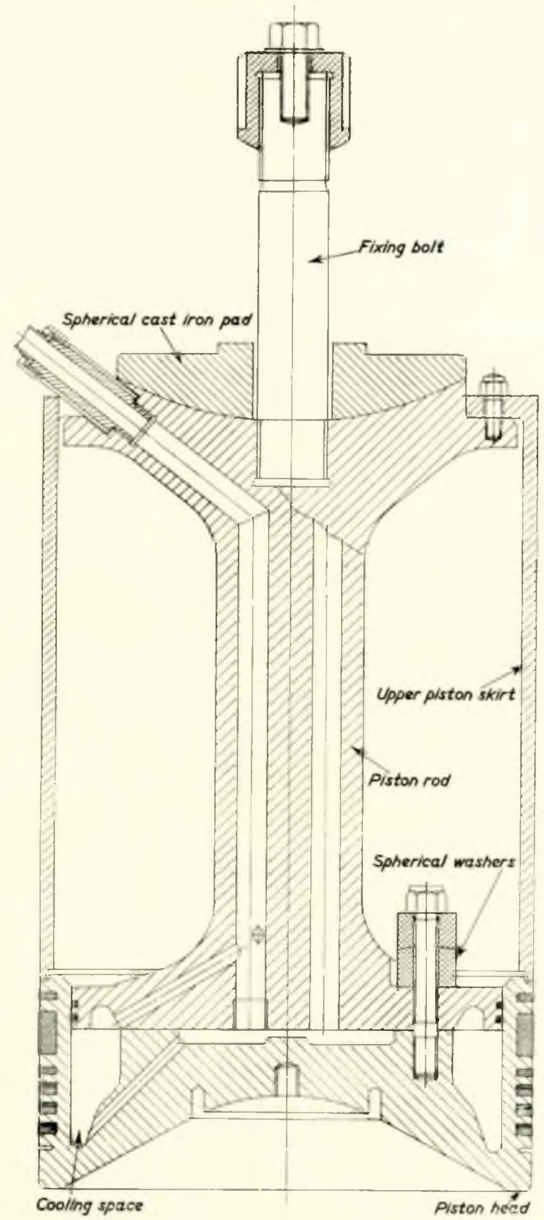


FIG. 20—*Upper piston rod, piston head and skirt assembly*

ing the atmosphere. This piston skirt is bolted to the upper face of the piston rod, as shown in the assembly drawing Fig. 20.

Transverse Beams (Figs. 21 and 22)

The transverse beams are solid steel forgings to which the upper pistons and side rods are attached. The upper piston rod is attached to the transverse beam by a centre bolt and a spherically machined cast iron pad is inserted between the piston rod and the transverse beam to permit self aligning of the piston. The cooling water telescopic pipes are attached to the transverse beam for conveying water to and from the upper piston. The arrangement of the upper piston with its rod and skirt is shown in Fig. 20 and Fig. 22 shows the attachment to the transverse beam. In the earlier engine the upper piston rod was attached to the transverse beam through a shaft which carried bearings, and the side rods were attached to the ends of the transverse beams through pins (see Fig. 21). These were unlubricated, and the slight movement of these pins caused by slight misalignment or incorrect adjustment or phasing of the two side crank pins resulted in wear of the centre pin and its cast iron bushes or fretting corrosion on the unlubricated side pins. It was therefore decided on the new engine to attach the side rods to the transverse beam in the rigid manner shown and to permit self-alignment of the piston relative to the transverse beam by means of the spherical pad.

Cooling of the Lower Piston Head (Figs. 5, 16 and 18)

As previously described the lower piston head is oil cooled, the reason for this being that with any type of cooling system there is bound to be leakage from either the joints of swinging links or from the glands of telescopic pipes, and it was found in the early days of operation on boiler fuel that when the lower pistons were water cooled leakage of water into the lubricating oil in the crankcase mixed with any sludge which penetrated into the crankcase from the combustion of the heavy boiler fuel and formed a dilute sulphuric acid which could cause severe corrosion of the crankshaft. It was decided therefore, to cool the lower pistons by lubricating oil through telescopic pipes and as previously described a bracket attached to the lower crosshead carries two telescopic pipes for the conveyance of the cooling oil to and from the piston, the oil being supplied from stand pipes mounted in the entablature. The oil is transferred from the telescopic pipes to the crosshead, up the piston rod and circulated through the piston head as shown in Figs. 5, 16 and 18. The cooling oil is circulated from the lubricating oil tank by a separate pump and cooler and air snifting valves are provided on the top of the stand pipes to minimize hydraulic hammer. The cooling oil is kept at a low temperature to avoid carbonization, the temperature of the inlet oil being about 110 deg. F. and that of the outlet

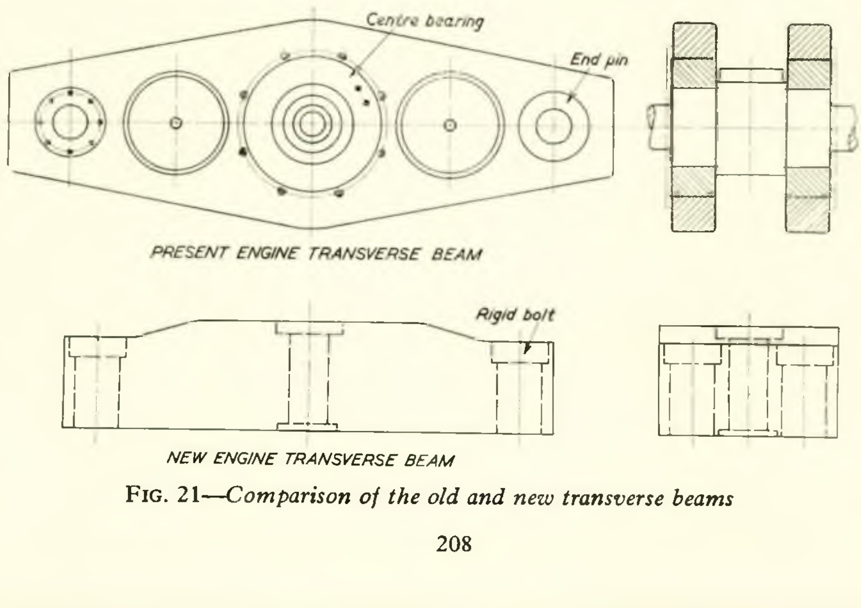


FIG. 21—*Comparison of the old and new transverse beams*

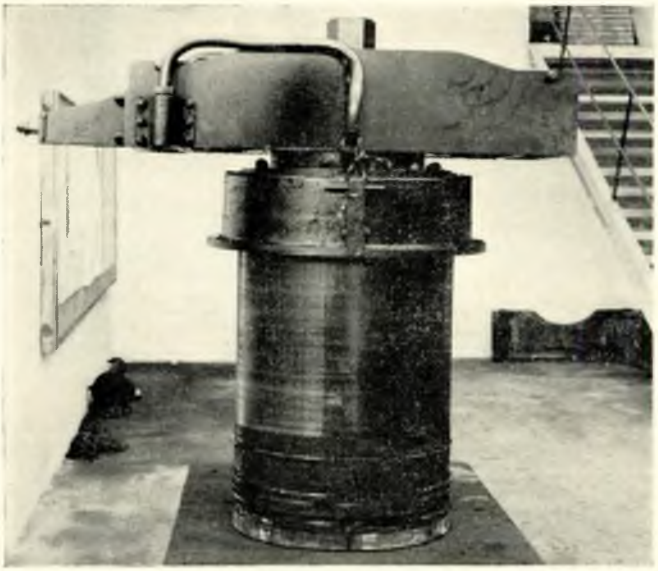


FIG. 22—*Piston and transverse beam assembly*

oil not exceeding 132 deg. F. at full load. The telescopic pipes are mounted on spherical bearings which permit of their self-alignment. This system has been entirely successful and is much simpler than the swinging link system of the earlier engines and since there is no possibility of water leaking into the lubricating oil there has not been and, cannot be, any case of corrosion.

Cooling of the Upper Pistons (Fig. 23)

Water has been retained as the cooling medium of the upper pistons since water is a better coolant than oil, and there is no question of corrosion due to leakage of water into the lubricating oil system from the upper pistons. Also should there be any leakage of the cooling medium from the upper cooling system it is easier to clean up a pool of water than a pool of oil. The cooling water is supplied to the upper piston through a telescopic pipe mounted on the side of the exhaust belt and an identical pipe conveys the water away after it has passed through the piston rod and piston head as shown in Fig. 23. Due to water being a better cooling medium than oil and having a higher specific heat, less than one half of the quantity of water has to be circulated through the piston for the same degree of cooling and since the upper piston is hotter than the lower piston water as the cooling medium keeps the temperature down to a satisfactory level and there is no danger of carbonizing of the cooling medium in the piston heads as with oil cooling. The cooling water temperature can therefore be quite high and it is usual to have an inlet temperature of 140 deg. F. and an outlet temperature of 170 deg. F. or even higher at full load of the engine.

Cylinder Liners (Figs. 24 and 25)

The cylinder liners are constructed in two pieces the lower incorporating the scavenging air ports and the upper

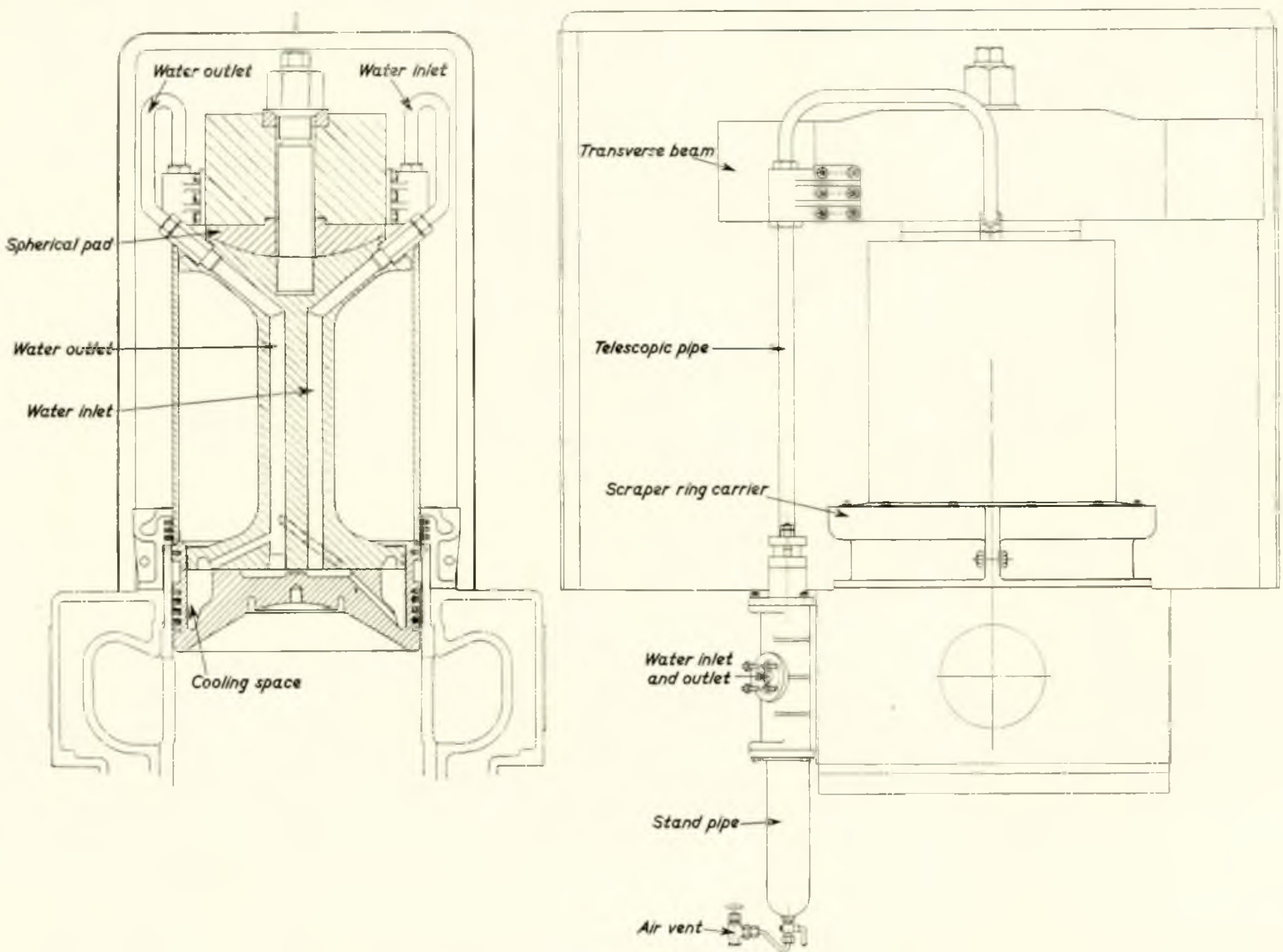


FIG. 23—*Drawing of the upper piston and cooling circuit*

FIG. 24—*Cylinder liners and combustion belt*

incorporating the exhaust ports. These liners are relatively simple, short length castings and are surrounded by water jackets which are bolted to a central cast steel combustion chamber thereby clamping the cylinder liners to this chamber through spigoted copper joints. Each half of the liner can be replaced separately if desired. The cylinder liners are relatively thin but have supporting ribs for the passage of the cooling water and steel rings are shrunk onto these ribs to provide additional strength to the cylinder liners for taking

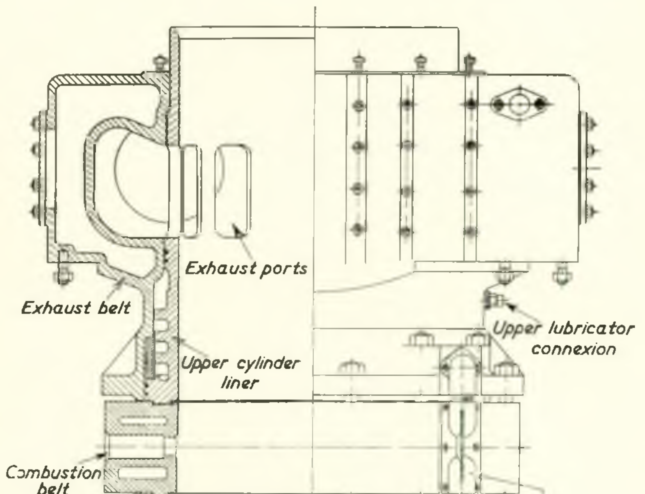


FIG. 25—*Drawing of the cylinder liners, jackets, combustion chamber and exhaust belt assembly*

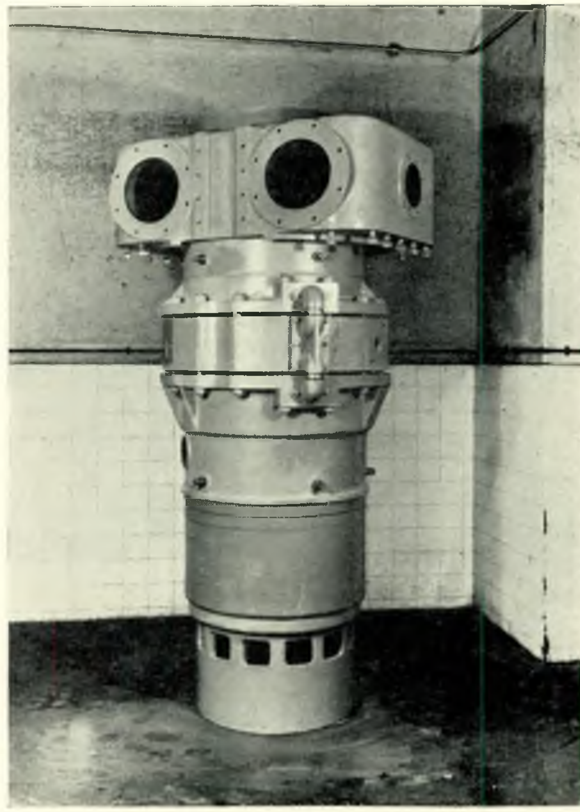


FIG. 26—*Cylinder liners, jackets, combustion chamber and exhaust belt assembly*

the combustion loads. The whole arrangement is water cooled, the cooling water entering the lower cylinder jacket circulating through the grooves of the lower cylinder liner and then being transferred externally to the combustion belt round which it circulates twice. It is then transferred externally to the upper cylinder jacket around the upper liner grooves and then into the exhaust belt before being taken away to the cooling water hoppers. Due to the cooling water being transferred into the combustion belt through external fittings there is no water on either the upper or lower liner joints and there is thus no possibility of water leaking into the cylinders. Lubricator fittings are provided for the supply of lubricating oil to both the upper and lower cylinder liners and the lubricating points are equally distributed around the liner and supplied with oil from mechanical type sight feed lubricators. A photograph of the separate cylinder liners and combustion belt is shown in Fig. 24, and Fig. 25 shows a drawing of the assembly.

Combustion Belts (Fig. 25)

The combustion belts are hollow steel castings with a central rib and cooling water is supplied from the lower cylinder jacket circulated twice around the combustion belt and then passed to the upper cylinder jacket through external connexions. This central combustion belt has front and back fuel valves situated opposite each other for supplying fuel into the combustion chamber through multi-hole nozzles. These fuel valves are mounted directly on the metal of the combustion belt and thus there is no necessity for the packing glands and tightening devices of the earlier engine. The starting air valve, relief valve and cylinder indicator connexion are all mounted in the combustion belt. Fig. 27 shows the assembly of the combustion belt and cylinder liners with the pistons in the position which they occupy at the end of the compression stroke. The fuel valves are show in position.

Exhaust Belts (Fig. 25)

The exhaust belts are iron castings, water jacketed, which

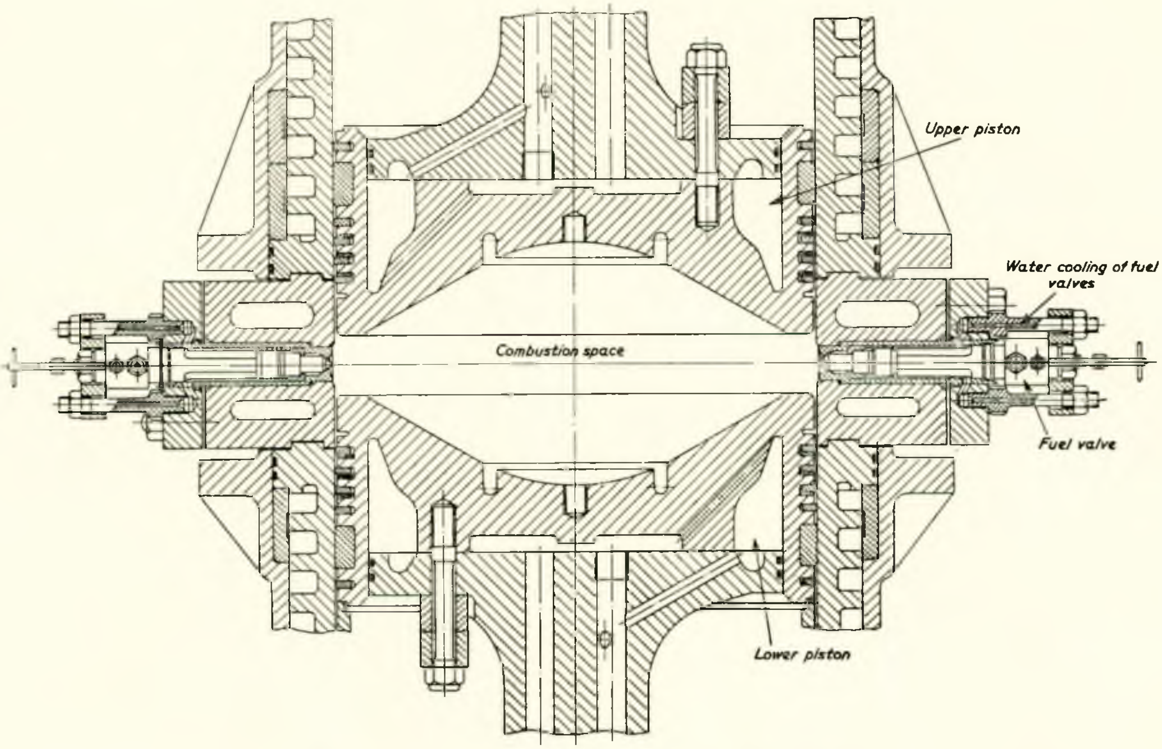


FIG. 27—*Assembly drawing of the combustion space*

surround the upper portion of the cylinder liner and have passages for conveying the exhaust gases from the upper cylinders to twin outlets between each pair of cylinders. These exhaust belts are mounted on top of the entablature and are extended to form the water jackets for the upper cylinder liners and are thus bolted direct to the central combustion chamber. The twin exhaust outlets arranged between each pair of cylinders simplify the exhaust pipe layout and give advantages on the turbo-charged engines due to their short length and small volume. They are connected directly into the turbo-chargers without bends and in this way the impulse energy of the exhaust gases is preserved for doing work in the turbines. Rubber ring joints are provided in the upper cylinder liners for sealing the water spaces from the exhaust ports. The assembly of the exhaust belt, upper cylinder liner, combustion belt and lower cylinder liner is illustrated in Fig. 25, and Fig. 26 shows a photograph of the assembly. After the cooling water has passed through the upper cylinder liner into the exhaust belt it is conveyed away to visible cooling water hoppers. A scraper ring carrier is provided on the upper side of the exhaust belt for preventing the passage of the exhaust gases past the upper piston skirt and for scraping the lubricating oil downwards and preventing it being thrown about.

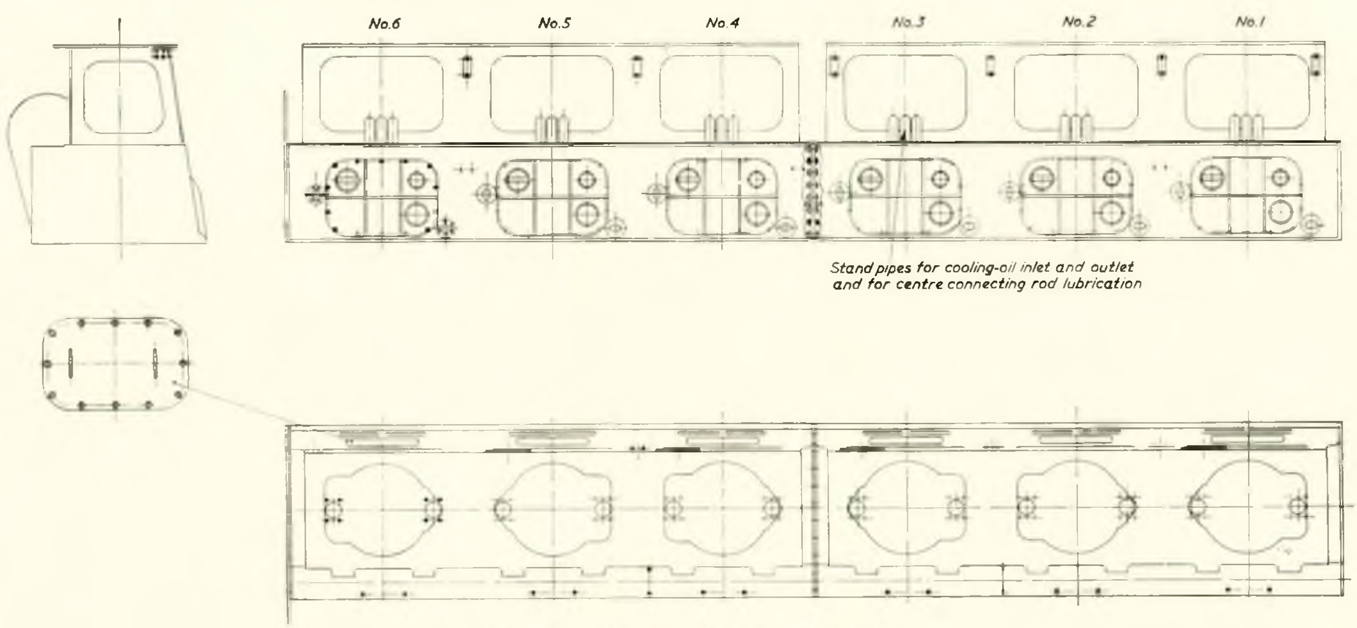


FIG. 28—*Drawing of the entablature*

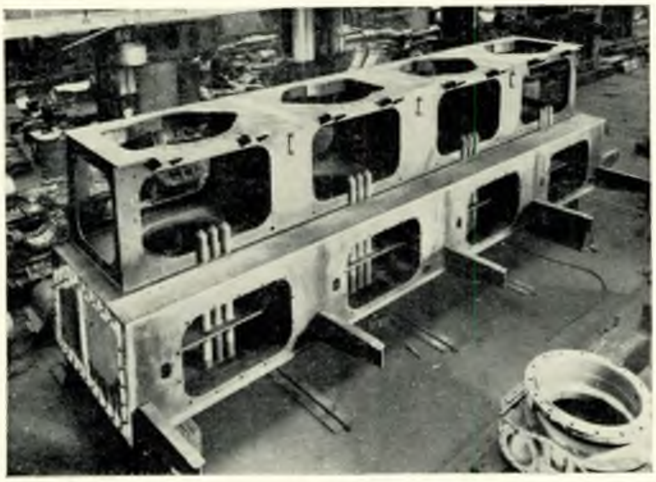


FIG. 29—*Photograph of the entablature*

Entablature (Fig. 28)

The entablature is mounted on top of the engine columns which extend from the bedplate and contain the rotating and reciprocating parts of the engine, crossheads, crosshead guides, connecting rods, etc. The entablature extends over the whole length of the engine and carries the engine cylinders which are supported from the exhaust belts as described previously. The lower portion of the entablature forms the receiver for containing the scavenging air to be supplied through the ports of the lower cylinder liner into the engine. A diaphragm chamber is incorporated in the entablature, the diaphragm glands being mounted on its lower face. The whole forms a very rigid construction and tubes are provided for the passage of the side rods through the entablature which are sealed at their upper ends with white metal bearings and sealing rings. An improved form of diaphragm gland assembly is mounted on the entablature and the piston rod passes through this and the gland can be withdrawn either upwards with the piston rod or can be lowered into the crankcase for examination or replacement of any of the diaphragm rings. The stand pipes for the piston cooling and centre top end lubrication are incorporated in the entablature, as shown in Fig. 28. Fig. 29 shows a photograph of the entablature.

Fuel Injection System (Fig. 30)

The timing valve injection system is employed on this engine for the injection of fuel into the cylinder. This has been described previously in the paper at the Symposium[1] on "Recent Developments in Marine Diesels". A multi-plunger pump is arranged at the aft end of the engine and is driven from the same chain which drives the camshaft. A separate pump unit is provided according to the number of cylinders on the engine (see photograph Fig. 31), but in normal operation all pump units discharge into a common rail. These fuel pumps charge accumulator cylinders with fuel oil to a pressure of 6,000 to 8,000lb./sq. in. and a separate timing valve for each cylinder is mounted over the camshaft to allow the oil from the accumulator cylinders at this pressure to operate the automatic spring loaded fuel valves as shown in Fig. 30. These valves are mounted on the combustion chamber and lift when the timing valves are operated by their cams on the camshaft. The operation of the timing valves regulates the opening and closing of the fuel valves and thus the commencement and end of injection into the cylinder. This governs the duration of fuel injection according to the load on the engine and the injection period can be regulated by the control lever situated on the engine room floor level. The duration of opening of each timing valve can be adjusted by means of a turnbuckle situated between the lay shaft and each timing valve thus regulating the load carried by each cylinder. The time of injection of fuel into each cylinder can be regulated by adjustment of the timing valve cam toe. The quantity of fuel delivered by each fuel pump can be regulated by a screw on each fuel plunger control rack. The quantity of fuel delivered by the fuel pumps to the accumulator cylinders is controlled by a handwheel which regulates the

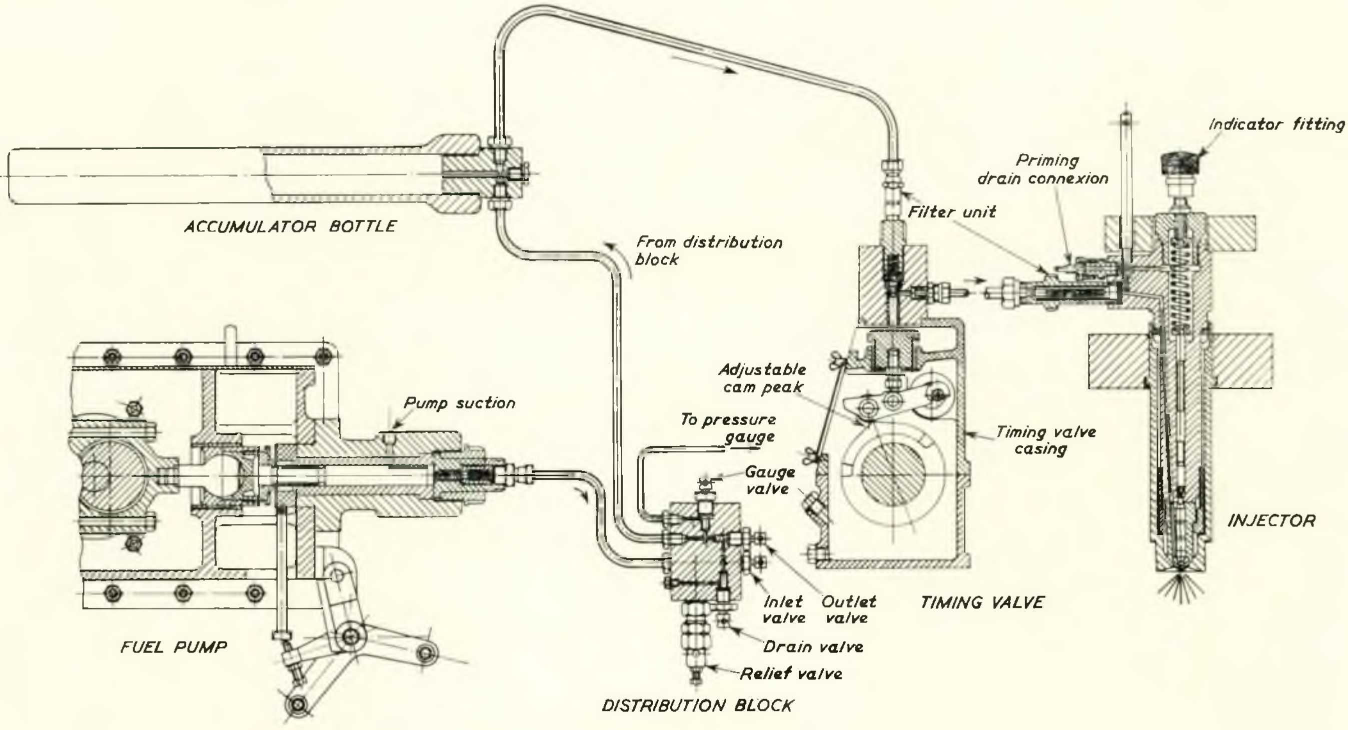


FIG. 30—*Fuel injection system*

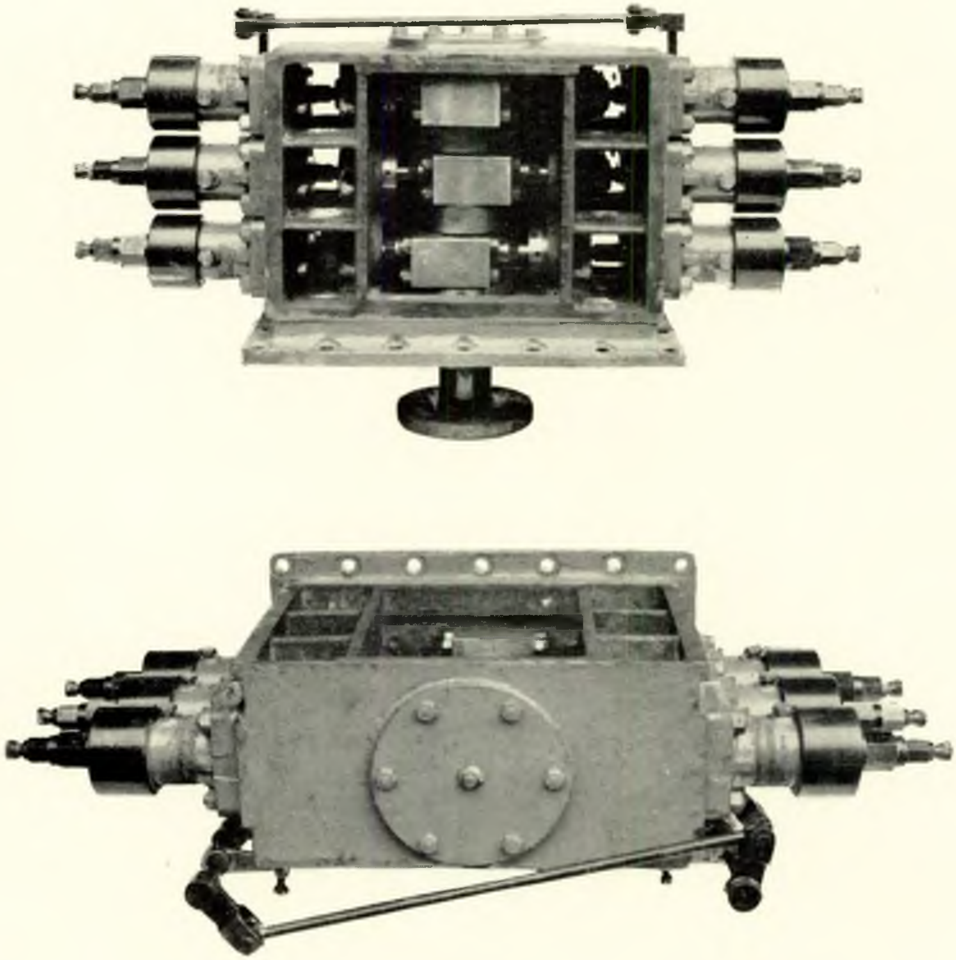


FIG. 31—*Fuel pump*

fuel pump racks. Each fuel pump plunger has a helical groove which shuts off the pump suction port and the pump then delivers fuel under pressure to the accumulator cylinders. This rotary movement of the fuel pump plungers regulates the fuel quantity and the load and speed of the engine. An overspeed governor of the Aspinall type is connected to the fuel pump and when this operates all the fuel plunger racks are lifted so that no fuel is delivered to the accumulator cylinders. Thus, should a propeller come out of the water the engine speed is limited by the overspeed governor to the speed at which it is set. Double timing valves are employed on this engine to ensure continued operation should one stick open as shown in Fig. 32.

The Starting Air System (Fig. 33)

The starting air system is also as fitted to earlier engines. Air valves mounted on the combustion chamber of each cylinder are operated pneumatically by pilot air to allow starting air under a pressure of 600lb./sq. in. from the starting air bottles to enter the cylinders and thus give motion to the pistons. These air valves are operated pneumatically by pilot air from a rotary distributor which governs the time and duration of opening of the starting valves in correct phase according to the firing order for starting the engine in either the ahead or astern direction. The starting air lever at the engine control station operates a change valve for admitting pilot air to the ahead or astern side of the distributor. In general the starting air distributors are mounted as close as possible to the starting valves which they operate there being two on five and six cylinder engines. The system is illustrated in Fig. 33. Normally two starting air bottles are fitted to six cylinder engines each of about 140 cu. ft. capacity and when charged to 600lb./sq. in. either of these will give more than the twelve starts required by the classification societies.

Engine Control Box (Fig. 34)

The engine control box contains two levers and a hand-wheel. One lever operates the starting air system to start the engine in either the ahead or astern direction of rotation according to whether it is pushed forward or pulled backwards and no reversing lever is required. The other lever controls the duration of opening of the fuel timing valves and thus regulates the fuel injection pressure. The handwheel controls the quantity of fuel delivered by the fuel pumps and thus regulates the load and speed of the engine.

The two levers are interlocked, so that when the air lever is moved forwards or backwards to give starting air to the engine the timing valve control lever can only be moved to notch 3 thus permitting only a small quantity of fuel to be delivered to the engine when starting air is on the cylinders. Similarly when more fuel is being supplied and the timing valve control lever is at any position between notches 3 and 10, the interlocking prevents the air starting lever being moved from its vertical position.

Thus the sequence of starting is to open the starting air valve on the air vessels and then move the starting air lever on the control box forwards or backwards when the engine will start in the ahead or astern direction. The fuel hand-

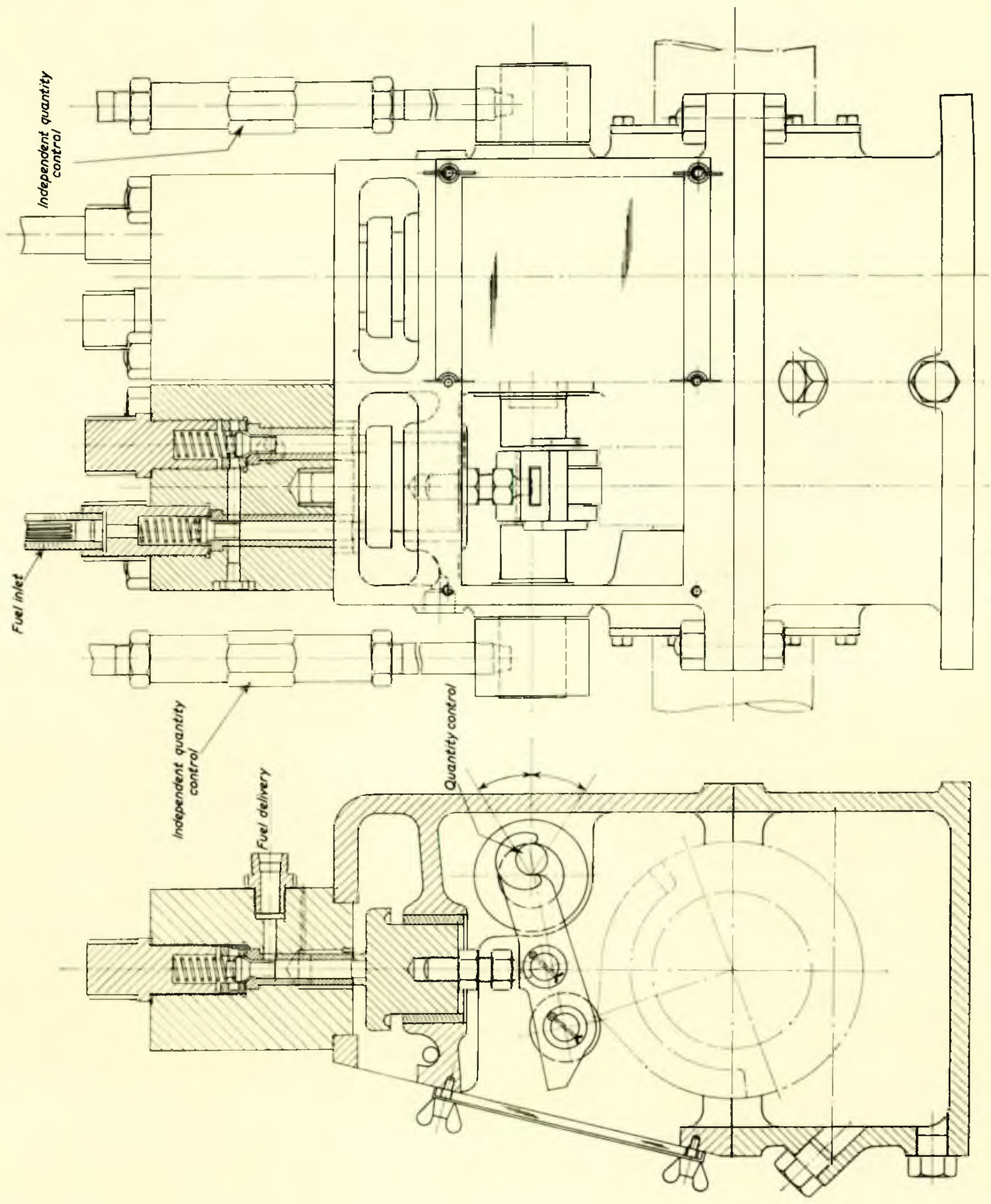


FIG. 32—*Double timing valve*

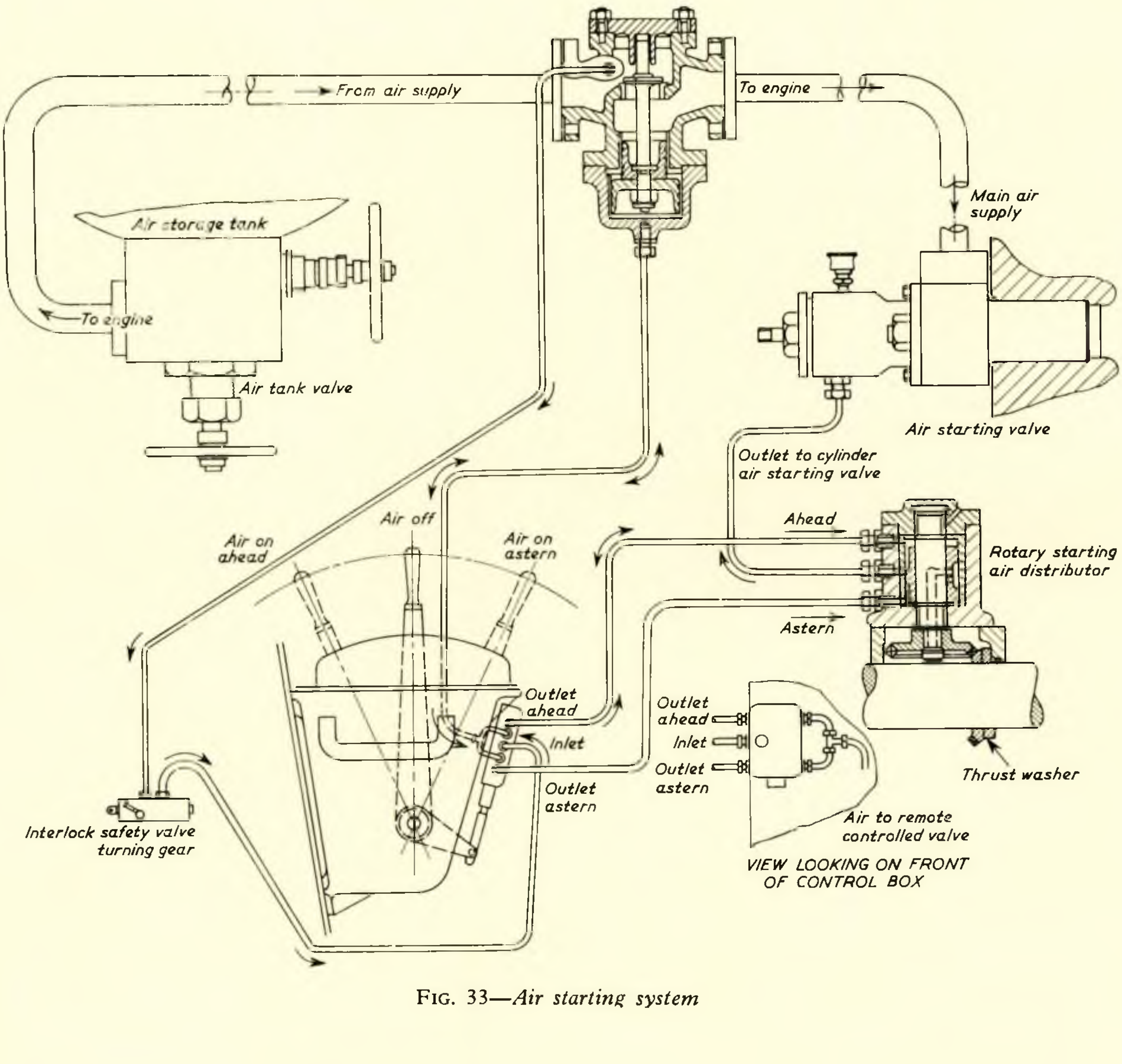


FIG. 33—*Air starting system*

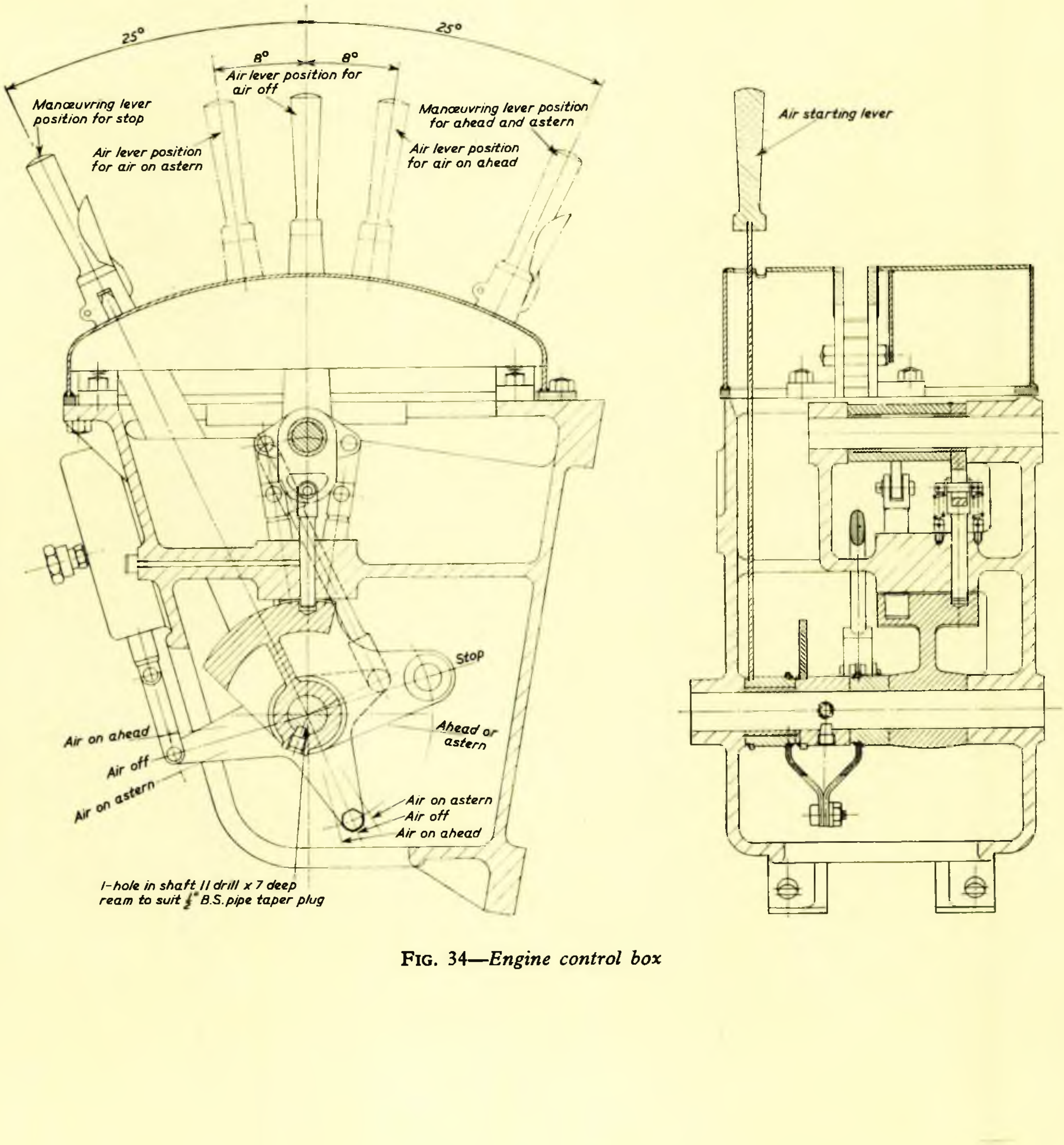


FIG. 34—*Engine control box*

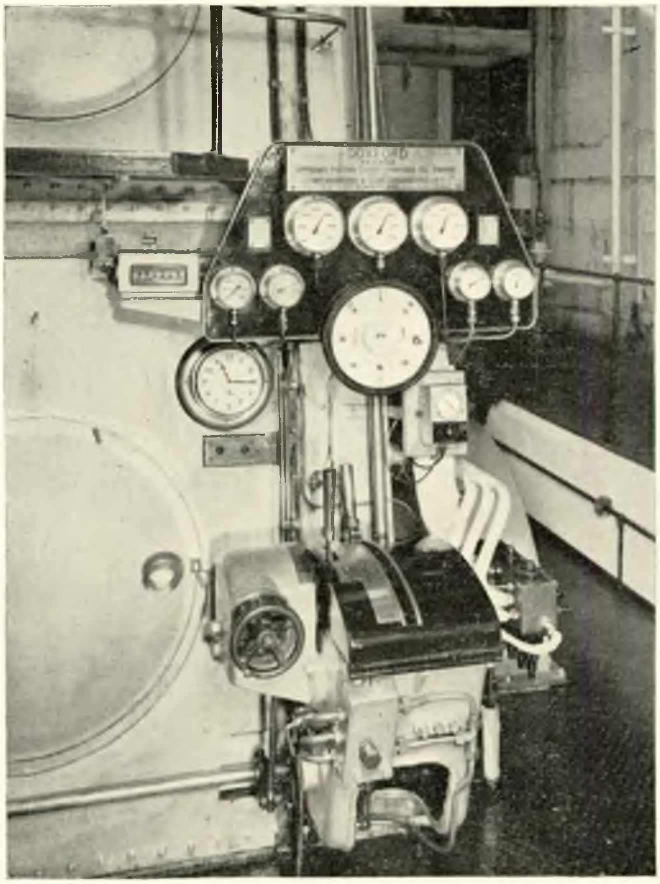


FIG. 35—*Engine control station*

wheel is then set to position 15 and the timing valve control lever is moved to notch 3. The engine will now be firing on fuel and the air starting lever is moved back to the neutral position. The fuel handwheel and the control lever can then be advanced to whatever power and speed is required from the engine. The engine can be started in a few seconds and manœuvred from ahead to astern in under 15 seconds on the test bed but three or four minutes are required to manœuvre an engine from full ahead to astern running at sea due to the speed of the ship.

A gauge board is provided at the control station to indicate the pressures of the fuel, water and lubricating systems and to show the engine speed as illustrated in Fig. 35.

Figs. 36 and 37 show drawings of the corresponding parts of the old and new engines from which it will be seen that the parts of the new engine are much simpler, shorter and lighter, yet stiffer than those of the old engine.

Scavenge Pumps (Fig. 38)

The normally aspirated engines are fitted with two small scavenge pumps per cylinder incorporated in the crosshead guides of each cylinder unit and these are driven from the side crossheads. The suction and delivery valves of each pump are mounted on the back face of the pump, i.e. at the back of the engine, and are readily accessible behind their respective covers. These scavenge pump valves are Monel metal plate valves, spring loaded onto their seats. The scavenge pumps are of small diameter with simple plate pistons and they deliver through internal trunking into the air space in the entablature. The whole arrangement is easily accessible and simple as shown in Fig. 34 and for the propulsion of the standard cargo ship requiring under 5,000 b.h.p. it is the author's opinion that the normally aspirated engine is the simplest and cheapest form of propulsion.

No scavenge pumps are fitted on the turbocharged engines.

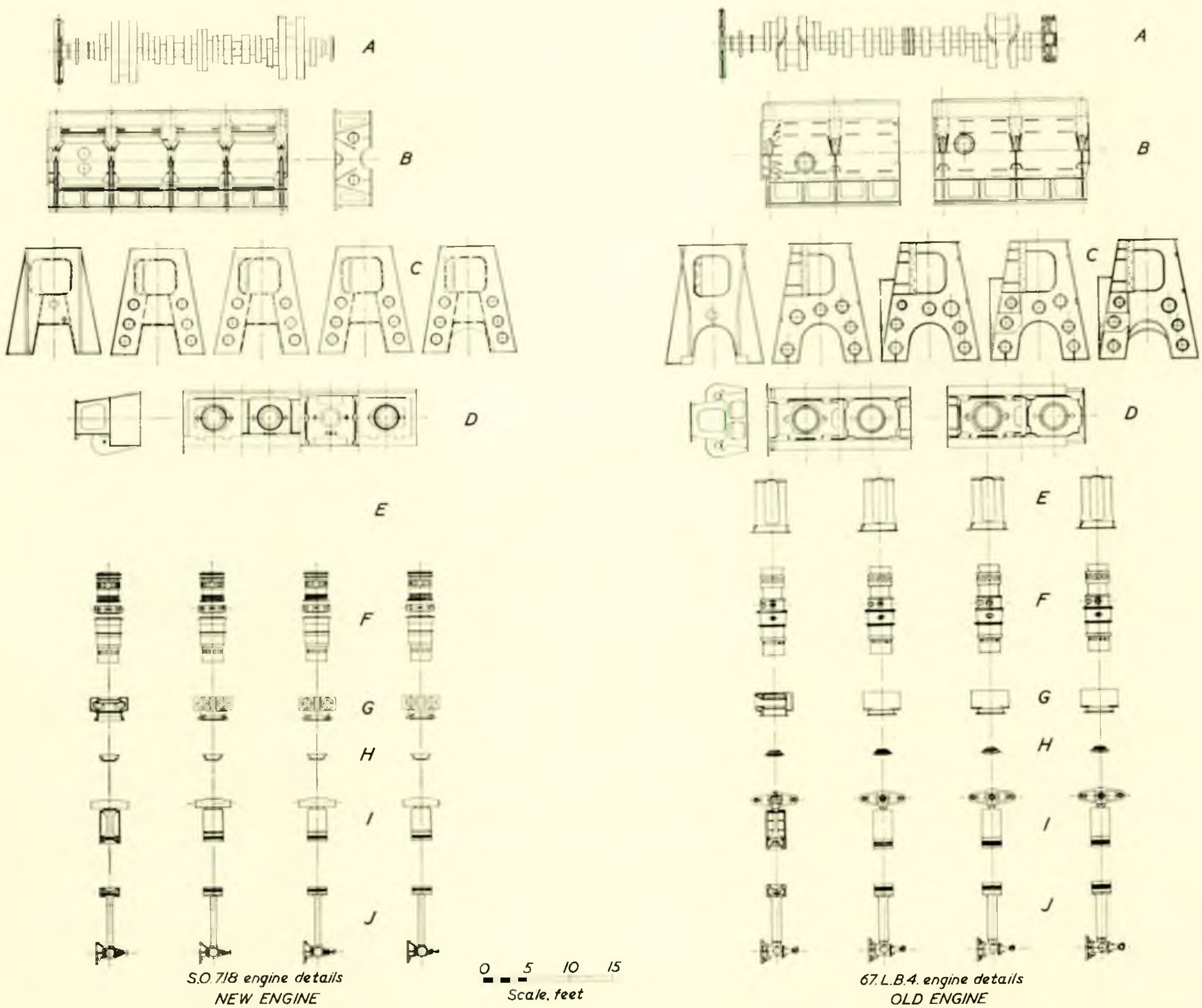
Turbocharged Engines (*Figs.* 39 and 41)

The engine is turbocharged on the impulse system by turboblowers of Brown Boveri or Brush manufacture.

The turbochargers are mounted on the upper platform level and are connected directly to the exhaust pipes from the various cylinders. The blower side of each turboblower draws its air from the engine room through a filter-silencer and delivers at 6 to 8lb./sq. in. pressure through intercoolers into a large receiver. This receiver is situated at middle platform level and in fact the upper face of the rectangular receiver is the platform on which one walks around the engine. This receiver is coupled directly to the engine entablature and this arrangement provides easy access around the engine without accessibility being impaired by large pipes. The exhausts from the exhaust gas turbines are connected into a large manifold which may convey the gases into a silencer or to an exhaust steam boiler if so required.

Two arrangements of exhaust turboblowers have been tried out on six cylinder engines one arrangement being with two turboblowers one for each set of three cylinders, and the other with three turboblowers one for each pair of cylinders. With the latter system the exhaust turbines can be mounted closer to the cylinders and the pipes from the exhaust belts are straight, short, and of small volume without any bends so that the exhaust impulse from the cylinders is preserved without loss of energy for doing work in the turbines. With this arrangement smaller exhaust turbines can be used than when two turbines are used and there is a greater safety margin should one turboblower break down. The arrangement is neater and simpler and more accessible and there is little difference in price or in performance between the two arrangements. These arrangements are shown in Figs. 39 and 41 and photographs 40 and 42 and if there are only two turbines for the engine it is desirable to have a motor driven fan mounted in the corner of the middle platform and so arranged that the fan draws its air from the engine room and delivers into the rectangular air receiver. This fan is not normally required for the operation of the engine should one blower break down though it is an additional safety and enables the engine to manœuvre better, and increases the speed range which is possible under emergency conditions. On four cylinder engines there are similar alternatives of one or two turboblowers the former being the neater and less costly arrangement though the latter gives a slightly better performance and greater power. A motor driven fan is provided with both these arrangements to enable the engine to operate and bring the ship into port should one turbocharger break down. These arrangements for the four cylinder engine are shown in Figs. 43 and 44 and are for a firing order of 1-2-4-3 which is the phasing of the cylinders which gives the best balance for the four cylinder engine and in fact the out of balance forces and couples on this engine are almost zero, both for primary and secondary couples. The two turbines for the four cylinder engine are each provided with two inlets, one inlet from each turbine taking the gases from numbers 1 and 4 cylinders respectively. The gases from numbers 2 and 3 cylinders are divided, half going into each turbine. Thus, the nozzles of each turbine are divided into two thirds and one third area each. Grids are provided on all turbine inlets for stopping pieces of broken piston ring or other metal parts from going into the turbine, and there is also a pocket with a detachable cover for collecting these pieces of metal and preventing them from being beaten against the grids until they are small enough to pass through. The five cylinder engine is the most difficult to turbocharge but it is the intention to have two turbines with divided inlets and to so arrange that the exhaust from the centre cylinder is divided into each turbine. Either Brown Boveri or Brush turbochargers can be employed and there is little difference in performance. The former have ball and roller bearings whereas the latter have plain bearings which can be lubricated from the engine lubrication system, adequate filters being supplied in duplicate with change-over valves.

With the above arrangement it has been found possible to turbocharge the engine and achieve satisfactory running at all loads and speeds without the assistance of a scavenging air



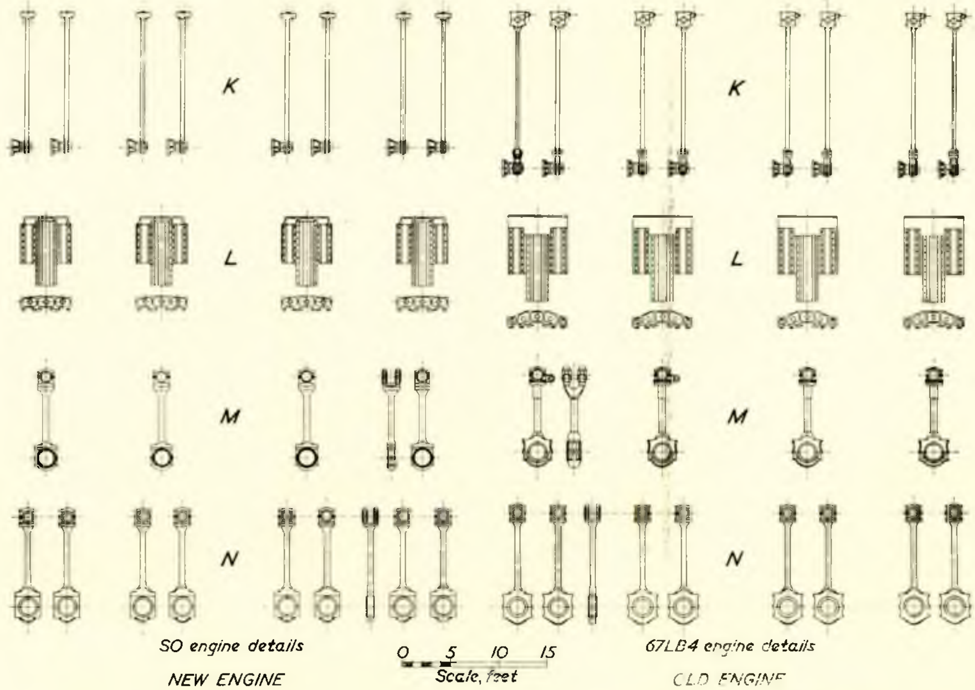
NEW ENGINE

A. Crankshaft with turning wheel.
B. Bedplate in one section.
C. Columns.
D. Entablature in one section.
E. No comparable part.
F. Cylinder liners in two sections complete with cooling jacket. Combustion chamber is a separate part.
G. Exhaust belts with two outlets.
H. Diaphragm glands.
I. Top piston assembly comprising transverse beam piston rod, piston head and piston skirt.
J. Lower piston assembly comprising: piston head, piston rod, crosshead guide shoe and oil service bracket.

OLD ENGINE

A. Crankshaft with turning wheel and detuner.
B. Bedplate in two sections.
C. Columns.
D. Entablature in two sections.
E. Top guides.
F. Cylinder liners complete with cooling jackets.
G. Exhaust belts, single outlet.
H. Diaphragm glands.
I. Top piston assembly, comprising transverse beam piston rod, skirt and piston head.
J. Lower piston assembly, comprising piston head, piston rod, crosshead guide shoe and piston cooling water bracket.

FIG. 36—*Comparison of crankshafts, beds, columns, etc. of the old and new engines*



NEW ENGINE	OLD ENGINE
K. Side rods and guide shoes with crossheads.	K. Side rods and guide shoes with crossheads.
L. Crosshead guides front view and sectional plan.	L. Crosshead guides front view and sectional plan.
M. Centre crank rods with crosshead bearings and bottom end bearings.	M. Centre crank rods with crosshead bearings, bottom end bearing integral part of rod.
N. Side crank rods with crosshead and bottom end bearings.	N. Side crank rods with crosshead bearings bottom end bearing integral part of rod.

FIG. 37—*Comparison of the running parts of the old and new engines*

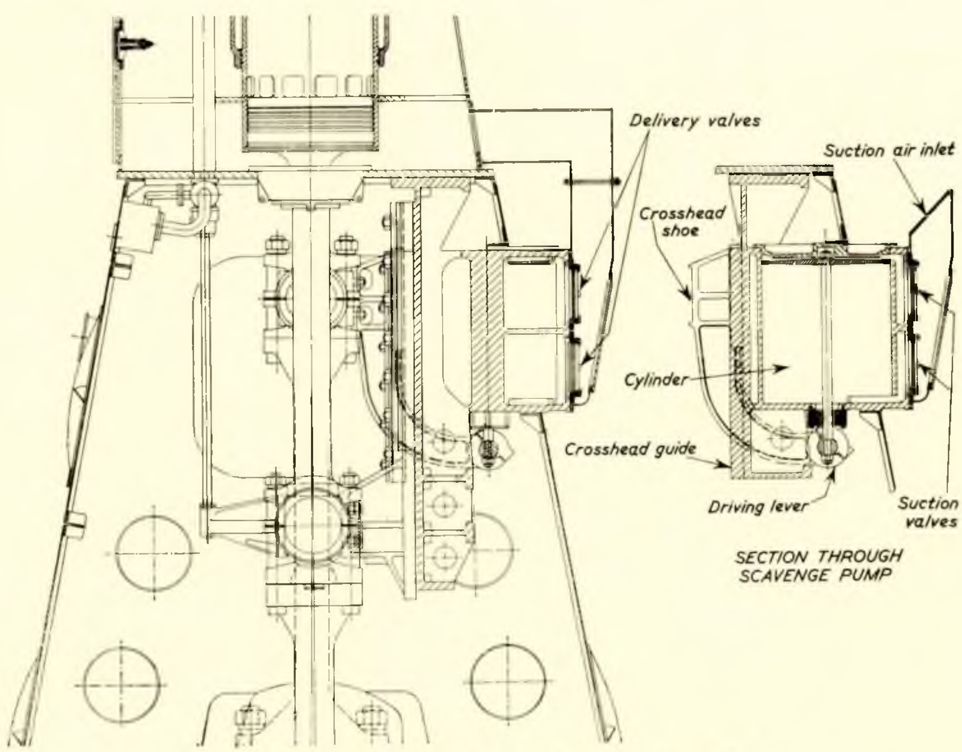


FIG. 38—*Arrangement of the scavenge pumps and drive*

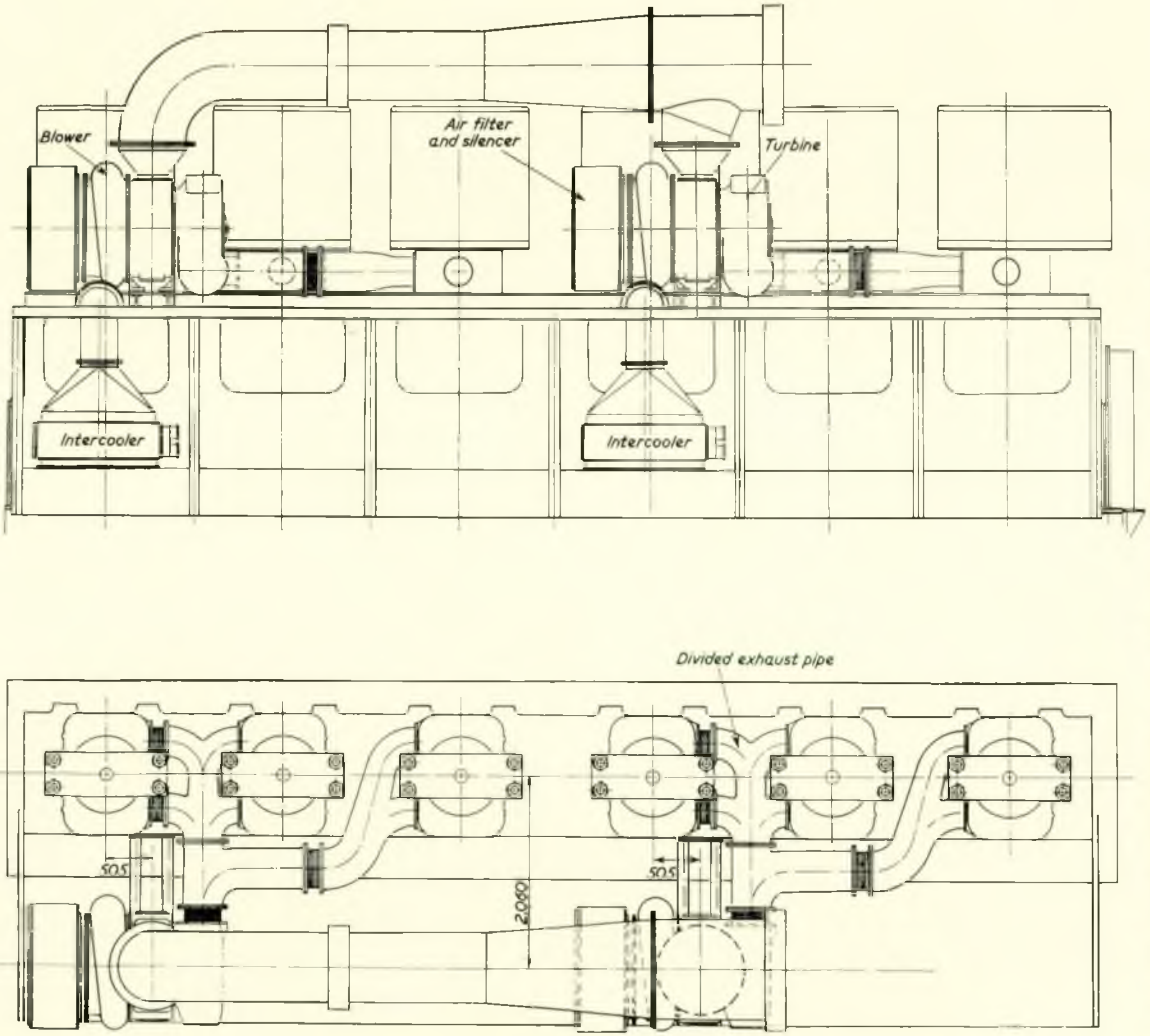


FIG. 39—*Arrangement of the six cylinder engine with two turboblowers*

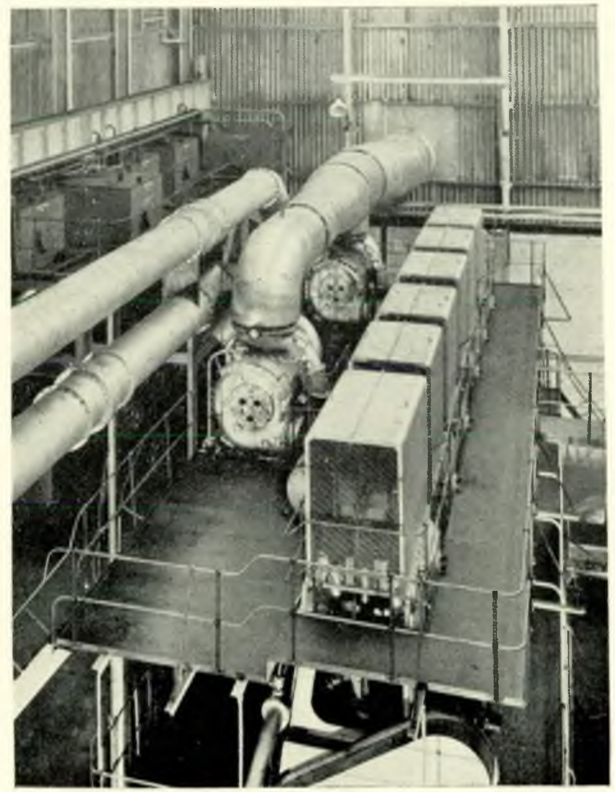


FIG. 40—*Photograph of six cylinder engine with two turboblowers*

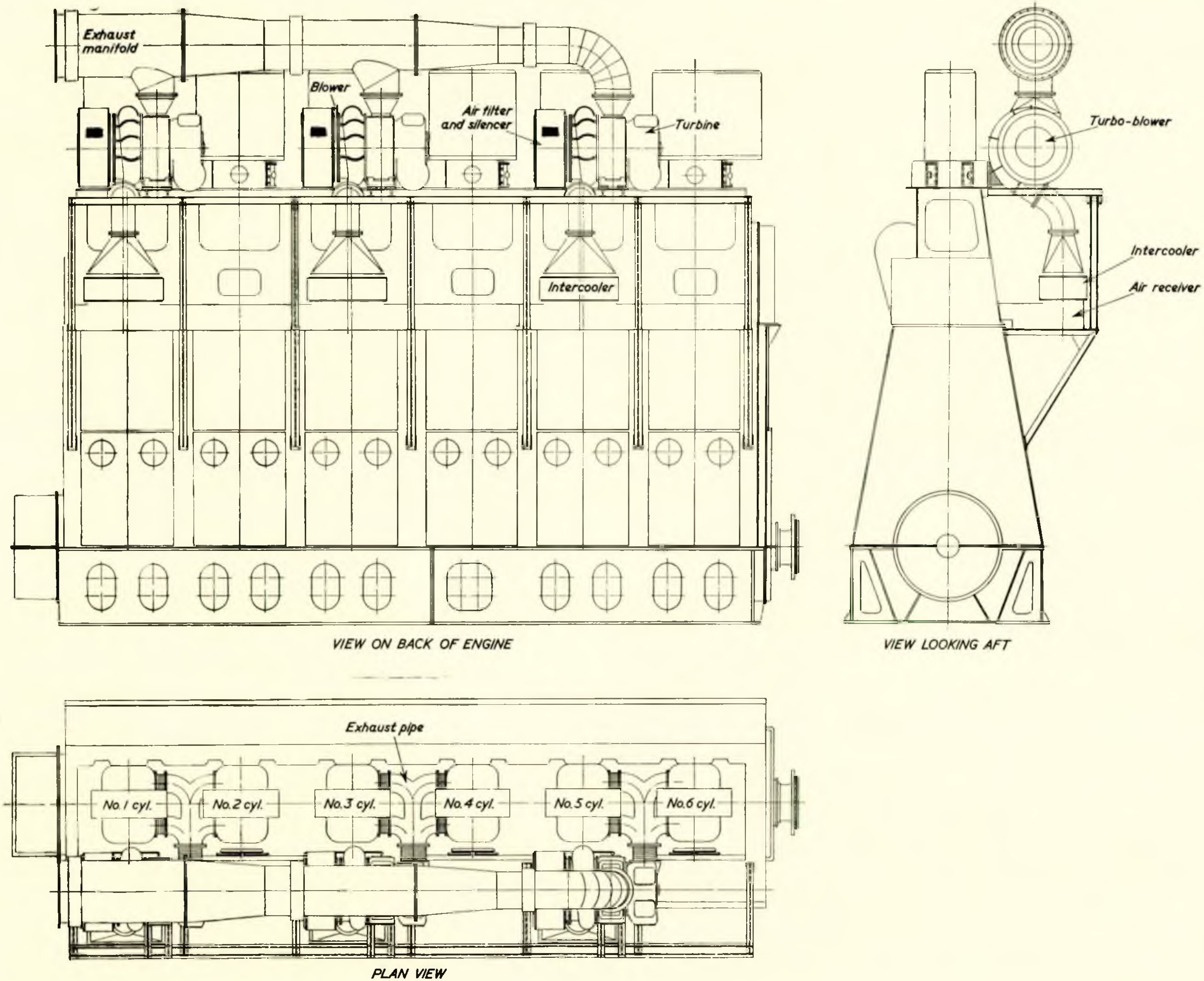


FIG. 41—*Arrangement of the six cylinder engine with three turboblowers*

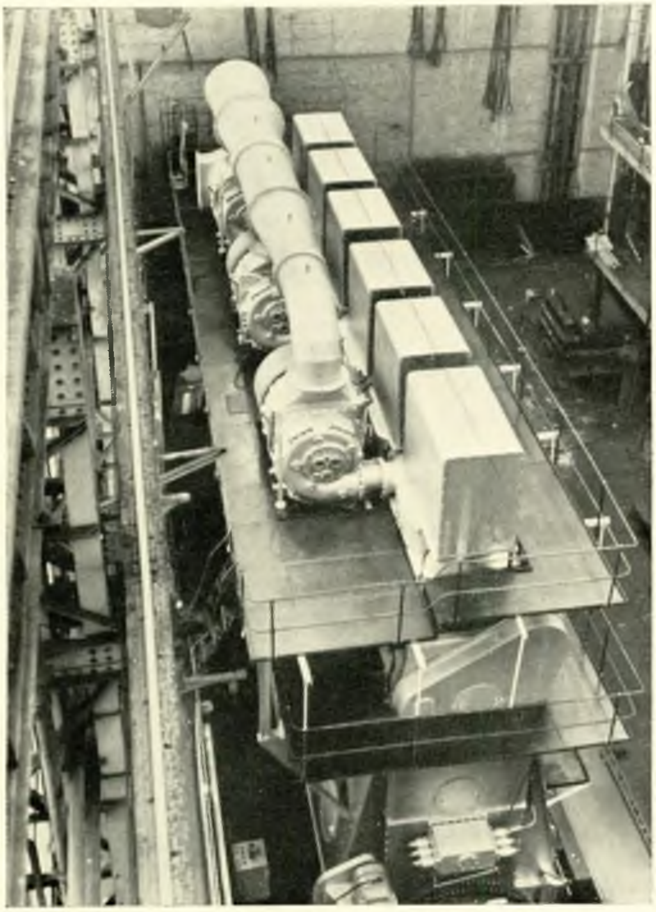


FIG. 42—*Photograph of six cylinder engine with three turboblowers*

pump; although as previously stated a motor driven fan is provided as an extra safeguard and to provide satisfactory running at slow speeds in the case of a breakdown. On the six cylinder engine should one turboblower break down out of the three turboblower arrangement then the engine will carry a power of over 6,000 b.h.p. at 90 r.p.m. which is sufficient to bring the ship home at about three quarters speed and the same conditions apply in the case of the two turbine arrangement should one of these break down when, without the assistance of the motor driven fan, one turboblower will give a power of about 5,000 b.h.p. at 75 r.p.m. Under slow speed conditions the engine will normally operate with both turboblower arrangements down to 20 r.p.m. but should one turbine be out of action the minimum speed of operation is about 25 r.p.m. and the engines with the three turboblower arrangement will pick up speed from slow running to half speed and to full speed without any assistance. With the two turboblower arrangement, however, and with one of these out of operation it is sometimes difficult for the engine to pick up speed from slow running at 25 r.p.m. (but it will do so from 30 r.p.m.) and it may be necessary to stop and re-start quickly under these conditions. There is however no difficulty whatever in starting on any occasion in either the ahead or the astern direction. The engine will perform satisfactorily astern though it is desirable not to exceed a power of 7,000 b.h.p. at 90 r.p.m. for more than a few minutes, due to the difference of timing between the ahead and astern positions. The astern power is thus reduced as otherwise some blowback will occur through the scavenge ports. This astern power of 7,000 b.h.p. at 90 r.p.m. is in fact as much as the propellers of most ships will take without causing undue vibration to the ship. It used to be an objection raised against the old Doxford injection system that it would not permit more than half engine power on the one fuel valve which was used in astern running, but when the new injection system was fitted and full power astern became possible all attempts made to utilize this power were unsuccessful since the ships began to vibrate excessively at over 100 r.p.m. This was the case with both Doxford built ships and with those built by other shipbuilders with either cargo ships or tankers with amidship or aft installations. There is thus no question of the astern power being insufficient on the turbocharged engines.

In order to ensure satisfactory performance on the turbocharged engine, the side cranks, that is those which are connected to the exhaust pistons, are given a lead of 9 deg. so that the exhaust port opening is 75 deg. before top centre and the closing is at 57 deg. after top centre. The scavenge ports are open for 42 deg. of the crank angle on either side of top centre of the lower piston; thus the lead of the opening of the exhaust ports is 33 deg. ahead of the opening of the scavenge ports which provides adequate time and area of port opening for the exhaust gases to be discharged from the cylinder before the air ports open and the exhaust ports are open for a period of only 15 deg. after the air ports

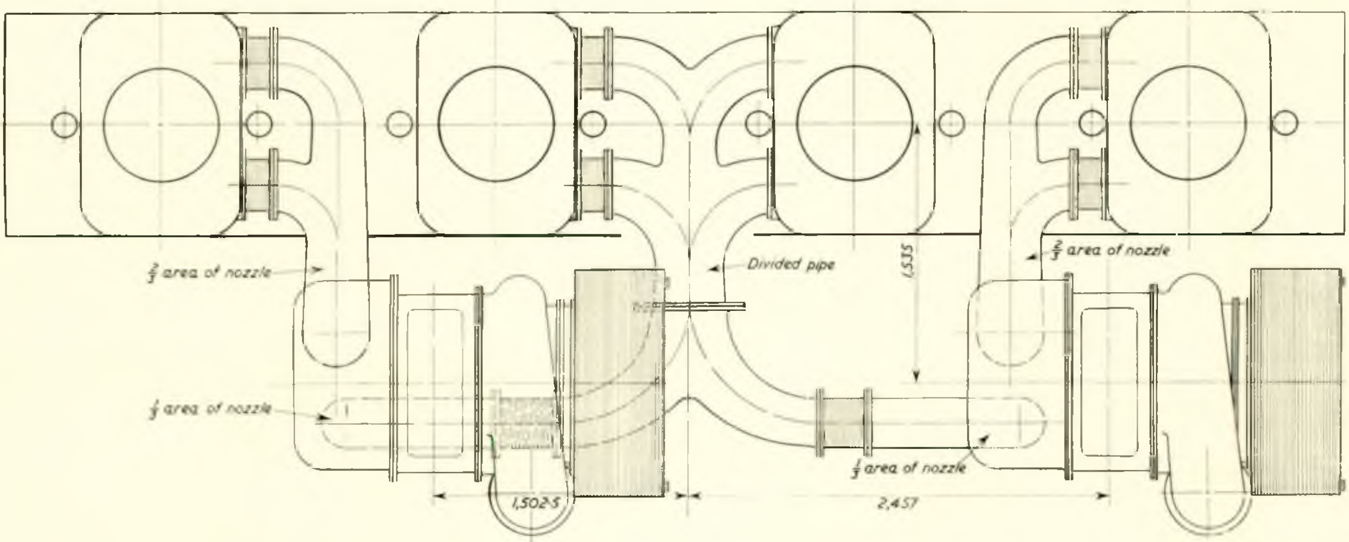


FIG. 43—*Arrangement of the four cylinder engine with two turboblowers*

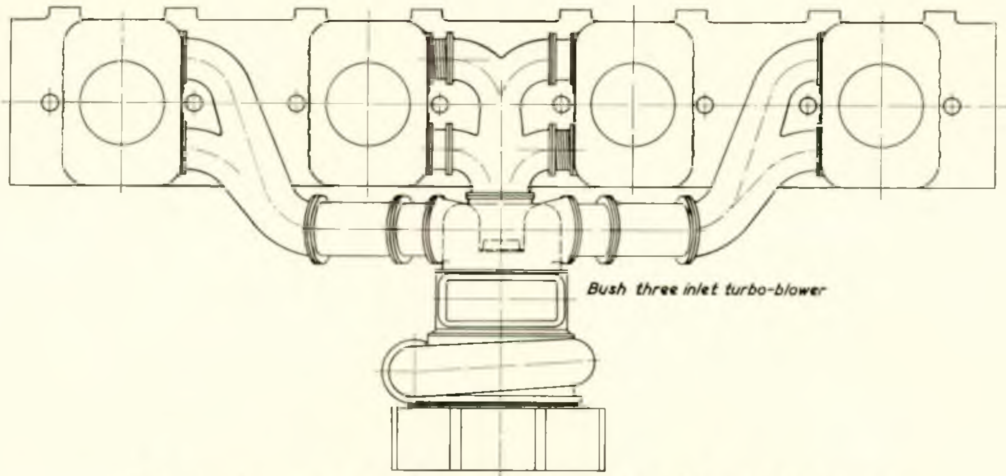


FIG. 44—*Arrangement of the four cylinder engine with one turboblower*

have closed and this delay of the ports closing is relatively small so that there is little loss of scavenging air from the engine cylinders during this period and the air trapped in the cylinders is available for the combustion of fuel. These port openings are shown in Fig. 45, and Fig. 46 shows a low pressure diagram taken from the engine cylinder to illustrate the pressure in the cylinder when the exhaust ports open, when the scavenge ports open, and when the scavenge and exhaust ports close. Fig. 47 shows a diagram taken from the exhaust pipe to illustrate the pressure of the exhaust impulse available for work in the turbine.

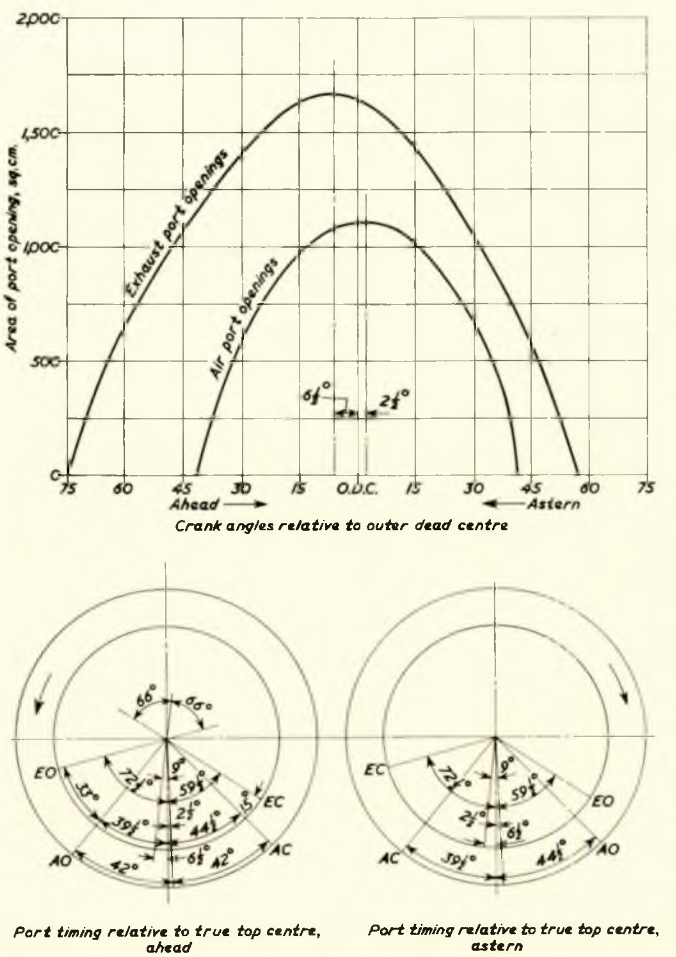


FIG. 45—*Port opening diagram of the turbocharged engine*

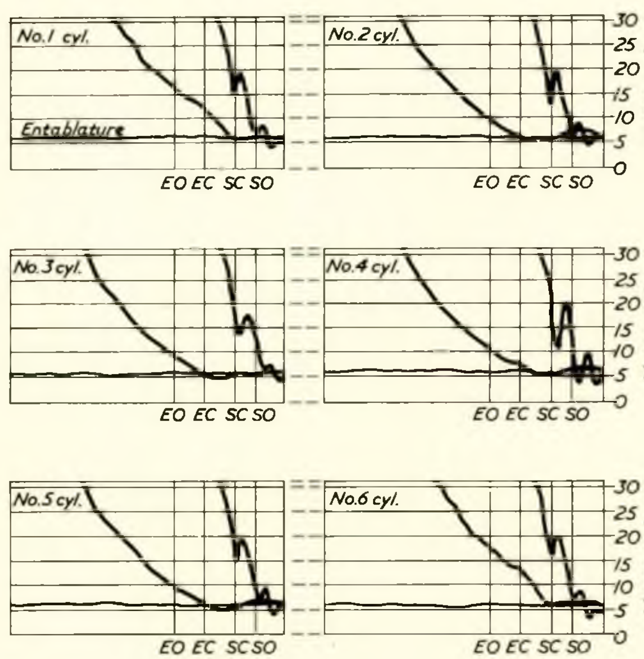


FIG. 46—*Low pressure diagram from the turbocharged engine*

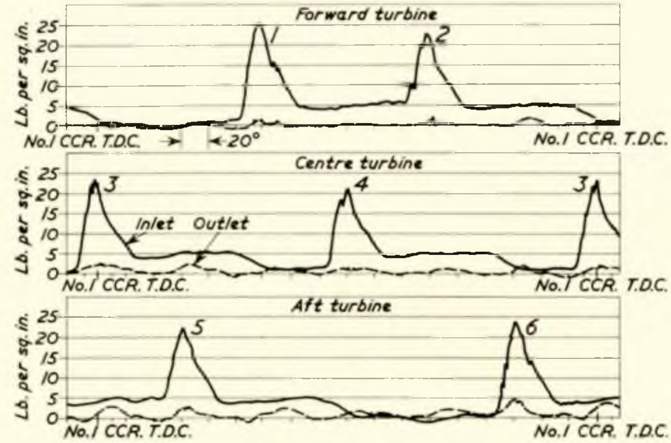


FIG. 47—*Low pressure diagrams from the exhaust pipes showing the exhaust pulses*

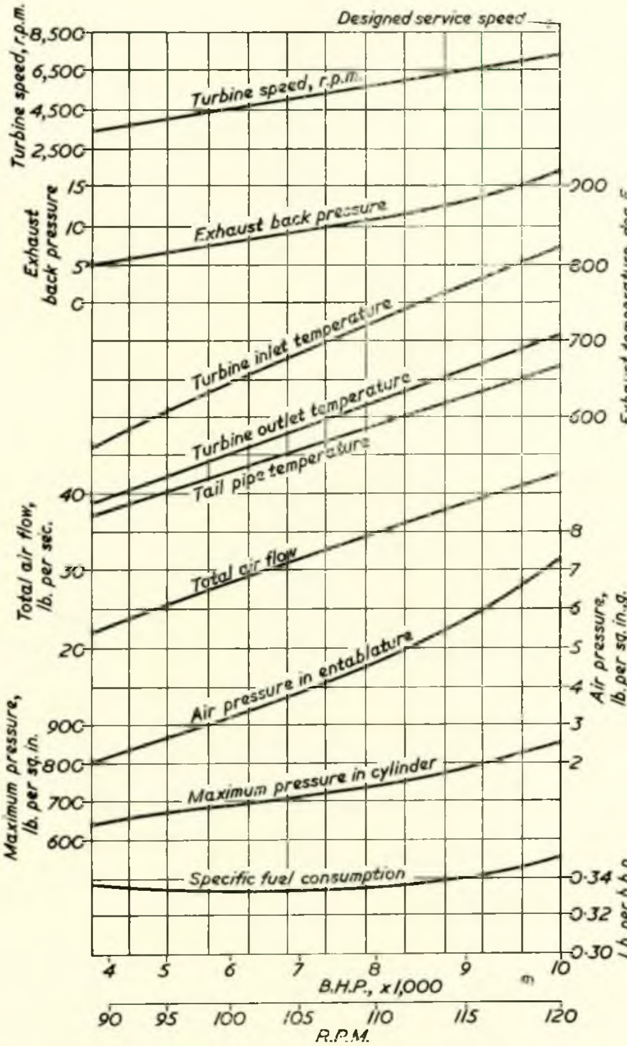


FIG. 48—*Test curves for the two turboblower arrangement*

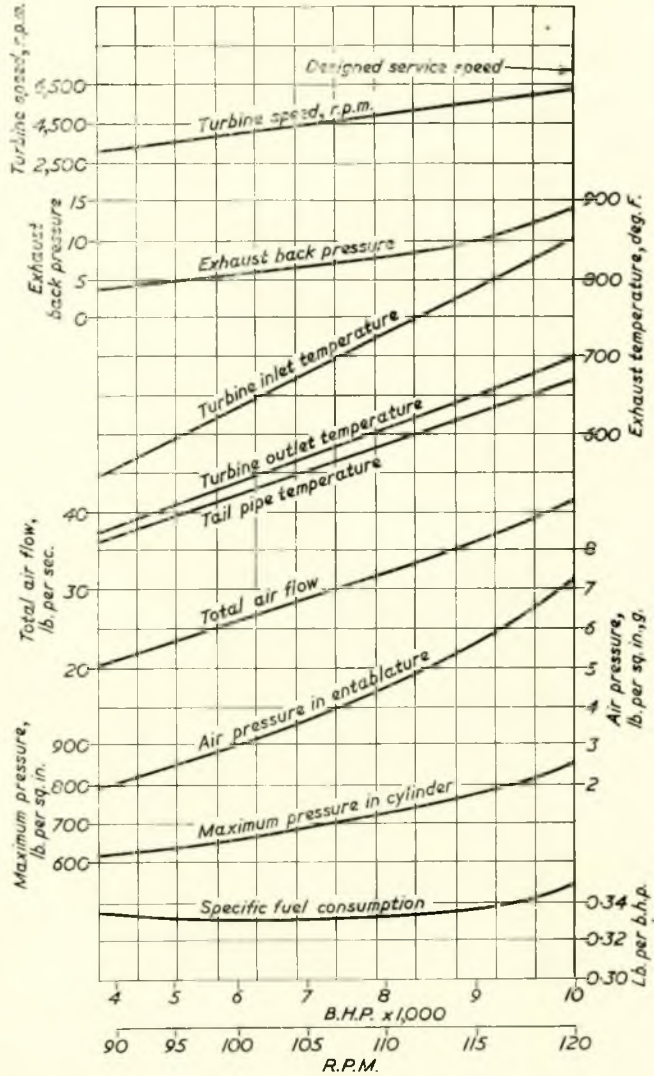


FIG. 49—*Test curves for the three turboblower arrangement*

Test Results

The first PT6 engine has been subjected to a long series of trials to determine the characteristics of the engine and turboblowers and to determine the load and fuel consumption at various speeds. These test results are shown in Figs. 48 and 49 for the two turboblower and three turboblower arrangements respectively. It will be seen that the minimum consumption of fuel at three-quarter load was below 0·33lb./b.h.p./hr. and this low consumption is mainly due to the high mechanical efficiency of the turbocharged engine which is as high as 93 per cent due to the elimination of the scavenging air pumps and the supply of all the air required for the engine entirely by the turboblowers.

After these performance tests the engine was subjected to two duration runs of 1,000 hours each during which period high viscosity fuel was used. After certain initial teething troubles had been eliminated the engine ran continuously during the second 1,000 hours and it is now being installed in the m.v. *Montana* which will be having its trials during the month of January 1961 and then proceeding to sea. It is hoped and expected that the engine will maintain the good name of Doxford's and will indeed be a worthy successor to the earlier engine of which there are over 1,500 at sea propelling motor ships and tankers all over the world.

REFERENCE

1) "Recent Developments on Marine Diesels" (Doxford), 1956, Trans: Vol. 68, p. 373.

Discussion

Mr. R. Cook, M.Sc. (Member of Council), in opening the discussion, said that he did not think it was too much to say that the paper read that evening dealt with a milestone in the history of the British marine oil engine. It had been pointed out by the author that it was 27 years since the previous Doxford engine was designed, and since that time he could not personally recall a single new British design of large slow speed marine oil engine, if they excepted the pioneering work of Harland and Wolff on the Burmeister and Wain type engine: a very important exception. The policy of those two principle British engine builders had clearly been to develop a sound basic design over a long period—a policy which, it was well to remember, had been followed with conspicuous success, in another field of internal combustion engine design by Rolls-Royce and Jaguar. The large number of orders that had been placed for the new Doxford engine clearly went to show the favourable impression it had made upon their profession, which was, no doubt, tremendously encouraging to the author and his colleagues. He was sure that everyone present would wish them success.

The paper contained a wealth of information, and it was clear that the author had not only made full use of service experience with the many Doxford engines on the high seas, but had also made shrewd use of the results of researches in the intervening period, a field of engineering to which he had long devoted a good deal of his time and energy.

There were three points in that connexion to which he wished to refer. In the introduction, mention was made of the good scavenging of the large opposed piston engine. Whilst that claim had often been made, it was only recently that quantitative evidence had been forthcoming, and as it amply bore out the author's statement, the meeting might be interested in a few details. Some years ago, an eminent authority on internal combustion engines had suggested to the British Shipbuilding Research Association that the scavenging of two-cycle engines left a good deal to be desired, and strong evidence had been adduced to show that that was certainly the case with high speed engines. A comprehensive research had been initiated and for that purpose, two experimental single-cylinder opposed piston engines of 5in. bore and 15in. combined stroke, had been designed and built to operate with end to end scavenge at 500 r.p.m. They had been fitted in the first instance with cylinder liners which were scaled-down versions of a typical opposed piston marine oil engine liner. One of those engines had been designed to operate with spark ignition on an over-rich hydrocarbon/air mixture to ensure that the whole of the oxygen retained in the cylinder was consumed. Under those conditions the I.M.E.P. was a direct measure of the scavenge efficiency.

Some of the results obtained with that engine were shown in Fig. 50. The top left-hand curve, marked "ideal displacement curve", represented the theoretical efficiency obtainable if the fresh incoming air was considered to be separated from the burnt gases by an insulated diaphragm. The lower right-hand curve, marked "ideal mixing curve", represented the theoretical scavenge efficiency if one imagined each small volume of air as it was introduced to be immediately dispersed amongst the burnt gases and the excess rejected to exhaust. Of course, ideal mixing resulted in a loss of efficiency, but it was possible to obtain even worse results than that, because it was possible for the air to take the shortest path from the inlet ports to the exhaust ports and to by-pass the whole process. That was the sort of thing that it was believed was more likely to occur with other forms of scavenge such as cross-scavenge and loop-scavenge. The actual results obtained from that experimental engine over a range of speeds were shown by the middle curve, and it would be seen that something between ideal displacement and ideal mixing had been obtained. Looking at it closely it would be found that at a scavenge ratio of 1·3 the scavenge efficiency was no less than 94 per cent, and that even when the scavenge ratio fell to 1·1 there was still 90 per cent scavenge efficiency. That, by the way, represented mixing to the extent of about 40 per cent. It could also be observed that those results were obtained over a range of speeds varying by about 2½ to 1, and had it been inferred from that, that the scavenge

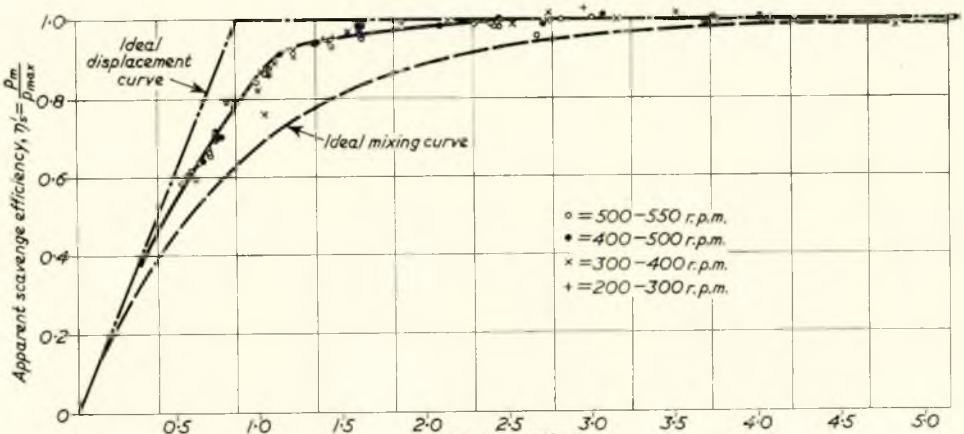


Fig. 50—*Calor-gas engine. Variation of apparent scavenge efficiency with scavenge ratio*

efficiency was also independent of cylinder size. For that reason it was believed that the middle curve represented the scavenge efficiency which would be obtained in an engine such as the Doxford.

One of the outstanding features in the new design was the adoption of large diameter short length bearings, with the accruing advantages of reduced engine length and increased crankshaft stiffness. Some years ago an experimental investiga-nical improvements made there could be no doubt that there was now a better Doxford engine than ever before. Rather than follow the trend of generalities and superlatives, with ambiguity prevailing over fact, he proposed to give one definite and typical illustration which would depict the progress made and the full significance and value of what the author had presented technically. The example dealt with engines available in 1958-1960:

Table I.

Year	Type	Length	Weight (tons)	Cost	BHP Continuous rating
1958	70LBDS6	61′ 1″	510	Basis	9,000
1960	67PT6	45′ 6″	375	—£44,000	10,000

tion at Pametrada had demonstrated the increased performance and the increased safety factor to be obtained by shortening turbine bearings, and that lesson had been applied in subsequent British Marine turbine designs. More recently, the British Shipbuilding Research Association had been investigating the use of shorter length bearings under the cyclic loading conditions encountered in marine oil engines. Those experiments were bing carried out on a specially designed test rig with a 3in. diameter pin. Recent results from this investigation suggested that the use of shorter length bearings was also advantageous under cyclic loading conditions. The author might therefore have built into his new engine an even larger safety factor and better performance than he imagined.

He next wished to refer to the use of pressed-in bobbin pieces in the lightening holes of the centre crankpin. That was ingenious and should be effective. He found himself wondering whether that device had been employed previously in marine practice, and would be interested to hear the author's comments on that point. The author would doubtless remember a very comprehensive investigation into the strength of shrink fits in built-up crankshafts which had been made for the British Shipbuilding Research Association some years ago, the results of which had been published*. That work showed that plastic flow started at a fit allowance equal to the yield stress divided by the modulus of the material. For the type of steel used in marine crankshafts that meant a fit allowance of about 1·2 thousandths of an in./in. diameter. It had been shown in experiments with full size crankpin and web assemblies of no less than 21in. bore, that fits in excess of that led to loss of fit and interface pressure. It had been concluded that the large fits used in marine engineering practice (and he believed the author's firm used a fit of two thousandths of an in./in. diameter) were one cause of the bell mouthing to which the author referred in his paper. It had also been concluded that there was much to be said for limiting the shrink fit to values at which little or no overstraining occurred. However, the author's reluctance to depart from a practice which had served well in the past was understandable and more so in view of the reduction of diameter observed in the lightening holes of the centre crankpin. Incidentally, the investigation referred to was at present being extended to a study of the effects of cyclic loading on specially designed machines.

In conclusion, he wished to congratulate the author on his excellent paper, and in parenthesis members of the Institute could congratulate themselves on having secured a valuable addition to their Transactions.

Mr. R. Atkinson, D.S.C., R.D., B.Sc. (Member) said that the fact that it had been possible to have this paper presented was a matter of satisfaction to his company and an occasion of great significance to the author. If he might be allowed to say so, it was also a matter of significance to himself and hence he would take this opportunity to add his congratulations to that of the previous speaker.

After hearing the author's factual account of the tech-

* Proc. I.Mech.E. 1954. Vol. 168, No. 32.

The advantages of the P range were clear cut; a saving in length of 14ft. 7in., a saving in weight of 135 tons, an increase in horsepower of 1,000 and a reduction in price of about £44,000. These accomplished facts were headline news for shipowners and shipbuilders both at home and abroad.

Regarding P range engines it was his intention to restrict his comments to engines other than the 67P type basically dealt with by the author.

Early results of the 67PT6 engine were so encouraging that it was decided to standardize on a range of engine sizes based entirely on that engine and covering a range of powers both above and below that of the 67P type. The entire P range as it existed at present was depicted in Table II.

The 560 mm. bore engine, the design of which was now well advanced, was to be offered as a turbocharged engine only and was primarily intended to satisfy the needs of the comparatively high speed, shallow draft, vessel such as the Mediterranean fruit trader. Nevertheless its low weight and small size relative to power, even at 120 r.p.m., will make it a very competitive engine for lower speed ships requiring from around 4,000 to 6,000 s.h.p.

In the case of the higher powered engines it was decided that an output of 18,000 b.h.p. would meet practically all likely requirements, but even then the problem of fixing cylinder dimensions could not be regarded as solved since three basic problems still required consideration. These are:

1) Sales require an infinite number of cylinder sizes to satisfy all enquiries.
2) Production requirements call for the minimum number of cylinder sizes in the interests of economy and standardization.
3) On the design side, large engines present vastly increased technical problems such as torque variations, torsional vibrations, optimum turbocharging arrangements and—perhaps most important of all—thermal stress problems in pistons and cylinders.

The final decision obviously was a compromise between these three conflicting interests. Although an 85PT6 engine was now being designed, it might be that for the power range between 10,000 and 18,000 b.h.p., six-cylinder engines would be preferred to those of fewer cylinders. Additional cylinder sizes of 725 mm. and 780 mm. bore enabled this entire range to be covered by six cylinders as was shown in Table III.

These proposals, in addition to the advantages of optimum balance, even torque and optimum turbocharging arrangements, basically satisfied the popular choice of 12,000, 14,000, 16,000 and 18,000 b.h.p. better than the continental competitor, as suggested in Table IV.

The comparison is approximate and excludes consideration of external or built-in thrust block or weight of water and oil.

In addition to satisfying the fundamental technical points given in Table IV, it was noteworthy that the new Doxford engine had the advantage in both length and weight in addition to its accustomed heritage of good accessibility and fewer cylinders for an equivalent horsepower. For the owners this meant more cubic, more deadweight and, since a large marine

TABLE II

67PT TURBOCHARGED ENGINES

Type	Continuous service rating			Principal dimensions			Weight (tons)
	BHP	RPM	BMEP p.s.i.	Length	Breadth	Height	
67PT4	5,800 6,080 6,640	105 110 120	121	34′ 10″	12′ 2″	31′ 2″	245
67PT5	7,250 7,600 8,300	105 110 120	121	41′ 3″	12′ 2″	31′ 2″	320
67PT6	8,700 9,120 10,000	105 110 120	121	45′ 6″	12′ 2″	31′ 2″	375

85PT TURBOCHARGED ENGINES

Type	Continuous service rating			Principal dimensions			Weight (tons)
	BHP	RPM	BMEP p.s.i.	Length	Breadth	Height	
85PT4	10,400 11,400 12,000	100 110 115	119	40′ 0″	14′ 6″	37′ 0″	480
85PT5	13,000 14,300 15,000	100 110 115	119	46′ 6″	14′ 6″	37′ 0″	570
85PT6	15,600 17,200 18,000	100 110 115	119	55′ 6″	14′ 6″	37′ 0″	670

67PN NORMALLY ASPIRATED ENGINES

Type	Continuous service rating			Principal dimensions			Weight (tons)
	BHP	RPM	BMEP p.s.i.	Length	Breadth	Height	
67PN3	3,210 3,360 3,675	105 110 120	89	28′ 7″	11′ 10″	31′ 2″	180
67PN4	4,280 4,480 4,900	105 110 120	89	34′ 10″	11′ 10″	31′ 2″	225
67PN5	5,350 5,600 6,125	105 110 120	89	41′ 3″	11′ 10″	31′ 2″	290
67PN6	6,420 6,720 7,350	105 110 120	89	45′ 6″	11′ 10″	31′ 2″	340

56PT TURBOCHARGED ENGINES

Type	Continuous service rating			Principal dimensions			Weight (tons)
	BHP	RPM	BMEP p.s.i.	Length	Breadth	Height	
56PT4	4,120 4,300 4,480 4,640	120 130 140 150	130 126 122 118	29′ 9″	10′ 6″	26′ 0″	160
56PT5	5,150 5,375 5,600 5,825	120 130 140 150	130 126 122 118	35′ 9″	10′ 6″	26′ 0″	195
56PT6	6,180 6,450 6,720 6,990	120 130 140 150	130 126 122 118	41′ 0″	10′ 6″	26′ 0″	227

TABLE III.

Type	Continuous service rating		Principal dimensions			Weight (tons)
	B.H.P.	R.P.M.	Length	Breadth	Height	
67PT6	8,700 9,120 10,000 10,800	105 110 120 130	45′ 6″	12′ 2″	31′ 2″	375
725PT6	10,450 11,500 12,000 12,500	100 110 115 120	48′ 0″	13′ 6″	33′ 6″	470
78PT6	12,600 13,200 14,100 14,700	100 105 112 117	52′ 3″	14′ 0″	35′ 0″	560
85PT6	15,600 17,200 18,000	100 110 115	55′ 6″	14′ 6″	37′ 0″	670

TABLE IV.—A CONTINENTAL COMPETITOR

B.H.P.	No. of cylinders	Length	Weight (tons)
12,000	7	51′ 3″	591
14,000	8	56′ 9″	659
16,000	9	62′ 3″	738
18,000	10	67′ 9″	811

Diesel engine sold at about £550 per ton weight or about £20 per b.h.p. continuous rating, a price advantage of over £55,000 existed in some cases.

For some unknown reason it was not normal to discuss prices in technical papers, but he made no apology for doing so on this occasion (nor any other) as he considered that price in itself was only second to reliability and, as such, the customer had every right to know precisely where he stood. Those were matters of supreme interest to the shipowner and shipbuilder alike.

MR. J. E. CHURCH (Member) said that he wished first of all to add his congratulations to Mr. Jackson on his excellent paper and, what was more important, to cimpliment him upon the outstanding new Doxford engine which had been designed and tested under his leadership. He was very proud and happy to be the Superintendent of the company which had the first of those new engines to be built for a British ship, now running on the test bed at Sunderland, and it followed naturally that he was in agreement with most of the contents of the paper, which described so well the details of this remarkable new engine and the ideas behind it.

After more than twenty years of trouble-free operating experience of Doxford engines, it seemed to him that Mr. Jackson had started off by setting himself the difficult task of designing an improved version of something that was already pretty good. Whilst the reasons for some of the changes made were clearly explained in the paper and understood, it appeared rather doubtful whether the reduced stroke of the upper pistons relative to the combined stroke, was altogether a good idea. Basically, the advantage of the opposed piston design was, other things being equal, to produce double the power from a given cylinder, and the first Doxford engines with equal strokes top and bottom had done so. From 1934 onwards, unequal strokes had been introduced, in which the upper piston contributed about 40 per cent of the power output, still sufficient to justify the complication of the crankshaft and additional working parts. Now that the upper stroke had been reduced to about 33 per cent, was there not a danger that some of that most important advantage would be lost?

The design of the cylinder liner was most interesting and it was hoped that it would prove satisfactory in service, at the higher ratings proposed. But it seemed to him that there was not much wrong with the previous liner that could not have been put right without introducing high pressure joints in the combustion chamber, and he was a little doubtful about a three-part liner which perhaps must give trouble sooner or later. There was, in his view, no advantage in being able to renew part of a cylinder liner at a time, because the cost of removing and replacing in the engine was greater than was justified, unless a full working life of the whole could be expected as a result.

The design of the engine as a whole, however, was so good, particularly inside the crankcases, which appeared to be as near mechanically perfect as possible, that he could not believe that Mr. Jackson could not have produced an efficient and leakproof water cooling running gear for the lower pistons, because the oil cooling of them was considered a backward step. The circulation of the large quantity of cooling oil necessary required the expenditure of an appreciable amount of extra auxiliary power, amounting to about one per cent of the engine output, which largely offset any gain in mechanical efficiency claimed for the new engine and increased the size of the electric generators which had to be installed, which, together with the larger pumps and coolers, added much to the cost of the installation. It was understood that the 850 PT engine was to have water cooled lower pistons, so that if the problem could be solved in that engine it was hoped that water cooling throughout the range would follow fairly soon.

The very much higher power output from these smaller engines was to be achieved by an increase in piston speed and maximum pressure, and it was said that the engine was designed for a mean indicated pressure of 100lb./sq. in. when not supercharged and 135lb./sq. in. if turbocharged, with a possible maximum pressure of 1,000lb./sq. in. Those ratings were a considerable step upward and to date, attempts to achieve them had met with little success. The limiting factor had previously been the ability of the materials used for pistons and combustion chambers to withstand elevated pressure and temperature stresses without eventual fracture in service. Could Mr. Jackson now assure them that those problems had been overcome so that they could confidently run the new engine continuously in service at full power? For his part he could not believe that maximum pressures of 1,000lb./sq. in. should be lightheartedly accepted, and he felt convinced that the aim should be to develop a system of pilot injection, which would start a small quantity of fuel combustion well before T.D.C., followed by the main injection a little later in an attempt to produce a lower, fatter indicator diagram, which would produce the power required with a lower maximum pressure, thereby enabling the crankshaft and running gear and bearings to be reduced in size, complication and cost. Otherwise, the size of the crankshaft and bearings for the 850 mm. bore engine would be quite formidable. Did Mr. Jackson hold out any hope of that as a future development?

So far as the turbocharged version was concerned, he was not altogether satisfied that there was any case for supercharging up to 135lb./sq. in. except for the higher powers, beyond which the scavenge pump engine could not cope, and it seemed that the 560 mm. bore engine would be better if built as a so-called normally aspirated engine, better described as "pump charged". To advance the exhaust crank 9 deg. to cause the exhaust ports to open 75 deg. before bottom centre, in order to produce sufficient turboblower output for high supercharging, seemed to be robbing Peter to pay Paul and the short effectiveness of the power stroke, which resulted by such early power gas release, was surely not the best way of producing power. Also, when going astern, compression would start very late and be low in consequence. Would there not be a risk of failure to start astern from rest as the engine got older and piston ring and liner wear took place?

On the other hand, the pump charged version was most attractive, and Mr. Jackson's arrangement of bracket driven scavenge pumps seemed to be admirable in every way. The first of those engines now on the test bed was looking and behaving very well indeed, and as that was to be rated at 100lb./sq. in. m.i.p., it was probably the simplest of any engine currently available for any power up to 7,350 b.h.p. In fact, it was probable that a 725 P.N.6 engine only 48ft. 0in. long giving 8,900 b.h.p. would be more satisfactory in service than any turbocharged engine of the same power and well worth considering. In passing he would say that the single scavenge pump on top of the upper piston, as used on the single-cylinder prototype described by Mr. Jackson, had decided possibilities.

It was probable that experience would show that the scavenge pump engine for medium powers would prove to be the most reliable and cheapest to run, and Doxford's new version of that would prove to be one of the best available for many years to come. He had been running the previous Doxford engines for many years at a cost of less than ten shillings per b.h.p. per annum for all repairs, replacement parts and surveys, and had every reason to believe that the new type being built would prove as economical, or better, in every way. In fact, it was thought that the P.N. engines described in the paper would prove even more important in the future than even Mr. Jackson realized, because in the face of industrial expansion ashore, coupled with the five day week and social advancement, seagoing for engineers was daily becoming less attractive, and very soon many ships would have to sail with four engineer officers at the most, in order to make the best use of dwindling available manpower. A simple, reliable, yet efficient Diesel engine would be an absolute necessity, and the easy maintenance of the scavenge pump engines described in the paper would make it one of the only types, if not the only one, capable of operating under such coming conditions. Perhaps Mr. Jackson should have entitled his paper more simply "The Future Marine Oil Engine".

MR. R. MUNTON, B.Sc. (Member) said that he did not think too many of them could add their congratulations to Mr. Jackson and to his company for something that had taken a very great deal of courage, i.e. the almost complete re-designing of a major engine. It was always very easy to criticize someone else's work, particularly without responsibilty for the work, and this applied to the design of engine presented in the paper. He did not think that, in general, the time for criticizing the engine was the present and indeed hoped the time would never arise, but he would look forward with interest to see the longer term performance results of the engine and the modifications that arose from service experience. He wished, however, to endorse Mr. Church's remarks about using lubricating oil for piston cooling, because he believed it to have been an incorrect decision and one that he hoped would go out of the window.

He wished to refer to page 203 of the paper, as he was not clear as to the meaning of what was written concerning the lubrication of the crankpins and journals. The paper said: "On the new engine the lubricating oil is supplied through the bedplate to the bottom of the main bearing housings and then through a peripheral groove which connects with a diagonally drilled hole through the crankwebs to the side crankpins." That seemed quite clear, but then it went on to say: "The centre of the lower half of the main bearing shell is not grooved . . ." He could not follow that part and so he hoped to have an explanation of it from the author. The apparent reason for that was the loss of lubricating oil pressure due to bearing wear in the previous engine where oil was fed in from the top of the bearing; but in the present engine if the groove went right round the bearing—and that was the point about which he was not clear—he could not see why there should not be the same loss of oil pressure with the same amount of bearing wear; but Mr. Jackson, no doubt, had a good explanation for that.

Finally, he would have liked to see in the paper—and he hoped Mr. Jackson would provide it—some information on the results of the running of the engine to date. He knew that some of it had been given elsewhere, but he thought it would be of great value to the TRANSACTIONS of the Institute if the particulars of the running results up to date were included with the paper.

MR. S. ARCHER, M.Sc. (Member) said that once an engine was designed and established in service the possibilities of improvement in its performance and reliabilty were severely restricted; in fact, the crucial importance of design decisions was sometimes brought home forcibly after years of previously successful service, when a crop of failures suddenly occurred on numbers of engines all built over a relatively short period. That, of course, was especially the case with fatigue. A new design therefore presented the opportunity of selecting and retaining those features of construction previously found successful and eliminating those which might have given trouble. The new Doxford series showed clear evidence, he thought, of that process, and Mr. Jackson and his colleagues were to be congratulated on the way they had heeded the lessons of the past whilst at the same time adopting the best features of modern practice within the framework of their own conceptions.

There were a few queries that he wished to ask Mr. Jackson to comment on, and one or two points of fact. On page 197, under torsional vibration, reference was made to stiffening of the crankshaft as having resulted in a higher natural frequency, but from his calculations it seemed that with the six-cylinder engine, 670 mm. bore, it had meant the substitution of a 9th order critical speed near the service r.p.m., which were admittedly higher than in some of the older engines, instead of the 7th order. Of course, with the node near the centre of the crankshaft, the 9th order vector summation was more severe than the 7th. The harmonic coefficient was smaller but the net result was that it was about as severe as the 7th, and possibly a little more severe, so that the mere increase in frequency had not necessarily eased the vibratory conditions, as implied on page 200.

On page 202, the reduction in the stroke of the upper piston, already referred to by Mr. Church, had contributed its share to the overlap of side pins and journals, since if the same ratio to the combined stroke had been maintained as in the old engine, that is, 890 mm. stroke instead of the 720 mm. now used, the overlap would have been less by one-third, so that that was a further advantage of the shorter upper stroke.

With regard to the bobbin pieces in the centre crankpin bores already referred to, he wished to know from Mr. Jackson whether he was carrying out any actual tests to demonstrate how effective the bobbin pieces were. He supposed that, provided the stress conditions in the web bore remained elastic, then theoretically, that should be an improvement, but of course, if there was plastic flow at the outer ends in consequence of the greater rigidity conferred by the "dumb bells" he might not get as good distribution as he thought; and, not clear as to the meaning of what was written concerning of the advantage he claimed from a balancing point of view; so he thought that the author ought to be quite convinced that he was getting a worthwhile advantage there.

The paper, as far as he could see, had not given any details of the crosshead guide construction. Was that bolted to the A-frames as in the older engine, or was some other form of connexion used? It might be that with the greater fore and aft stiffness of the engine, some of the troubles that had been experienced with broken guide bolts in the older engines might be eliminated, and he hoped that would be so.

The transverse beam seemed to be rather a key component in the new engine, inasmuch as it was a departure from previous practice, and although Mr. Jackson had stated that the transverse beam was supported by a spherical cast-iron pad, was that in fact capable of accommodating differences of wear-down between the side rod bearings, or was it an independent adjustment which could only be made with the engine stationary?

On the question of the performance of the non-spherical bearings, of course, he quite appreciated that they had performed very well so far, but presumably they had not yet been really proved at sea; and he felt that was the only true proving ground for departures from past practice.

With regard to the bedplate, it was stated that double transverse girders were only provided on six-cylinder engines. He would have thought that that would depend more on cylinder bore than on the number of cylinders.

With regard to the crankshaft, was the integral thrust abandoned, and, if so, why? That had been a Doxford feature.

On page 213 the paper dealt with the starting air system. A minimum of twelve starts was an old-established classification society requirement, but probably that would be considered rather meagre rations, and he wished to know whether Mr. Jackson had any figures showing how many starts in fact, were possible with the starting air pressure and capacity stated, *viz.* 600lb./sq. in. and 140 cu. ft.

On page 217, where the paper dealt with the rather important business of manœuvring and, in particular, the time to obtain astern running from full ahead, three to four minutes did seem rather lengthy, and in view of current discussions, he wondered whether the Doxford designers might consider the use of some braking system, such as the North Eastern Marine device, or something like that. What percentage of the ahead power was obtainable astern with the normally aspirated engine? Presumably the figure of 7,000 b.h.p. at 90 r.p.m. referred to the PT6 engine?

The author had stated that blowback into the scavenge main could happen under certain conditions of astern running owing to the difference of timing between ahead and astern operation, if the power was too great. This conjured up visions of fires and explosions! Surely that might call for some automatic limitation of astern power to be incorporated in the engine, since they were all too familiar with those unfortunate occurrences at the present time.

Rear-Admiral A. L. P. Mark-Wardlaw (Member) said that he wished first of all to pass on the congratulations of his friend Mr. van Asperen, who much regretted his inability to be present, and he knew that the author would miss his contribution.

He had read the author's paper with great interest and wished to make a few comments upon it. It seemed to him that, so far as cylinder output at revolutions between 112 and 170 was concerned, the proposed Doxford engines compared very favourably with the competitors—for example, B. and W., Sulzer, M.A.N., Götaverken, Fiat and Mitsubishi. That also applied to the new Stork/Werkspoor range.

With regard to the performance curves on page 224, he had been able to make some comparisons. The very satisfactory low fuel consumptions were favourably comparable with the competitors. He assumed that the fuel used was the normal 18,000 B.T.U. calorific value. It was comparable with the Sulzer RD 90 and the Stork/Werkspoor 850 mm. bore of 2,100 b.h.p. per cylinder at 115 r.p.m.

In the range of engines competitive to Doxford one noticed cylinder output of 600-700 b.h.p. at higher revolutions around 150 to 170, and it would have been of much interest to know whether the opposed piston engine was suitable for those conditions.

In conclusion, he wished to know whether there was possibly a misprint on page 224 for the weight of the 78PT6, which he thought should be perhaps 560 instead 360.

Mr. G. Victory (Member) said that the majority of uncommitted marine engineers, who had had experience with various types of motor vessels, would agree that for over twenty years the Doxford Marine Oil Engines were the finest in the world. What a twist of fate it had been that Doxfords, who first claimed some forty years ago, that their engine would run on almost any fuel, were among the first to be brought face to face with the stern reality of crankshaft corrosion when the use of residual fuels became commonplace. He thought they might do well to consider the prime factors which had conspired to bring about the end of the Doxford engine as they knew it. Primarily it had been brought about by the growing demand for engines of a power unthought of when the present engine was designed, aided to a great extent by the increase in the sulphur content of the fuels at present in use. In his opinion, the change in the operational methods of engine room staffs, which was an aftermath of the war, influenced perhaps by the introduction of the officers' pool, had also played its part. He was not saying that present day engineers were not as good as in pre-war days, but they did not stay with the same ship long enough, and in some cases did not perform enough of their own maintenance to know what was going on in the engine. It seemed probable also that the shortage of engineers and the competition for their serivces had led to rapid changes in personnel, to the detriment of engine room maintenance. That had affected the Doxford more than some other types, as it had always been an engineer's engine which gave its best when in good hands. The existing non-diaphragm type Doxford engine still had many good years' service in store, but it was evident from Mr. Jackson's paper that its days were numbered, and he felt that they should show proper regret at its passing. However, that was in the past, and they were looking forward to the bright future of another thoroughbred from the same stable. When they were informed that it was over ten years since the seed of the new engine began to germinate in Mr. Jackson's mind they could see the amount of painstaking care and the continuity of effort which had been required to produce an achievement of that magnitude.

Whilst proceeding with the primary purpose of designing the engine for powers in the 20,000 h.p. range, Mr. Jackson had also locked, bolted and barred the ways by which any "gremlins" could have got in. There were very few points that he would care to raise in discussion as he found himself in agreement with nearly all the changes which had been made; however, there were two or three.

On several occasions the Institute had heard discussions on the relative advantages of oil and water piston cooling systems. Depending on which system was employed in a particular engine they heard either that all was well with oil cooling and all wrong with water, or *vice versa.* This was the first time they had heard the eulogies of both systems extolled at the same time, and he wished to congratulate Mr. Jackson on the excellent way in which he had remained balanced on the wall. However was Mr. Jackson's balancing act necessary? From the arguments given on page 209 he was sure, that other things being equal, water was the better coolant, and he felt that Mr. Jackson would agree with that.

That brought him to the question: Was oil cooling of the lower piston really vital in a diaphragm engine where the gases and corrosion elements could not get into the crankcase? Surely if there were any water leakage—and if the design were right that would be a very rare occurrence—it was then only a matter for separation as no acid would be formed, and with no acid there could be no corrosion. The adoption of oil cooling involved acceptance of lower operating temperatures, when the trend elsewhere was to increase the temperatures in water cooled systems even to the extent of pressurizing for cylinders and cylinder heads.

Even if they accepted the argument that oil cooling was required for the lower pistons, were telescopic pipes the best method of supplying it? He did not like to see change for change's sake, and he thought that a system of articulated pipes, as in the existing engine and a number of Continental engines, was perfectly satisfactory and less liable to initiate a hammering effect.

On page 208 Mr. Jackson mentioned that snifting valves were provided on the stand pipes, and there in Fig. 23 was their old friend the pet cock, now called an air vent. Had they ever known any pet cock give reliable service over a period of years? He had not, and neither apparently had Mr. Jackson, who showed a shut-off cock which allowed a defective pet cock to be put out of action presumably with a view to changing it. Perhaps it would be changed, perhaps not, but meantime the cooling system hammered like the hobs of hell. Did the telescopic system have enough advantages to outweigh its disadvantages?

The new crankshaft had been discussed before, and it avoided any possibilities of the weaknesses dealt with in the recent paper. The advantages of recessed fillets had not been convincingly demonstrated in Dr. Weines' paper given before the Institute in 1956. What had been established was, that a large fillet made for strength and if an adequate external fillet could not be incorporated, then some improvement could be effected by adopting a larger fillet at the expense of a moderate recess in the web. Obviously there was an optimum recess for a given thickness of web where the increased strength obtained by using a larger fillet was not nullified by any weakening of the web. Although no details were given he knew that that recess was relatively small, but he felt that the wording on Fig. 7 which said "small partial recessed fillets" should have made it clear that it was the recesses and not the fillets which were small.

On the question of balance, surely the reversal in the balancing characteristics had been dictated by the adoption of the diaphragm with its skirtless piston, which, if primary balance of the reciprocating forces was to be maintained, resulted in the adoption of a positive abortion of a piston rod as shown in Fig. 1. It appeared to be suffering from elephantitis and because it was out of proportion it offended the eye. He was glad that Mr. Jackson had got rid of it, just as he was glad that he had not offered them a rectangular scavenge pump, though he agreed that his objections to that would have been mostly aesthetic.

Looking into the future, unlike Mr. Church he felt that the 1,000lb. maximum pressure was right, as he had been with an engine in 1940 of over 800lb. maximum pressure, and he felt that with reasonable development they should be approaching the 1,000lb./sq. in. mark.

He also felt that the separate scavenge pump would be replaced by blowers, even in the smaller ratings. The scavenge pump was not without its faults. He hoped that they might see a stronger effort made to eliminate camshafts and their operating gear. The new Doxford engine had lost one camshaft. Could it now do without the other by utilizing compression operated fuel injection? That system had given Mr. Jackson excellent results some time ago, but was liable to faults which he felt might be overcome in future development. If the camshaft was not required to operate the timing valves he felt that means for operating the lubricators, starting air distributors and indicator gear drive could be easily evolved.

Finally, he thought that when the shouting and the tumult died they might see the lower piston water cooled again, and by some system other than telescopic pipes.

As they had no doubt realized by that time, he was a bit conservative, and had a strong liking for the old Doxford engine. However, changes in engine size, fuel usage, and technical progress demanded that some old standards must pass away. He was glad that Mr. Jackson had made accessibility and ease of maintenance a prime consideration in his new design, and he felt they should congratulate him on retaining the smoothness, balance and many of the essential characteristics of the familiar Doxford engine.

Mr. A. Norris (Member) said that in the introduction to the paper Mr. Jackson had mentioned the easy accessibility, a feature which had made the earlier engine designs so popular with operating engineers. Although not mentioned in the text he presumed that provision had been made for hydraulic or pneumatic power manipulation of the large nuts on piston rods and bearings, as was being arranged by other makers on newly designed engines. The correct pre-tensioning which could be obtained by those means was considered to be particularly important for the side connecting rod crankpin bearings, where a number of breakdowns in earlier engines had been attributed to incorrect tightening of the nuts and to bolt heads not being hard home due to locating dowels not being correctly re-engaged.

The question of blowback on astern running had been previously referred to by Mr. Archer, and on page 222 of the paper a reference was made to blowback through the scavenge ports when operating at 7,000 h.p. astern. It would have been interesting to learn if blowback commenced at any lower astern powers, and if, with worn piston rings and liners, entablature fouling was expected to be more rapid than in earlier engines. The large access doors shown in Fig. 28 would make the unpopular job of cleaning that space relatively easy, but if fouling extended into the air receiver space cleaning would be difficult in view of the restricted height shown in Fig. 41. Sub-division of the entablature, with a separate supply to each cylinder through non-return valves from the common air receiver (as used in other engine designs) would probably reduce both blowback and fouling but could interfere with the engine balance. Would Mr. Jackson comment on that?

He would agree with the comment on the astern power which was required. In steam turbine engined vessels it was normal practice to specify the astern power requirements as 60 per cent of the ahead power when two astern turbines were fitted, or about 40 per cent ahead power when only one astern turbine was fitted. There was no reason why a Diesel engined ship should be required to produce more than that, and on three recent sea trials of such ships the astern power recorded had been 40 per cent for two ships and 70 per cent for the third ship.

His company always included a crash astern manœuvre in acceptance sea trials on the grounds that, although such a movement was seldom required, it might be ordered at any time during the life of a ship. On page 217 it was stated that some three to four minutes would be necessary to manœuvre the new engine from full ahead ship speed condition to astern running. Mr. Archer had commented on that aspect. He assumed that the time was considered satisfactory, on the grounds that the stopping effect on the ship of rotating the propeller astern was not initially pronounced. Nevertheless it was possible to stop and reverse other makes of engines much faster.

As a matter of interest, he had examined the records for a recent class of nine vessels fitted with turbocharged engines of 8,000/8,750 b.h.p. where that movement was carried out from an ahead speed of about 15 knots and where the times recorded had been as shown in the following table:

With regard to the starting procedure, as given on page 213, he wished to suggest that in his opinion it was not a desirable arrangement, as operators would be inclined to shut off both air vessels at sea, and a master stop valve should, in his opinion, be inserted between the air vessel and the starting control valve. One air vessel could then be left open to make air immediately available if an emergency order were received at sea, and would thus avoid the watch-keeping engineer having to hurry to a remote corner of the engine room to open up an air vessel and enable stopping air to be available for the engine. That master valve would also restrict the amount

TABLE V.—ENGINE TYPES

	Opposed piston		Single acting	Doxford 650 bore	Doxford 670 bore
Full ahead to full astern	A	B	C	D	E
	m s	m s	m s	m s	m s
Telegraph order "Full astern"	0 0	0 0	0 0	0 0	0 0
Shaft stopped turning	0 21	0 56	0 16	4 3	1 35
Shaft commenced turning astern	0 30	1 0	0 18	4 3	1 38
Shaft revolving at 70 r.p.m.	1 6	1 35	2 0	5 10	1 58
Vessel stopped in water	4 9	5 34	7 7	9 15	6 23
Full astern to full ahead					
Telegraph order full ahead	0 0	0 0	0 0	0 0	0 0
Shaft stopped turning	0 10	0 8	0 9	0 21	—
Shaft commenced turning ahead	0 12	0 10	0 13	0 21	—
Shaft revolving at 50 r.p.m.	0 17	0 15	0 23	0 53	—

'A' is an average for 3 ships, B and C are averages for two ships respectively.

of air starting line liable to fouling if any leakage should take place from the engine starting valves.

With regard to starting air pressure, it might well be that the retention of 600lb./sq. in. pressure influenced the reversing time mentioned and, as other makers used lower pressures, Mr. Jackson might like to give them his comments on the reasons for retaining that higher pressure for an engine which probably had a greater compression pressure than the earlier designs.

The inclusion of many engine details on the figures answered many questions which would otherwise be asked, but during the demonstration run of the engine, a slight sway had been observed on the upper piston telescopic pipes. In view of the difficulty of keeping similar glands tight on other makes of engine it would be interesting to learn if that had had any adverse effect on the glands and which type of packing had been found most suitable. The absence of top guides made that sway more apparent, and perhaps Mr. Jackson would advise them if those guides were to be re-introduced.

The test curves given on page 224 provided useful reference data. It was presumed that the specific fuel consumptions recorded were for Diesel fuel. To complete the data he hoped Mr. Jackson would be good enough to give the calorific value of the fuel used and, since Diesel fuel was very rarely used on engines of that size, the corresponding rate on high viscosity fuel with a gross calorific value of 18,500 B.t.u./lb. It would also have been of value to have the air temperature at the air cooler outlets during the trial.

In conclusion, he wished to say that the information given to the many superintendents who had been invited to the demonstration of the engine, together with the detail given in the paper, created a most favourable impression, and they looked forward to seeing the new type of engine enhance the reputation of its designers and makers.

MR. C. L. G. WORN said that he wished to add his congratulations to those of earlier speakers to the author for the excellence of his paper.

The contribution of the turbocharger to the increase in specific power of marine engines was of course well known. As his company had been fortunate in obtaining considerable operating experience with a new design of turbocharger during the shop trials of the six cylinder P type engine, it might be of interest to amplify some of the remarks made by the author concerning those turbochargers.

The main design feature worthy of mention was the use of plain white metal lined bearings to support the rotor shaft, with a tilting pad thrust bearing giving end location. Lubrication was either completely self-contained, when an integral gear pump worm-driven from the rotor shaft was fitted or, alternatively, oil could be taken direct from the main engine lubrication system. In the latter case it was desirable to have additional filters fitted.

Fig. 51 showed bearing assemblies taken from a turbo-

FIG. 51—*50 series turbocharger journal and thrust bearing assemblies after 1,200 hr. test*

charger after running on the Doxford engine. That particular turbocharger had engine lubrication and apart from slight scoring the bearings were in perfect condition. During initial tests the bearing frictional losses had caused sluggish starting of the engine, but after reducing diameters to those indicated on the figure, engine behaviour had been the same as with turbochargers fitted with anti-friction bearings. Although it was a well established fact that plain bearings operating under steady and lightly loaded conditions had extremely long life, removable journal sleeves had been provided so that in the event of accidental damage complete replacement of wearing surfaces could be made without the need for re-balancing the rotor shaft.

With the reduced diameter bearings the performance of the turbochargers had been quite satisfactory without the need for re-matching of aerodynamic components, but the noise level had been considered to be unacceptable. That had been due to the characteristic whine of the centrifugal impeller giving a frequency equal to turbocharger speed multiplied by the number of impeller vanes. Measurement of sound level at that frequency, with a microphone at various positions around the turbocharger, showed that a high proportion of the noise was being emitted through the intake filter.

In Fig. 52 on the curve they would see the original and the result, showing the emission from the filter. Provision had been made in the design for the spacing between the annular silencing rings to be altered and by increasing the number of rings from 7 to 11 a reduction of 78 per cent in noise level at the relevant frequency had been achieved.

Fig. 53 gave the overall noise analysis for the same sequence of tests, indicating the great reduction of the aerodynamic noise in relation to the general engine noise

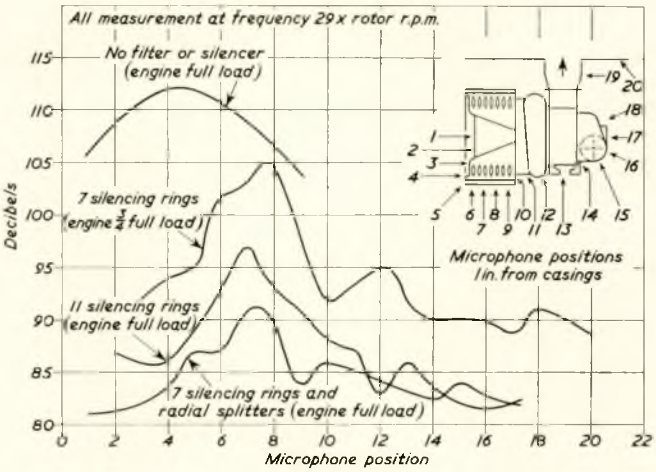


FIG. 52—*Noise level survey 50 series turbocharger on Doxford 6-cylinder P type engine*

level. Although the result with increased number of silencing rings was quite acceptable, the impeller whine being hardly discernible above the general noise level, an alternative solution of combined annular and radial splitters showed a slightly better result. However, that arrangement was more cumbersome during assembly and was difficult to clean and was therefore abandoned in favour of an increased number of annular silencing rings.

Referring to Figs. 48 and 49, it was interesting to note the similar engine performance achieved with the two different turbocharger arrangements. One might conclude from that evidence, that the improved utilization of pulse energy with the short exhaust pipes of the three-blower arrangement was balanced by the reduced partial admission losses in the turbine with the two-blower arrangement. It had to be appreciated, however, that a given frame size of turbocharger operated at optimum efficiency for a certain air mass flow, determined by the impeller/diffuser and nozzle/turbine blade configurations. In the case of his own company's turbocharger, for example, the compressor efficiency when matched for two-blower arrangement showed an improvement of four per cent. To obtain a more exact comparison of the relative merits of the two systems from a performance point of view it would be necessary

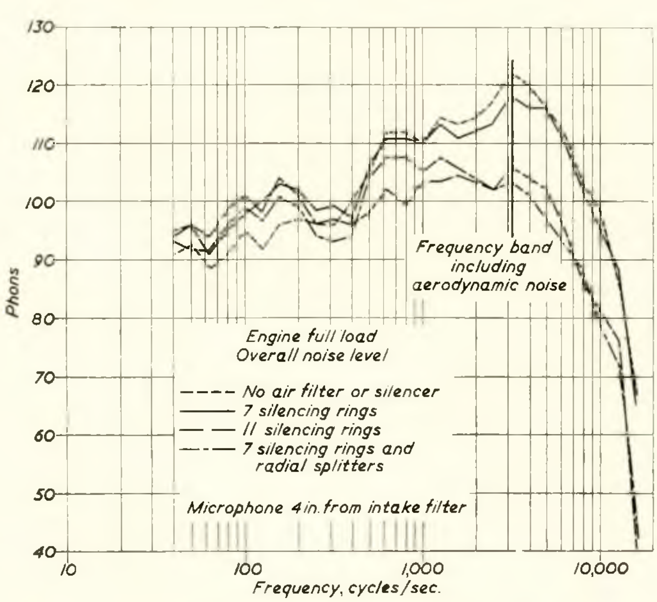


FIG. 53—*Noise measurement 50 series turbocharger on Doxford 6-cylinder P type engine*

to use a smaller turbocharger frame size for the three-blower arrangement.

REAR-ADMIRAL W. G. COWLAND, C.B. (Member) said that he wished to associate himself with the other speakers who had congratulated Mr. Jackson on an extraordinarily good descriptive paper. In his notes he had described the paper as a very good illustrated catalogue of information, and he did not think that much more information could be given in a paper of that sort. He wished, however, to refer to the meagre information given on engine performance. One or two other previous speakers had mentioned it, and he was thinking that one or two questions had not been asked.

On page 217 the author said that the turbo discharge pressure was 6 to 8lb./sq. in. That was a fairly wide range, and he would have thought that the engine performance would be somewhat sensitive to things of that order. Were the horsepower figures given by the author for the various engines for full power at sea? He was interested in the performance of the turboblowers and wondered whether they were of the same make for the two- or three-blower arrangement, on the six-cylinder engine? Were they of comparable efficiency? He would have thought that the two-blower performance would have been a better and cheaper installation but from the curves there did not seem to be much in it. One point he thought worth mentioning was that with the two-blower arrangement they got a pulse in each exhaust turbine every 120 deg. With the three-blower arrangement there was a pulse at zero degrees another at 120 degrees and then a gap of 270 before the next, and that was always a bad thing. That loss probably offset the advantage that the three-blower had of having the very short direct pipes. It was his opinion that in the majority of six-cylinder cases the two-blower engine would be found to be cheaper, with a somewhat better engine performance and requiring less maintenance. The point had been mentioned that blowers did go out of action and that there was an advantage there in the three-blowers, but in his experience they very seldom went out of action, and he would not have liked to send any ship to sea with either two or three blowers without an auxiliary emergency fan. In saying this, he had in the back of his mind the loss of blower performance that sometimes occurred with deposits building up over a period in the turbine nozzles and blades and dirt accumulating in the compressor.

MR. YELLOWLEY (Member) said that he would like to comment on one or two items in the order in which they were presented in the paper.

On page 197 the author dealt with balancing and the ratio of the lower and upper strokes, and he wished to ask which was the major consideration in reducing the stroke of the upper piston—correct balance or the desirability of a large overlap between the main journal and the side rod crankpin—as good overall balance could be obtained with a longer upper piston stroke, which had the advantage of giving more power per cylinder and perhaps better exhaust characteristics.

On page 200 the author dealt with engine ratings for present and future requirements, and stated that the engine was designed for an m.i.p. of 135lb./sq. in. at 135 r.p.m., but went on to say that the engine was also designed for a possible increase in those ratings to an m.i.p. of 150lb./sq. in and a maximum pressure of 1,000lb./sq. in. over the next ten years. It was not clear just what provision was made in the design for the latter very high rating.

On page 202 he had dealt with the crankshaft and its bearings, which was the basis of the whole design and was a notable step forward in the development of the Doxford engine, incorporating, as it did, short cylinder centres, plain bearings and one rigid coupling.

As a measure of their faith in the single rigid coupling design, that feature had been incorporated in a 9,500 b.h.p. N.E.M.-Doxford turbocharged engine having six cylinders of 700 mm. bore fitted in the tanker *Plumleaf* which had now been in service for five months, and had given satisfactory service.

With regard to the main journals with a very small length/diameter ratio, he wished to ask if the author could give any information regarding wear down and alignment during the extended shop tests.

On page 204 the author described the changes in the design of the bedplate, which now incorporated double transverse girders carrying the main bearing housing; but he himself still felt that a single plate girder with a cast steel bearing housing was to be preferred.

On page 206 the connecting rods were described, and the author was to be congratulated on the successful operation during extended shop tests of the centre top end assembly. He wished to ask whether the author could give his opinion as to whether the new method of direct lubrication on the segmental pad had contributed most to that achievement.

With regard to the method of securing the side rods to the side rod crosshead, this could be the subject of a debate in itself, and it might be that the devil they knew was better than the devil they did not know, and he would say that he still preferred the Doxford differential nut if properly hardened up with a heavy spanner and heavy hammer, and not, as generally instructed, with a light spanner and a light hammer.

In the new design shown in Fig. 17 it was essential that the screw plug be adequately hardened up but it was not clear how that could be done in the restricted space available.

On page 207 reference was made to piston heads, and he would have liked to know the reasoning behind such a radical change from well established designs in which the load was taken through the piston ring belt. Many hours of testing in the single and multi-cylinder engine had no doubt proved the piston design to be satisfactory but he doubted if the rubber rings alone would be adequate to prevent oil leaking from the piston.

The cylinder liner shown in Fig. 25 was an important departure from normal Doxford practice and should be an improvement over earlier designs. As the author was aware, his company had introduced a prototype three-piece cylinder liner some four years ago, which was interchangeable with the standard Doxford liner, and a number were now proving satisfactory in service.

The timing valve injection system, with which they were all now familiar, was described on page 212, and there was only one point he wished to ask the author: did he think that the double timing valve was justified in view of the satisfactory performance of single timing valves of the latest improved design?

Doxfords had been the pioneers in airless injection of fuel and the timing valve system was simple to produce and operate, but it might well be that the jerk pump would ultimately be adopted, together with improved governing arrangements.

The author, in his introduction, had stated that he hoped the Doxford PT engine would form the basis of the production, in the Pallion Works, for the next twenty years and that the idea and object in the development was to reduce length and weight and remedy the defects in the earlier engine. There must, he thought, be a large measure of agreement that he had achieved his object and, as the author stated in his concluding remark, they all hoped that the PT engine would be a worthy successor to the earlier engine.

COMMANDER C. M. HALL (Member) said that he had noticed the excellent fuel consumption figures quoted for both, the early engine and the new P range, and he was sure that Doxfords were to be congratulated on that performance. The point he wished to make was that good advantage could be taken of that engine efficiency by the use of engine-driven pumps, as power taken direct from the engine was surely the most economical method of obtaining it. He recalled the earlier engine with lever driven pumps which had contributed to the economy already mentioned, although he thought Mr. Jackson would be the first to agree that a certain amount of maintenance was required with them, and that the substitution of rotary pumps for reciprocating would be a normal development. He therefore wished to suggest that in the new range it would be an advantage to have built-in facilities for driving modern rotary pumps direct from the main engine. With a direct mechanical drive there was some complication in that the pumps themselves had to be of the uni-directional flow type, and it was not possible to use centrifugal pumps. If, however, a hydraulic drive were used, it meant only driving one lubricating oil pump from the main engine; the remaining pumps could be driven by a hydraulic motor and would be situated conveniently anywhere in the engine room. That practice was in growing use on the Continent, and he suggested that the U.K. was a little behind the times in not using it more extensively.

One further point, with reference to what Mr. Church had already said, was that with high power engines the power required for the engine service pumps grew quite considerably, and besides the actual running economy in using engine driven pumps, as outlined above, a considerable saving in capital cost could be achieved in a new ship.

Correspondence

MR. H. R. ALLAN wrote that he would like to congratulate Mr. Jackson on a frank, comprehensive and profusely illustrated paper on a very interesting engine and that he wished the builders every success with the new design.

The crankshaft was a great improvement. The reduction in cylinder centres and the increase in journal and pin diameters, resulting in a stiffer shaft, had allowed the sphericals to be eliminated which was a great benefit in time and cost in overhauling and would result in lower initial cost of this part of the new engine. The bedplate was improved also inasmuch as it was stiffer with the longitudinal box girders and the deeper section. He would have preferred to have retained the normal thickness of white metal in the bearings, but was in full agreement with the longer connecting rod.

He would have liked Mr. Jackson to expound a little more fully on the design of the centre crosshead bearing; having always understood the crosshead to deflect slightly under load, therefore, the white metal would wear quicker and the load on the lead bronze spherical pad increase. The three point bearing was also more difficult to adjust. He would have preferred the orthodox design of crosshead with a slightly larger diameter of bearing and to eliminate the spherical lead bronze pad.

The piston rod was fastened to the crosshead by a flat palm with four short studs. The piston would move, especially as the cylinder liner wore and the strain would be taken on the studs. Also it appeared to be difficult to harden down the two studs nearest to the guide shoe.

The side rod was screwed into the side crosshead and locked by a screwed plug as shown on Fig. 17. It would appear difficult to test the tightness without removing the side rod top ends and connecting rods. Would Mr. Jackson explain the method of tightening this plug.

The piston heads appeared to have been considerably shortened with the top firing ring much nearer the firing space. Was this likely to cause carbonization and eventual sticking of the ring? The inward springing bearing rings were an improvement on the welded cast steel wearing rings.

The upper piston and transverse beam with the spherical cast iron pad, instead of the guides, was in his opinion a retrograde step. He anticipated fretting and eventual seizure between the surfaces of the pad and the upper piston rod, allowing blow past. He regretted that the upper piston guides had been discarded in the new engine as it had been found that they carried out their designed purpose of centralizing the upper piston, inasmuch as they required adjusting at times and occasionally remetalling.

The liner in three parts was a complication which increased the difficulty and cost of changing and he imagined the initial cost would be more than the old type of liner. The cast steel central portion was a complicated casting and it was most unlikely that when the time came to renew a liner, that any part of the old liner would be utilized, as upper and lower liners would require grinding into the central portion. Would the cost of this work be an economical proposition? With the engine running 2,000 hours, were any cylinder liner wear figures available?

The fuel injection system was a great improvement on the old system and functioned extremely well.

He preferred the turbochargers to be fitted with plain journals and to have their own lubricating system. If lubricated off the main engine system there was a danger of fine particles of carbon causing abrasion in the bearings of the turbocharger.

As Mr. Jackson visualized, in the not too distant future, a further increase in the power rating, necessitating maximum pressures of 1,000lb./sq. in. with an increase of exhaust temperatures, it would appear that a return to water for the lower piston cooling would be necessary, for with the crankcase efficiently sealed off from the products of combustion of fuel with a high sulphur content, and with an efficient arrangement of water cooling pipes, the risk of a small water leak was not the danger it was in the old engine.

Mr. S. Archer, M.Sc. (Member), in a written contribution said that, on a more careful reading of the paper, he would like to raise one or two additional points as follows:

1) On page 200 it was stated that the new turbocharged engine was designed with a rating of 150lb./sq. in. m.i.p. and 1,000lb./sq. in. maximum pressure in view. It was possible that the combustion chamber, liners and pistons might ultimately be capable of sustaining such a rating, but from a classification society point of view the crankshaft scantlings would undoubtedly require reconsideration for such loadings.

2) On page 203 the author stated that the material of the large dogleg forgings was 28/32 ton steel, whilst that of the main webs was 34 ton steel slabs. He pointed out that, at least for engines to be classed with Lloyd's Register of Shipping and on the basis of 900lb./sq. in. maximum pressure and 135lb./sq. in. m.i.p., the minimum tonnage of the dogleg forgings would depend upon the number of cylinders and for example, for the PT6 engine developing 10,800 b.h.p. at 130 r.p.m., the minimum u.t.s. would be 32 tons/sq. in. unless the side crankweb scantlings were suitably increased.

3) As regards the design of the centre crosshead, it seemed a pity that the elimination of the forked connecting rod by means of the rectangular palm could not have been further exploited, to provide a fully supported crosshead pin, thus eliminating deflexion and bearing edge loading. However, it was appreciated that the problem of piston rod connexion without the conventional nut attachment was not easily solved. At the same time the use of a copper-lead segmental pad to take the combustion shock deflexions and thereby protect the white metal bearings, whilst ingenious, seemed to be fraught with certain difficulties of ensuring correct load-sharing. Furthermore, should the white metal bearings wear, or become hammered, the full combustion load might well have to be carried by the centre segment with attendant possibilities of excessively high running temperatures, an undesirable, and possibly even hazardous, feature in a crankcase.

4) Could the author confirm that the bedplate crossgirders would be of double-plate construction throughout the PT series?

5) What was the designed compression pressure and would it still be necessary to use hot water circulation before starting as in the earlier engines?

6) In the table on page 224, should not the weight of the 78 PT6 engine have been 560 tons rather than 360 tons as printed?

He would appreciate the author's comments on the foregoing additional points.

Mr. E. Davies (Member) wrote to say that Mr. Jackson was to be complimented on the very excellent paper he had prepared on the subject of the new P type Doxford engine. In following through the development of this new engine one could not help but be impressed with how Mr. Jackson had succeeded in producing an extremely compact design with very clean appearance and yet maintained the ease of overhaul which had been a feature of the Doxford engines.

The introduction of the double transverse girder for the main bearing support in the bedplate was a most desirable feature, and should tend to eliminate the possibility of any weld failures at this point where considerable trouble had been experienced in the past with the old design. The introduction of stress relieving of either the bearing girders or the whole of the bedplate should also ensure better performance from the welds, although the bearing girders had been stress relieved in the old design for several years without attaining complete freedom from cracked welds.

He thought that it was a pity that the rectangular scavenge pump attached to the top piston did not find favour with certain of Mr. Jackson's clients, as this feature had been used in the Cammell Laird Fullagar engine in 1920, and was one of the parts which gave no mechanical trouble in both the fast and slow running versions of this engine. This arrangement of scavenge pump required no additional driving mechanism, when compared with the lever driven type, and must therefore lead to reduced maintenance costs during the life of the engine.

The elimination of spherical bearings in conjunction with the considerably shortened and rigid crankshaft, should reduce the cost of the engine. From Doxfords' latest report on the complete trials of this engine, these parts had apparently run satisfactorily over the prolonged test-bed running carried out at Palmers Hill.

He was not so happy about the proposal to use white metal 5 mm. thick, which probably had behaved quite satisfactorily on the rigid test bed, but it remained to be proved whether this was a good decision for operation under seagoing conditions, and in the longer term policy whether these bearings could be successfully re-metalled by indifferent labour.

It had been noted also that the side cheeks of the side connecting rods were white metalled, but it had been found that this was quite a difficult white metalling operation in which to obtain satisfactory results, both during manufacture and in operation.

The design of the piston head appeared to permit too much flexibility for the ring carrier portion, and would allow distortion at this point as the piston crown heated up to working temperature.

It would be of interest to hear from Mr. Jackson as to the appearance of the large spherical pad on the top piston after prolonged running, as it would appear this part might be difficult to keep satisfactorily lubricated, particularly should any water leakage occur in the piston cooling connexion.

Some details from Mr. Jackson as to the method adopted in the supercharged version of the engine for guiding the upper piston would also be welcome.

One final criticism he wished to mention was the attachment of the side rod to the side crosshead. The locking of the lower end of this side rod appeared to be rather inaccessible, which could lead to serious trouble if the locking nut was not satisfactorily tightened up after, say, an overhaul.

He concluded by once again complimenting Mr. Jackson on his extremely interesting paper, and for the frank way in which he had shown all the detail particulars in the illustrations.

MR. W. F. JACOBS (Member) in commenting on Mr. Jackson's paper, wished to say that he had been very interested in the Doxford engine since he first saw the plans in 1920, though his actual experience had been rather limited.

The nearly perfect balance and airless injection (when almost all other C.I. engines had air blast) were very good features, though the scavenge pump crank seemed quite out of harmony with the power cranks.

There were a few items in the new engine which he wished to mention however.

The matter of one part or tripartite liners seemed to be mainly one of cost spread over the life of the vessel, though it might have seemed more logical to have had the section which was in contact with the first blast of the firing of some material which was suitable for it, while the portion subjected to ring wear was made of material of a type standing it better.

The joint between the various parts of the built-up liner was not in the same category as that of a cylinder cover, and only had to stand the stress of keeping the joint tight, and no pressure power stresses were placed on the joint bolts.

The latest design of scavenge pump seemed to be the simplest and maybe cheapest in the normally scavenged engine. They were light, simple and easily serviced and out of the way.

There were, however, some sharp corners in the air passages which might be eased into a more streamline form without difficulty.

With this design, if one pump failed, the effect on the running of the engine would probably not be marked.

In the new design it was noticed that the exhaust ports were right-angled and not cut at an angle to the cylinder axis of about 20 degees as he recalled. Was there any noticeable difference in ring wear or breakage so far noticed?

There was one point in the paper applicable to all screw engines, the astern power.

It was stated that the greatest astern power was not as great as the available ahead, and that a certain engine could only develop 7,000 h.p. astern, but he could not determine what the full power ahead was of the engine in question.

Now there was a great difference between the functions of the screw in what it was required to do, between full ahead and full astern.

Full ahead was required as a normal rule to drive the vessel as fast as required on her course, while full astern was only required as a brake, except in such vessels as ferries, icebreakers, etc.

It might be that fairly high powered vessels were not able to take advantage of full astern power because of the various adverse reactions between the screw stream and the hull form, bossings, rudder and so on and it might very well be that any power over a certain fraction of "full" would only be dissipated in shaking the ship, causing vibration, etc.

It was doubtful that any mathematical solution could be found, and any facts could only be ascertained by experiments.

Tank experiments might be made, perhaps had been made, but if so the results were not broadcast to mariners as far as known.

One of the ways in which a reliable full-sized experiment might be made was that the vessel should be fitted with an astern thrust meter on the shaft, moored, with a very long warp indeed, to a pinnacle rock in as deep water as could be found, with a dynometer fitted in the warp at the bow. Then the engines should be put "full astern" and after static conditions had formed, readings of the thrust meter and the dynometer could be made, and compared.

If this were done with a vessel having engines which developed the same power, full ahead or full astern, it could be ascertained if the peak of the astern power had any effect. When anyone had watched a ship going full astern, it caused doubts that much of the churning was anything more than spectacular effect.

Here he mentioned, that, if in service, a "double full astern" was ordered and the ship did not vibrate with the mainmast describing a circle in the sky, the Master would think, and might be voluble in saying that the "Old Chief" had not given her all he had!

If it was found that the ship could not effectively use the peak of astern power, there was no reason for troubling over the astern power being less than the full.

He wished to suggest that the lower the full speed, the greater the percentage of ahead required for the full astern.

The Master of a lower power turbine vessel, of about 10 knots when she could manage it, with about 60 per cent of astern power, told him that if he rang "full astern" once or twice when going full ahead, something started to happen after a quarter of an hour.

A 10 knot vessel might be able to accept the same astern power as ahead, but it was doubtful that a 15 knot vessel would do so.

Shaking the ship was not putting the retarding pull on it very much.

In respect of the rectangular scavenge pump on top of the transverse beam, he saw no reason for anxiety as to its efficiency or reliability, but it did come in the way of removing the upper piston easily, needing various stripping outside the piston gear itself and it did need extra trunking, and might be more costly in the end (building cost), he would not venture to comment; though it might be slightly more expensive and it looked a clumsier engine.

In lower piston cooling, if it was practicable to have the coolant pressure glands inside the entablature, above the crankcase diaphragm, it would be safe to use water for cooling, using a wiper gland on top of the diaphragm to stop drips, not under pressure, going into the crankcase. It would need the top of the stand pipes raising higher, and it was not clear if there was room. Water was a better coolant and the pumping power would be less and would not need so big an oil cooler, or one for piston cooling oil only, if such was fitted.

At the discussion at which he was present, none of the speakers mentioned the matters he had mentioned, at least in the same way.

As for the other details of design, much better and more competent minds than his had already commented and would no doubt continue to do so.

MR. J. H. MILTON (Member), wrote to say that the design of this new engine, so comprehensively detailed in Mr. Jackson's very interesting paper, showed that the new PT engine, when compared with its well known predecessors, was outstanding in many features. The noteworthy ones which appealed immediately were first and foremost the crankshaft; secondly the abolition of spherical bearings; thirdly the increased depth of bedplate and strength of main bearing cross girders and fourthly the stress relieving of the bedplate.

There were several other new features about which he would like to ask Mr. Jackson for further enlightenment:

a) The general adoption of shell bearings—did this mean that generally speaking such bearings were, as in automobile practice, meant to be turned into place as supplied? Alternatively, was any provision made for adjustment, always bearing in mind that it was the clamping effect of the top half which maintained the shell solid and stopped movement and subsequent frettage at the backs of the shells.

b) The piston heads were of unorthodox design in as much as they appeared to have lower coolant capacity than most, and also were bolted to the piston rod towards their centre so that they were, as was stated, free to expand towards their peripheries. Whereas this latter statement sounded common sense, the heat transferred from the piston crown, which appeared to be very near to the top of the piston rod, in which the cooling oil passages were drilled, could, one would have thought, possibly be high enough under some conditions to cause carbonization of the oil and subsequent restriction of the passages in the top of the piston rod.

Furthermore the relatively hot piston crown, in close proximity to the piston rod, was secured to it by a ring of collar studs and, in view of the closeness and rigidity of these two parts, one might expect trouble with the studs through differential expansion of the items they secured together.

c) The upper piston at the outer end of its stroke was so far out of its cylinder that it would have appeared essential, especially for heavy weather conditions, that guides be fitted.

d) What provision was made for adjusting the top end bearing pad?

e) Were any arrangements fitted, such as air injection, to combat hammer in the piston cooling systems.

f) It was noted that the centre connecting rod bearings, top and bottom, also crosshead, were now lubricated by oil from one of three telescopic pipes; this additional telescopic pipe, feeding as it did, such important bearings, would appear to be an unnecessary hazard.

Author's Reply

Mr. Jackson, in replying to the discussions, thanked all those who had taken part for their kind remarks and appreciation of the new Doxford engine and for their contributions which would make the paper all the more valuable. A number who had contributed had raised similar questions so that in such cases a reply would be given only to the first speaker.

Mr. Cook had raised some interesting questions and had pointed out that the British Shipbuilding Research Association were conducting experimental work in connexion with a number of the features embodied in the new engine. Mr. Jackson said that probably the most interesting B.S.R.A. reports in connexion with the Doxford engine were those connected with the air flow of the original 5-cylinder engine and a later one which had described investigations into the temperature of the pistons and cylinder liners, this having shown that the Doxford liner was, if anything, too cool immediately below and above the exhaust ports and this had led to the decision not to cool the upper liner above the exhaust ports.

The question of pressed-in bobbin pieces in the centre crankpin had been raised by Mr. Archer as well as by Mr. Cook and while there were no experimental figures available to show that these were advantageous, nevertheless, it was the opinion of those engaged in building up the crankshaft that the crank webs had a better grip on the crankpins with the bobbin pieces than without them. Mr. Jackson was aware of the B.S.R.A. tests showing plastic flow with a shrink fit exceeding 1·2 thousandths per inch diameter but in view of Doxford's many years of successful experience with a shrink of 2 thousandths per inch diameter, he was not prepared to change this successful feature.

Mr. Jackson thanked Mr. Atkinson for his remarks and for the tables which he had given of various proposals for a range of Doxford P type engines. These various tables had been considered along with others to cover the range of powers up to 18,000 b.h.p. Maybe at a later date a paper would be given on the engine which eventually resulted from these various exercises.

Mr. Church had raised the question of the unequal strokes of the Doxford engine and Mr. Jackson agreed that to get the maximum power from an opposed piston engine the upper stroke should be equal to the lower, but there were other factors to consider. The Doxford engine of the 1920's had such an arrangement but it was a tall engine and there had been trouble from vibrations arising from the out-of-balance couples of such an engine. In consequence, Mr. Keller had introduced the balanced engine about 1927 and had achieved balance by reducing the upper stroke so that the WR's of the upper and lower reciprocating parts were equal. Mr. Jackson had reduced the upper stroke still further on the P engine in order to obtain a more rigid crankshaft with a considerable overlap of the crankpin and journal as had been described in the paper "Some Crankshaft Failures; Investigations, Causes and Remedies". With regard to the three-piece cylinder liner there were several reasons for the adoption of this construction apart from the desire to replace the upper and lower halves independently. It was also desired to eliminate the possibility of water leaking into the combustion chamber through the fuel valves, relief valve and starting valve pockets which was a source of weakness in the earlier engine. Also, at the time the P engine was designed there was difficulty in obtaining a supply of castings of the long one-piece cylinder liner and the division of the liner into two pieces opened up further sources of supply so that at the present time a considerable number of liners were being obtained from a foundry which had made equipment for centrifugally casting the liners.

With regard to the question of oil cooling versus water cooling, the relative advantages were dealt with in the paper and it was possible that the adoption of oil cooling in addition to the diaphragm to eliminate crankshaft corrosion with boiler fuel was a double remedy. Nevertheless it was not easy to obtain a completely leak-proof cooling gear inside the crankcase for the supply of water to the lower pistons and water should be kept out of the crankcase lubricating oil. With the close cylinder centres of the P type engine, it was not possible to fit the swinging link gear of the earlier engines and thus the decision to adopt telescopic pipes; these were simpler and there was no greater tendency for hydraulic hammer with telescopic pipes than with swinging links.

With regard to the adoption of a maximum cylinder pressure of 1,000lb./sq. in. for highly rated engines, this was a necessity if good combustion and good fuel consumption were to be achieved. With turbocharging, a compression pressure of 360lb./sq. in. at light loads would increase to well over 500lb./sq. in. at 135 m.i.p. due to the increase in boost pressure with increase of loads. A new type of engine had to be designed for a possible life of twenty years and during that period, powers, mean pressures and boost pressures would increase so that a maximum pressure of at least 1,000lb./sq. in. would be required and no devices of pilot injection and long, slow, injection of the main charge would give an equivalent fuel consumption.

With regard to the 9 deg. advance of the exhaust crank and the exhaust port opening of 75 deg., there was no loss of power from the engine cylinders due to this. The first turbocharged engine produced by Doxford had a crank lead of 4 deg. and the exhaust ports opened 67 deg. before bottom centre, but this engine would not run satisfactorily at low loads without the assistance of an engine driven scavenge pump. The next turbocharged engine had a lead of the exhaust crank of 6 deg. and the exhaust ports opened 70 deg. before bottom centre but again, though better, this engine was not satisfactory at slow speeds without the assistance of scavenge pumps. The P type engine therefore had been built with a lead of the exhaust crank of 9 deg. and an opening of the exhaust ports of 75 deg. before bottom centre and this engine would run at slow speeds, both ahead and astern, without any assistance from an engine driven scavenge pump or motor driven blower and there had been no loss of power from the engine cylinders. The earlier opening of the exhaust ports had given a greater impulse of the exhaust gases to the exhaust turbine, somewhat at the expense of the engine cylinders, but the increased quantity of air and greater pressure from the turbo-blower had more than compensated in increased power and efficiency from the engine. There would ultimately be a point at which the engine cylinder would begin to suffer—possibly at about 12 deg. of advance of the exhaust cranks and an exhaust port opening of 85 deg. There had been no failure of an engine to start astern, due to the lower compression.

The author agreed with Mr. Church that the normal scavenge pump engine was the simplest for powers of under 5,000 b.h.p. but nevertheless the orders for P type engines up-to-date showed over thirty engines of turbocharged type against only one with normal aspiration and scavenge pumps. Mr. Jackson thanked Mr. Church for his many questions and remarks showing a thorough understanding of the P type engine.

Mr. Munton's question regarding the relative merits of piston cooling by lubricating oil and by water had been answered in the reply to Mr. Church. With regard to the lubrication of the main bearings, the lower half shell was not grooved all the way round and there was no groove in the upper half so that the tendency for lubricating oil to leak in the case of excessive bearing clearance was much reduced. The arrangement of the oil supply to the main bearings was shown in Fig. 54.

bearings were performing satisfactorily on the three ships at sea. He personally liked the double transverse girders provided on the bedplate of the six-cylinder engine though he appreciated that this was a more costly construction and it was not a necessity on the shorter four-cylinder engine. With regard to integral thrusts, this was still a feature of the four- and five-cylinder engines but not of the six-cylinder, since the aft half crankshaft with an integral thrust became too long. Many Doxford engines had previously been built with external thrusts and in fact one very large tanker company did specify independent thrust blocks even on four-cylinder engines. One 140 cu. ft. bottle for the air starting system would give over 18 starts and on a favourable trial some 24 starts of the engine.

With regard to manœuvring, the PT engine could be stopped and reversed in eleven seconds on the test bed but in a loaded tanker it was necessary to overcome the considerable kinetic energy of the ship and this was common to all engines. The N.E.M. braking device was advantageous. The PT6 engine and others of the PT range would give a power of 65

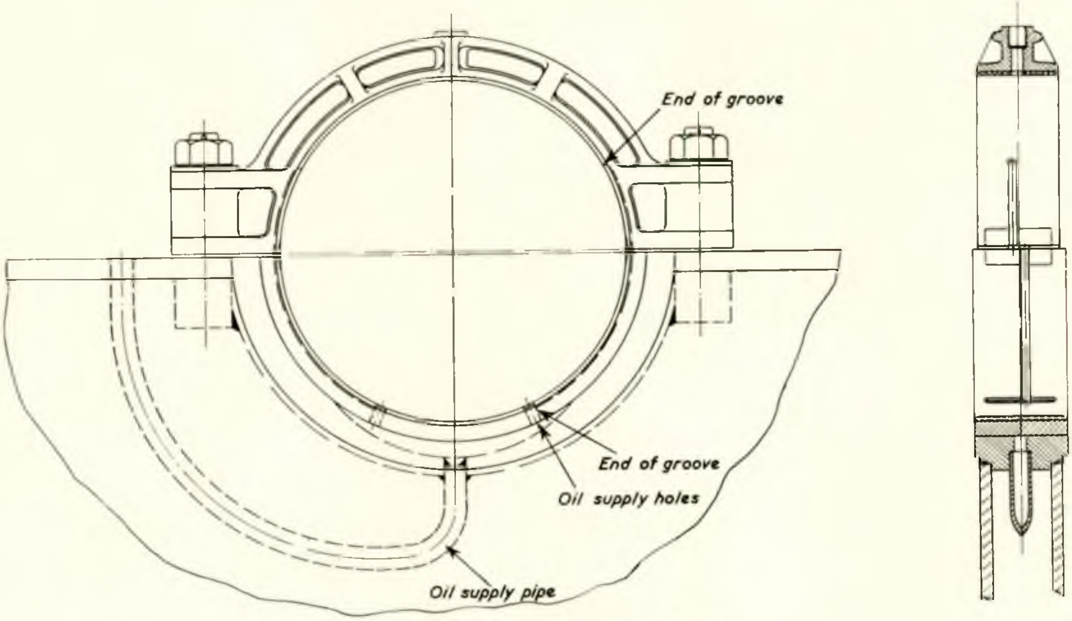


FIG. 54—*Oil supply to main bearings*

Mr. Archer's comments on the torsional vibration conditions of the six cylinder engine were correct and the ninth order II node critical speed did occur at about 128 r.p.m., but it was sharply tuned and above the running speed, whereas in the older engine the ninth order occurred at about 85 r.p.m. and was thus in the running range. The six-cylinder engine did require a detuner but there were now over twenty-five four-cylinder engines on order and not one of these was requiring a detuner, nor did the five-cylinder require one. Mr. Archer's comments on the reduction of stroke of the upper piston and on the bobbin pieces in the centre crankpins had been dealt with in the replies to Mr. Church and Mr. Cook respectively. Mr. Jackson agreed that the bobbin pieces added weight to the crankpins.

The crosshead guides were bolted to the A-frame as in the earlier engine but the bedplate had been considerably increased in longitudinal stiffness as also had the crankshaft and, in addition, the cylinder centres had been much reduced so that the deflexions, which had caused broken guide bolts on the older engines, would be considerably reduced. With regard to the fitting of the transverse beam and upper piston, the nut on the end of the upper piston rod was given a clearance of 0·004in. to 0·006in. so that it was possible to obtain an independent movement of the piston relative to the rigid transverse beam to secure satisfactory alignment. The non-spherical

to 70 per cent at about 90 r.p.m. when running astern without any increase in temperature of the air in the entablature. There had been no case of fire in the scavenge mains of turbocharged Doxford engines with the opening of the exhaust ports 75 deg. before bottom and the opening of the air ports 42 deg. before bottom, thus giving a lead of the exhaust opening of 33 deg. This gave ample time for the exhaust blow down before the scavenge ports opened.

Replying to Rear Admiral Mark-Wardlaw, Mr. Jackson stated that the fuel consumptions given in the performance curves were with a medium high viscosity fuel of about 1,100 seconds Redwood No. 1 viscosity, having a net calorific value of 17,600 B.t.u.'s. Mr. Jackson regretted the misprint on page 28 to which Rear Admiral Mark-Wardlaw had drawn attention.

As Mr. Victory pointed out, there were over 1,400 Doxford engines of the original type at sea, which had given satisfactory service in the propulsion of cargo ships and tankers all over the world. There had been some defects with this engine such as corrosion of the crankshaft when using residual fuels, but this had been entirely overcome by the adoption of a diaphragm separating the cylinder from the crankcase. It was not in any way due to this that the new engine had been developed but it was in order to obtain a lighter and more compact engine of less cost and one capable of deriving the greatest benefit from turbocharging. The question of oil cooling and water cooling had already been dealt with in the replies to Mr. Church. It had also been explained why telescopic pipes had been adopted. It was not possible on the P type engine to fit swinging links. It was desirable with either telescopic pipes or swinging links to introduce air into the system to avoid hydraulic hammer and this could be done either through snifting valves or by supplying air from a compressed air system and while snifting valves had to be changed on occasions, they seemed simpler than the valves and filters required with the compressed air system.

With regard to the suggestion that the one remaining camshaft should be eliminated from the P engine, Mr. Jackson was of the opinion that a camshaft was the simplest method of driving the several components which were necessary for the successful operation of an engine, such as timing devices for the fuel injection, starting air distributors, lubricators and indicator gear, as well as a revolution counter and wrong direction alarm. The trials which had been made on a Doxford engine with a compression operated pump (which was an essential to the elimination of the camshaft) had not been entirely satisfactory even on the normal aspirated engine and the problem of operating such a pump on a turbocharged engine was greater, due to the increase of compression with load and the different compression for ahead and astern running, resulting in later injection when running astern. He was convinced that a light camshaft for operating the timing valves with its simple chain drive was the best method of driving these various components and, in addition, the fuel pump and sometimes a booster pump had to be driven.

With regard to the question raised by Mr. Norris, most of the bolts of large engines were tightened by the normal hammering up process with a substantial spanner and a large hammer. Doxford's had, however, adopted a hydraulic method of tightening the locking plugs underneath the side rods and side rod top end nuts and considerations were being given to extending those processes and also to tightening some of the nuts by means of a scissors gear.

With regard to astern running, the power of 7,000 h.p. given in the paper was for the PT6 engine which had a power ahead of 10,000 b.h.p. at 120 r.p.m. There was no blow back of the exhaust through the scavenge ports up to 7,000 h.p. when running astern and the entablature was not subject to fouling in either ahead or astern conditions up to the powers mentioned and this had been amply verified by the successful running of the *Montana, Tudor Prince, St. Rosario* and others. Sub-division of the entablature was not necessary and such a device caused considerable fluctuations of pressure, leading to panting of the entablature frame and doors unless these were very substantial. Sub-division was only necessary where it was essential to give assistance to the turboblower for slow running, as in a loop scavenging engine, and where the underside of the piston was used as a scavenge pump. The question of the time taken for running from full ahead to full astern had already been dealt with and the time to stop the ship, which was the important question, was shown to be quite as low with the opposed piston engine as with other types, according to Mr. Norris' own tables. There was no reason whatsoever why it should take longer to reverse a 650 engine than a 670 engine and the difference must be due to either the speed or loading of the ship or due to the method of operation. Doxford's had, on occasions, supplied a master air stop valve close to the engine control for opening up the air bottles quickly in case of emergency. With regard to the pressure of the starting air bottles of 600lb./sq. in., it was entirely a matter of economics, and, with this pressure the PT6 engine fitted with two 140 cu. ft. bottles would give over 18 starts from one bottle, whereas if the pressure was reduced to 350lb./sq. in. the capacity of the bottles would have to be nearly treble at 400 cu. ft. to give an equal number of starts. Such bottles were of enormous size and added to the weight and space occupied.

With reference to the question of the packing of the piston telescopic pipes, these pipes were mounted on a spherical end which also had radial freedom so that the pipe could adjust itself in true alignment and, in addition, the gland itself had radial freedom. The packing, which had proved most successful was Beldam's Panda packing. With regard to upper piston guides, these had not proved to be necessary or even desirable. Mr. Jackson knew about the difficulties of the wear of the top guides on the original engine and when the single cylinder prototype engine was conceived, he fitted the best possible guides that he could devise, consisting of aluminium shoes mounted to run in guide channels in the upper structure, which formed the side faces of the rectangular scavenge pump as shown in Fig. 2. These slippers had to be eased so often on this engine that ultimately he concluded that they were not even bearing in the guide surfaces and were performing no useful function so that they were taken off and the engine subsequently ran equally well, or better, without them. The movements of the side rods transmitted to the transverse beam were caused by slight misalignments of the side connecting rods and side rods, inevitable in such a long line and they were also caused by the side clearance of the side crossheads and also by any discrepancies in the lengths of the two side connecting rods and side rods and by the twist which occurred in the crankshaft between one side crankpin and the other. It was not possible to correct or restrain these movements and experience of the engine so far had shown that the upper pistons were running quite satisfactorily, without any guides, even in rough weather, which caused pitching of up to 16 deg. The test curves given on page 224 were from the engine operating on a fuel of about 1,100 secs. viscosity having a net calorific value of 17,600 B.t.u.'s and the air temperature at the air cooler outlets was about 85 deg. F.

Mr. Worn had given information of the trials of the Brush turbochargers on the first PT6 engine and these turbochargers did give equal results to those obtained with the Brown-Boveri turbochargers and the lubricating system for the Brush turbocharger, with plain bearings lubricated from the engine's lubricating oil supply, was very simple. Investigations were also made, as described, of the noise from the turboblowers until this was reduced to quite acceptable levels.

Rear-Admiral Cowland and other speakers had asked for more information of the performance of the engine, but as Mr. Munton had remarked, some information had been given elsewhere and, in any case, it would be premature to give all the information at that moment since the author had been asked by the Institute to give a paper to the International Conference to be held in May of next year. The h.p. figures given were for continuous service at sea and the blowers used for the two and three-turbocharger arrangement on the six-cylinder engine were respectively of the Brown-Boveri VTR630 or the Brush 50 series turbocharger. These blowers were primarily designed for the two-turboblower arrangement and were too big for the three-blower arrangement and the next six-cylinder engine, on test in June, was being fitted with Brown-Boveri VTR500 turbochargers which would be more suitable for the conditions. With regard to cost, there was little difference between the two VTR630 and the three VTR500 turbochargers, though it was agreed that there was one less running unit with the two-blower arrangement. On the other hand, due to the extra pipes from numbers 1 and 6 cylinders, the blowers were further from the engine which increased the volume of the exhaust pipes and made them more tortuous and also the platform spaces around the engine were more restricted.

Mr. Yellowley asked a number of questions. With regard to the ratio of the lower and upper strokes, the major consideration was to increase the strength and rigidity of the crankshaft by a greater overlap. Mr. Keller had reduced the upper stroke in 1927 to produce a fully balanced engine since there had been vibration difficulties with the previous engine having equal strokes for the upper and lower pistons. Mr. Jackson reduced the upper stroke still further in order to strengthen the crankshaft. The PT engine was designed for an m.i.p. of 150lb./sq. in. and a maximum pressure of 1,000lb./sq. in. to provide for developments over the next decade. During the shop trials, extending over some 3,000 hours, there was relatively little wear-down of the main journals which were of large diameter and short length, without any sphericals and the maximum increase in deflexion was at the forward end of about 0·006in. due to the fitting of the detuner. With regard to the spherical pad and the direct lubrication for the centre connecting rod top end, it was not possible to say which of these factors was most responsible for the successful operation. The loading of the top end bearings, the distortion of the centre crosshead pin, the alignment of the connecting rod and the lubrication had all been considered and attention had been given to each factor. It was considered that the locking plug inserted in the side crossheads for tightening up the side rods was an improvement on the differential nut with which there had been difficulties, due to insufficient hardening up, due to dirt and due to crossed threads. The screwed locking plug had been tightened by means of a hydraulic jack. The author agreed with Mr. Yellowley that it was doubtful whether the double timing valve was justified. At the time that it was introduced, some timing valves were sticking and therefore the double valve was designed in order to provide a double security, but since that time, with improved materials and manufacture, it was now very rare to have a timing valve stick. He doubted whether the jerk pump system of fuel injection was simpler than the common rail timing valve system, since a jerk pump imposed a considerable load on the camshaft which would necessitate a much larger camshaft than employed with the common rail system, together with a heavier driving chain and also some form of reversing gear would be required for manœuvring. The timing valve system using a symmetrical cam did not require any change in the camshaft setting for ahead or astern running.

With regard to Commander Hall's suggestions, while Mr. Jackson agreed that the engine driven pumps were more economical than electrically driven pumps, in respect to the power required from the auxiliaries, yet it was a considerable complication on the engine to make arrangements for driving rotary pumps and the hydraulic drive system was very complicated.

Regarding Mr. Archer's written contribution, it was appreciated that the crankshaft scantlings might require reconsideration for the ultimate rating of the engine at 150lb./sq. in., m.i.p., and 1,000lb./sq. in., maximum pressure, though the PT6 crankshaft was submitted to Lloyd's Register of Shipping for approval at these ratings in the initial stages of the design some years ago. It was agreed that the material for the doglegs would require to be 32 tons/sq. in. though on large forgings of this type the Izod Value of the test pieces was generally reduced with increase in tensile strength. With regard to the design of the centre crosshead pin, the segmental pad did provide full support for this pin though it had been found that the segment had to be fitted some 4 to 6 thousandths clear to allow for the deflexion of the crosshead pin, as otherwise it took more than its share of the load.

With regard to the bedplate cross girders, while double girders were being provided for the five- and six-cylinder engines, it was the present intention to have single girders with cast steel bearing housings, as advocated by Mr. Yellowley for the four-cylinder engines, since this was a cheaper construction and it had been on the longer bed plates of six-cylinder engines that deflexion and misalignment had caused difficulties.

The compression pressure of the PT engine varied with load, as previously pointed out, and was as low as 360lb./sq. in. at low loads rising to 520lb./sq. in. at full load. It was still desirable to circulate hot water around the engine before starting. In the original prototype P engine, the author had increased the compression ratio to provide cold starting, but the subsequent running of the engine gave maximum pressures some 80/100lb./sq. in. higher at full load and the combustion of the engine was rougher, so that the compression ratio was reduced to that of the earlier engine.

In answer to the written contribution from Mr. Davies, the plain short length bearings of the crankshaft were running satisfactorily in the three engines at sea so far (May 1961) and the bed plates had also proved stiffer than the bed plates of the original engine as shown by deflexion readings. Thin white metal of 5 mm. thickness had been used in Doxford engines intermittently by both Doxford and some licensees for over ten years. The first engine with such bearings, in the *British Loyalty*, was fitted with this type of bearing throughout, in 1949, and about two years later Mr. Bulman, of Hawthorn Leslie, fitted bearings with white metal 5 mm. thick and eliminated the sphericals on the side rods of an engine. The lubrication of the pad on the upper piston was by an initial smearing of molybdenum disulphide grease and there was very little movement on this pad. The question of guides for the upper piston had been dealt with in the replies to Mr. Norris and others.

Mr. Jacobs raised a number of questions but the reason for the adoption of a two-piece liner and central combustion space had already been dealt with in the reply to Mr. Church. The upper and lower half liners were of cast iron and the central combustion chamber was a water cooled steel casting and the author considered that these were the best materials for withstanding the conditions. The exhaust ports in the P type engine were cut in line with the cylinder axis since this gave a greater area of the exhaust ports for the passage of the large volume of exhaust gases of the turbocharged engine and, due to the reduction of the upper stroke, these ports and port bars were quite short and so far there was no noticeable difference in the ring wear due to this feature.

The 67PT6 engine would develop 7,000 h.p. astern, which was 70 per cent of the ahead power and Mr. Jackson agreed with Mr. Jacobs that the power did not follow the normal Propeller Law when running astern. With regard to the cooling of the lower piston by water, the use of a gland in the entablature had been considered, though even in this position, any serious leakage of water would be into the air space with the consequent danger of blowing some into the cylinder.

Mr. Jackson thanked Mr. Allan for his kind remarks and for his appreciation of the merits of the crankshaft and bedplate of the P type engine. With regard to the centre crosshead, this did deflect slightly under load and experience had shown that the segmental pad had to be fitted some 4 to 6 thousandths clear and then it took its fair share of the load. The white metal did not appear to wear any quicker than the bronze pad since, if this did happen, then the pad would be carrying more load and would wear in consequence. With this three-piece bearing construction all the bearings wore down together and there was no undue difficulty in adjusting them, but should the white metal bearings get out of adjustment, then liners could be fitted between the bearings and the connecting rod. The method of attaching the piston rod to the crosshead was quite a common feature of some Continental engines and had been well proved. The tightening plug under the side rods had been tightened by a hydraulic jack to give a pre-loading on the side rod threads. With regard to the spherical cast iron pad between the upper piston and the transverse beam, this did permit initial alignment of the upper piston so that it could find its own centre and some pistons were in fact being run with the upper nut subsequently tightened up. The questions regarding the three-part liner construction had already been dealt with and also the question of cooling the lower pistons by water.

Mr. Milton raised the question of the shell bearings and while it was hoped that these could be turned into place as supplied, nevertheless they might require a certain amount of fitting. These bearings were 1½in. thick so that they were hardly the shell bearings of the motor car engine. With regard to the cooling passages in the piston head, it was well known that high velocity of the coolant increased heat conductivity and the heat was taken away from the crown of the P type piston without the top end of the piston rod getting unduly hot. There had been no case of carbonization of the oil on any Doxford oil-cooled piston though this was feared when oil cooling was first adopted but Doxford's did circulate a large quantity of oil in order to keep the oil temperature down to less than 135 deg. F. There was trouble with the piston studs on the earlier engine and the author considered this could be due to the expansion and deformation of the periphery of the piston and the studs of the new engine, being closer to the centre, would not be subject to such expansion and deformation. The question of guides for the upper piston and adjustment of the spherical pad had been dealt with and with regard to the three telescopic pipes, it was not considered that the centre pipe, driven by the same bracket as the cooling oil supply and return pipes, was any real hazard and certainly was less so than the drilling of the crankshaft and cross connecting pipes of the earlier engine, together with the risk of failure of the oil pressure with this system.

In conclusion, Mr. Jackson thanked all the many speakers and writers who had contributed to this discussion.

OBITUARY

SAMUEL BORDER (Member 3622) died after a short illness on 16th May 1961, two months before his eighty-first birthday. After thirteen and a half years sea service, Mr. Border, who held a First Class Board of Trade Certificate, was appointed works manager of Mazagon Dock, Bombay, in 1919. On 3rd June the same year he was elected to full membership of the

Institute of Marine Engineers. In 1928 he was appointed general manager, Garden Reach Workshops, Calcutta, an appointment he held until 1939. During his forty years membership of the Institute he held the office of Vice-President for Calcutta from 1933 until 1938 a year before his retirement. After his return to the United Kingdom, in 1941 he joined the Ministry of War Transport in London as technical adviser, and remained with the Ministry until shortly after the war.

ROBERT WILLIAM CARSON (Member 23468) died on 11th April 1961 aged 48 years, having been elected to full Membership of the Institute only a month before. He served his apprenticeship with Cammell Laird and Co. Ltd., Birkenhead between 1928-33, as an engine fitter. Concurrently, Mr. Carson studied part-time at Birkenhead Technical College. He held a First Class Board of Trade Steam Certificate with Motor Endorsement, and spent his entire civilian career in the service of the P. and O. Steam Navigation Co. Ltd. During the Second World War he served as Lieutenant(E) in the Royal Naval Reserve aboard H.M.S. *Canton* from October 1939 to April 1944, and in H.M.S. *Boxer* from April 1944 to September 1946 when he rejoined P. and O.

Mr. Carson served on numerous ships of the P. and O. Line and, as chief engineer, on motor vessels *Socotra, Somali, Cannamore, Soudan, Salmara, Iberia,* and *Carthage.* He was promoted to chief engineer in 1953.

DAVID JENKINS (Member 4074) was born on 13th April 1873. His apprenticeship was served with Messrs. Williams and Metcalf, Aberystwyth. Between 1895 and 1908 he saw service at sea and obtained an Extra First Class Board of Trade Certificate. In 1920 he joined the London Guarantee and Accident Company as chief engineer and left them two years later for the Municipal Mutual Insurance Co. Ltd. Seventeen years later Mr. Jenkins was appointed chief engineer and manager of the engineering department of this company and retired in 1948. He became a Member of the Institute of Marine Engineers on 2nd November 1920. Mr. Jenkins died on 2nd October 1959.

PERCY LOWN (Member 10496) died on 7th April 1961 aged 42 years. His apprenticeship was served with Richardsons Westgarth and Co. Ltd., Hartlepool between 1935 and 1943. Between 1935 and 1940 he attended West Hartlepool Technical College and obtained the Higher National Certificate in the latter year. In 1943 Mr. Lown began a three-year tour of service in the capacity of fifth engineer, and later of fourth engineer, aboard *Llandovery Castle,* a vessel of the Union-Castle Mail Steamship Co. Ltd. In 1945 he obtained a Second Class Ministry of War Transport Steam Certificate, and gained, three years later, the First Class Steam Certificate. From April 1946 until September 1947 he served as second engineer with the Esso Transportation Company and in February 1948 joined the Billingham, Co. Durham, division of Imperial Chemical Industries Ltd. For the next ten years Mr. Lown was employed as assistant technical officer (draughtsman) in the engineering design department of the division. For the remaining part of his service until his death, he was engaged on work study duties in connexion with the maintenance of wharves and stores in the division's commercial works.

Mr. Lown was elected an Associate of the Institute in 1945 and became a full Member in 1948. He was also an Associate of the Institution of Mechanical Engineers.

JAMES WATSON (Member 7005), who was born on 7th September 1903, died in Preston Hall Chest Hospital, near Maidstone, Kent, on 13th April 1961. His apprenticeship was served with Messrs. Dundas Engineering Co., Grangemouth and from 1923-30 he was at sea, obtaining in this period his First Class Board of Trade Steam Certificate with Motor Endorsement. In 1931 he was appointed a chief engineer with Messrs. Ravelston Steamship Company and six years later took up an appointment with the Customs, Excise and Trade Department of the Palestine Government. During the Second World War Mr. Watson was port superintendent of the Port of Haifa and thereafter held positions in the Cape Verde Islands and Aden. In 1952 he was appointed senior marine engineer and slipway officer in the Port and Marine Department, Government of Sierra Leone. In the following year he joined the Burmese Government Marine Service at Rangoon and in 1957 was employed with a commercial firm in Singapore. His overseas service was finally terminated when he left his position in Hong Kong to return to England on account of his wife's health. His wife predeceased him by one year.

Mr. Watson was a Member of the Marine Engineers' Association and was elected a Member of the Institute in 1932.

S. D. WILKIE (Member 7624) died on 23rd November 1960, as the result of an aeroplane crash. Born on 25th November 1904, he served his apprenticeship with the Caledon Shipbuilding and Engineering Co. Ltd., Dundee, from 1921 until 1926. He obtained a First Class Certificate with Motor Endorsement and in the same year that he finished his apprenticeship joined the Anglo-Persian Oil Co. with which he remained for three years. In 1929 he served with the China Navigation Co., Shanghai, as junior and later senior engineer until 1934 when he took up an appointment with the Chinese Maritime Customs as senior engineer assistant marine surveyor. The same year he was elected a member of the Institute. In 1949 Mr. Wilkie joined the Standard Vacuum Oil Company as assistant terminal superintendent—Marine Department, and the following year was appointed manager of the Mayon Engineering Co. For the next ten years he was with the Visayan Stevedoring Corporation serving as first supervisor and then manager until the time of his tragic death.

INSTITUTE ACTIVITIES

Minutes of Proceedings of the Ordinary Meeting Held at The Memorial Building on Tuesday, 10th January 1961.

An Ordinary Meeting was held by the Institute on Tuesday, 10th January 1961 at 5.30 p.m., when a paper entitled "The Future Doxford Marine Oil Engine" by P. Jackson, M.Sc.(Eng.) (Member of Council) was presented by the author and discussed.

Mr. W. R. Harvey, O.B.E. (Chairman of Council) presided at the meeting at which 286 members and visitors were present.

In the discussion which followed twelve speakers took part.

The Chairman proposed a vote of thanks to the author which was accorded by prolonged acclamation.

The meeting was concluded at 7.55 p.m.

Section Meetings

North Midlands

An ordinary meeting of the North Midlands Section was held at Sheffield on Thursday, 1st June 1961, at 7.15 p.m.

Chairman of the Section, Mr. J. C. Proudfoot, presided at the meeting which was attended by twenty-eight members and visitors.

The Chairman introduced the speaker, Dr. R. V. Hughes, B.Sc., A.M.I.Mech.E., who presented a paper on "The Need for Research in Diesel Engine Development".

The talk was admirably illustrated by models, diagrams, films and slides and the audience showed themselves most interested in hearing how problems were being solved by modern techniques.

A vote of thanks to Dr. Hughes proposed by the Chairman, was seconded by Mr. C. W. Parris (Member).

The meeting closed at 9.40 p.m.

South Wales

The Annual Golf Meeting of the South Wales Section was held on Friday, 16th June 1961, at the Glamorganshire Golf Club, Penarth.

Forty members and friends attended and thirty participated in the competition. The David Skae Cup was won by Mr. T. G. Whitelaw (Associate Member) and the Visitors Tankard by Mr. E. H. Ridgewell. The runners up in the match were Mr. F. R. Hartley (Chairman of Section) and Mr. C. Pilkington. A special prize comprising a stem of bananas, which was presented by Mr. C. Pilkington for the best net score for the middle six holes, was won by Mr. W. E. Brennan (Member).

Mr. David Skae (Vice-President) presented the prizes to the winners who made suitable response.

A vote of thanks to the Captain and members of the Glamorganshire Golf Club was proposed on behalf of the Institute by Mr. H. G. Wickett (Member) and was replied to by the Captain.

Sydney

A meeting of the Sydney Section was held at Science House, Gloucester Street, Sydney, on Wednesday, 24th May 1961. Captain G. I. D. Hutcheson, C.B.E., R.A.N. (Vice-President, Australia) presided at the meeting which was attended by thirty-six members and guests.

A paper entitled "The West Sydney Cove Passenger Terminal" was presented by Mr. E. Ian Griffin, and in the discussion which followed Capts. Hutcheson and Parker and Messrs. H. W. Lees, L. J. Flaherty, D. G. Parker, J. Munro and A. B. Smith took part.

A vote of thanks to Mr. Griffin was proposed by Captain R. G. Parker, O.B.E., and carried by acclamation.

After the meeting supper and the usual refreshments were served.

On Friday, 16th June 1961 a meeting of the Sydney Section was held at Science House, Sydney, at 8.0 p.m.

Captain R. G. Parker, O.B.E., R.A.N. (Member of Committee) was in the Chair and there was an attendance of sixty-six, eleven members, and fifty-five students and apprentices, who were welcomed by the Chairman.

Mr. W. Fyffe delivered a lecture entitled "The Maintenance of the Doxford Diesel Engine" which was followed by a good discussion, numerous questions were asked by the students and these were fully answered by the lecturer.

A vote of thanks to the lecturer was proposed by Mr. D. Gillies and was responded to with warmth. Supper was served at the conclusion of the meeting and an opportunity given to the young men to meet members who were present.