# The Doxford Three-Cylinder Engine

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**Introduction**

In earlier days when Doxford built normally-aspirated engines the three-cylinder engine was quite popular and altogether 137 engines were built between 1929 and 1961. Almost half of these engines are still in service, which is an indication of the reliable performance they have given.

When Doxford began considering turbocharging their engines in 1950, a three-cylinder engine was chosen for the early experimental work. This same engine also became the first turbocharged Doxford engine to enter service. The turbocharging was basically a success, but starting difficulties made it necessary to reduce the lengths of the scavenge and exhaust ports to such an extent that the engine became short of air. Two rows of scavenge ports were used and non-return valves were fitted in the upper tows to prevent flow of exhaust gas from the cylinder into the scavenge belt during the blow-down period. This arrangement allows relatively large scavenge ports for a restricted exhaust port opening. Valves in the scavenge ports are, however, liable to fouling in engines running on H.V. fuel and they are considered to be an undesirable feature today. Further, with the development towards higher mean indicated pressures it became necessary to use bigger ports. Thus, satisfactory starting could no longer be achieved for the three-cylinder engines and these disappeared from the engine ranges offered by Doxford in the sixties and early seventies.

Makers of single-piston engines tend to avoid the designing of three-cylinder units because of the out-of-balance problems. However, with the opposed-piston engine design no real problem materialises. Following the successful turbocharging of the Doxford Seahorse Engine it was decided in 1975 to redesign the Doxford direct-drive engines for constant-pressure turbocharging and calculations showed that the best results would be obtained for an exhaust piston crank lead of 180°. This also made possible complete primary balance of the running gear weights in relation to the strokes for the upper and lower pistons. The next logical step was to re-examine the possibility of a turbocharged three-cylinder engine.

As will be known to most marine engineers the Doxford engine is an opposed piston two stroke engine. It employs no valves in the scavenge-exhaust system and its longitudinal scavenge of the cylinders is relatively unaffected by fouling of the ports. The supply of H.V.F. for marine engines has been greatly affected by the fuel crisis which occurred a few years ago and, since the world energy crisis is likely to become worse, the supplies of fuel for marine propulsion will be further affected in the future. This development has been influenced by two main factors, i.e. the increasing cost of crude oil and the more stringent pollution requirements for land applications. Both these factors encourage the oil refineries to take as valuable a cut of the crude oil for land use as possible and leave a poorer and poorer residual oil which can only be used by marine engines. Finally, drilling for oil now takes place in most parts of the earth, which means that there is a bigger variation in the composition of the residual fuel on the market than say ten years ago.

The above development has made the ship owners look for engines which can tolerate the residual fuels offered. A dangerous feature of the residuals supplied in some parts of the world is the high vanadium content. If sodium, often derived from sea water, is also present, highly corrosive sodium-vanadium compounds are formed during combustion. These compounds become increasingly active on surfaces above 500°C and can cause severe damage to exhaust valves, which are usually operated at about 700°C. They also form deposits on the valve seats, which prevent the valves from sealing properly and thereby cause blow-by and burnt seats. A second danger lies in the big variations which can occur in sulphur content, i.e. from 0.5% to as high as 5%. Sulphur in the fuel promotes corrosive piston ring and cylinder liner wear and the cylinder lubricating oil is therefore made alkaline to match the fuel. Too little wear can result in scuffing, which is very damaging, and too much wear can also be costly. There is also a danger that acid sludge from the cylinder can run into the crankcase and contaminate the crankcase. This is a severe problem on trunk piston four stroke engines and necessitates the use of alkaline lubricants in the crankcase. On two stroke, crosshead engines scraper-ring glands round the piston rods were introduced in the fifties when running on residual fuel became common. Corrosion and sludging problems in the crankcase have since then been rare in this type of engine.

Lastly, high contents of asphalt in the fuel results in general fouling of ports and pistons. A recent investigation revealed that residual fuels with as much as 8% asphalt are offered today. In cross and loop scavenge engines fouling of the scavenge ports destroys the scavenge pattern, which in turn results in poorer combustion, higher fuel consumption and further fouling.

It follows from the above that the two stroke, opposed piston principle used in the Doxford engine is the best for accepting a wide variety of fuel oils. In addition, since this engine has almost complete balance of the primary forces per cylinder, it can be offered with as few as three cylinders, which in turn results in low cylinder maintenance. These two factors were probably the main reasons for a renewed interest in the Doxford engine from the end of 1976 and in spite of the poor trading conditions for marine engines, orders from traditional as well as new customers have been received.

Doxford Engines Ltd. took this opportunity to introduce a number of design improvements based on lessons learned from the service performance of earlier J-engines and test experiences with the Doxford Seahorse engine. Our order book at the moment contains a series of 58JS3 engines and a series of 76J4 engines, both of which

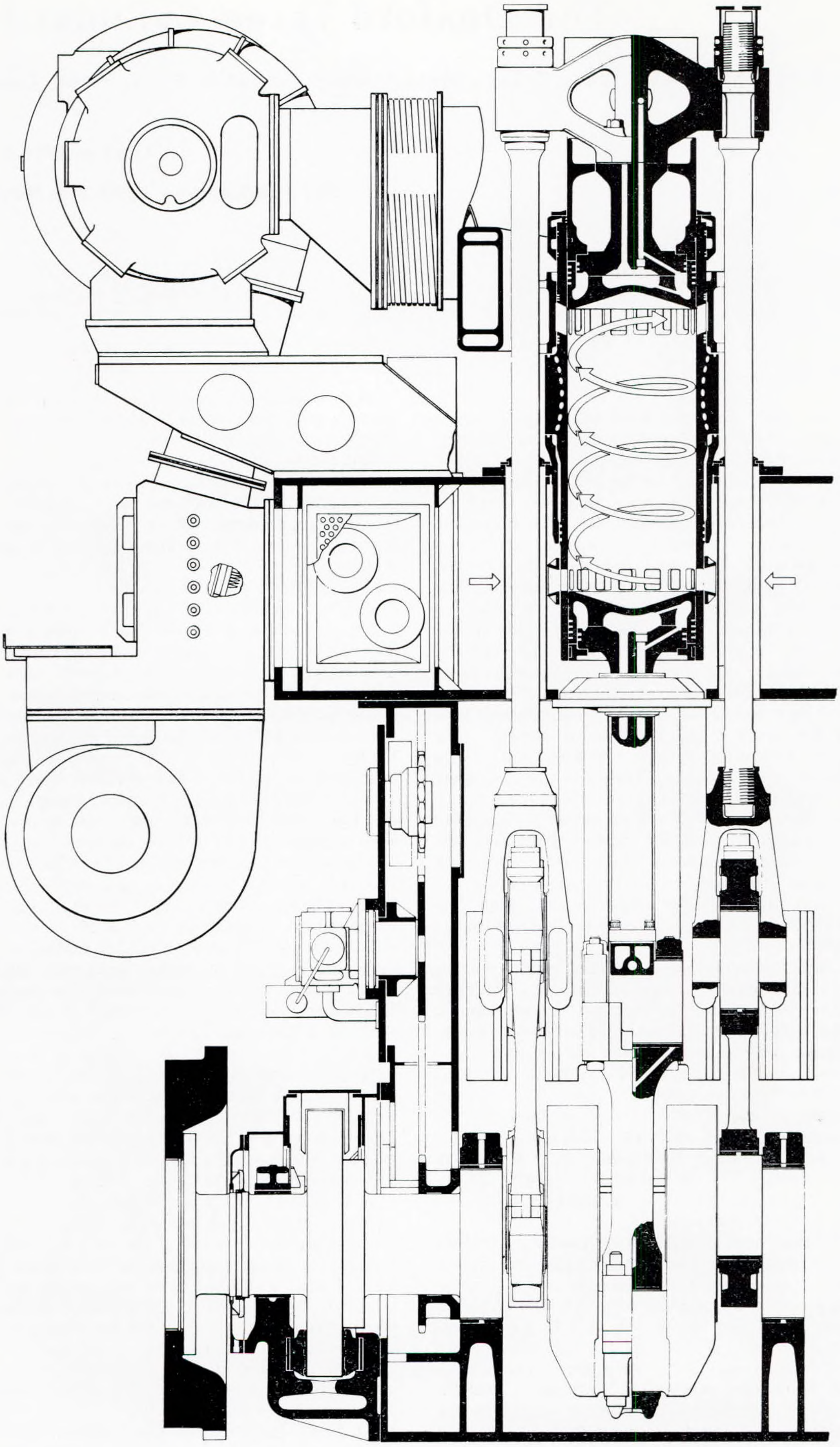


*Fig. 1a—Longitudinal Section of 58JS Type Engine*

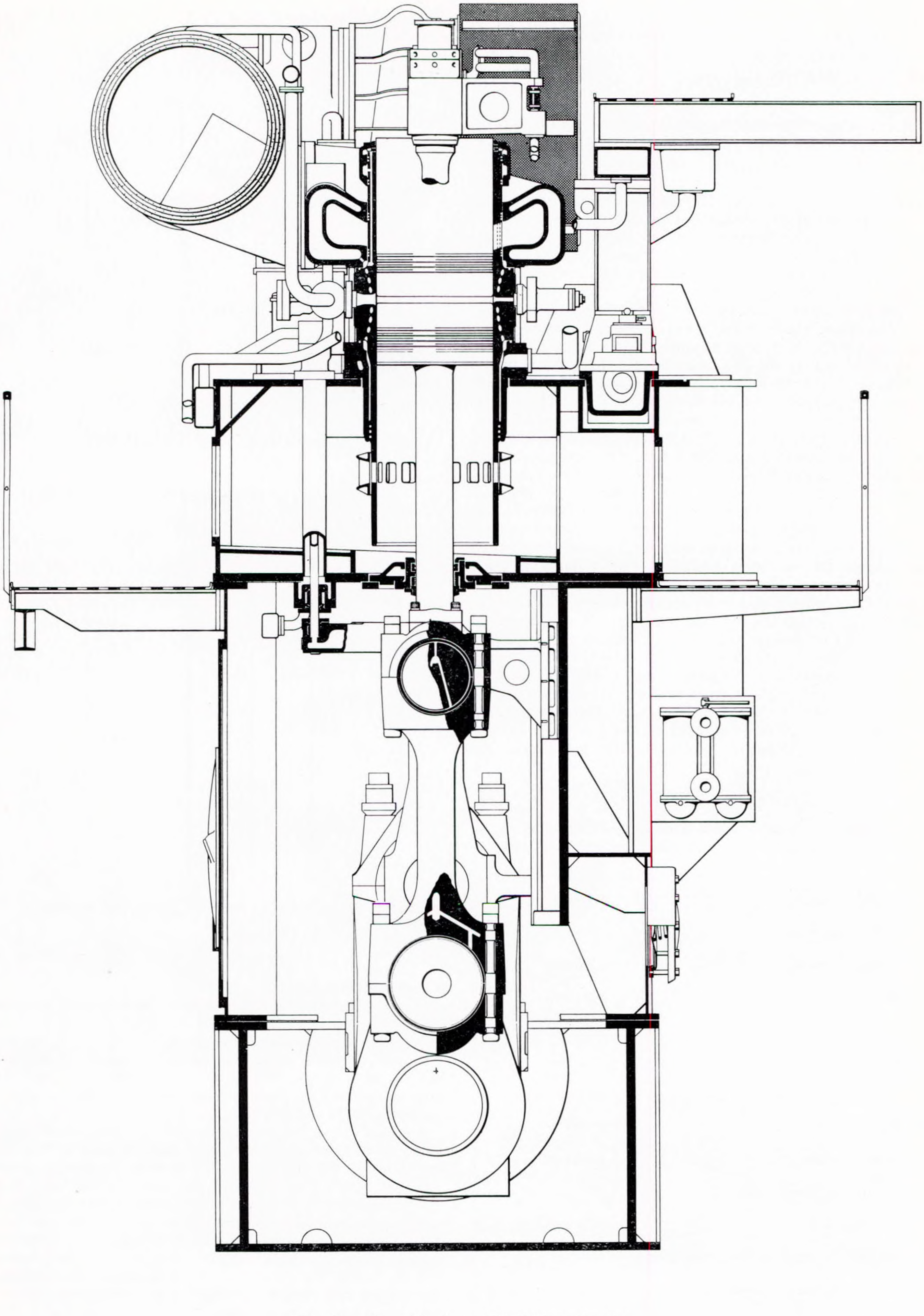


*Fig. 1b—Cross Section of 58JS Type Engine*

are constant pressure turbocharged. Design improvements and test performance of the three-cylinder engine are described in detail and some indications of further development are given.

**Design**

The design of the 58JS3 engine started in the autumn of 1976. The possibility of fitting a Doxford engine to a small container ship built by Appledore Shipbuilders gave the impetus to start the design and the engine was in many ways tailored for the ship. The required height, and speed of the engine at first suggested an engine based on experience from the Seahorse engine. However, the short delivery ruled out prolonged development testing and it was therefore natural to use proven features from the J-engines as much as possible.

Table I shows the leading particulars which were decided upon. The bore and lower piston stroke were made the same as for the Seahorse engine. The speed of 220 r.p.m was stipulated by the shipbuilders and this gave a mean piston speed for the lower pistons well within the J-engine experience, which went up to about 7 m/sec. The upper piston stroke was fixed to give perfect primary reciprocating balance per cylinder. The contracts required a maximum continuous rating of 5000 b.h.p., which corresponded to a brake mean effective pressure of 10.38 bar. This was well within Seahorse experience. However, to provide some scope for development the engine was designed for 5500 b.h.p. Using the above upper and lower piston strokes resulted in an overhauling height of 6200 mm, which was acceptable to the ship builders. The remaining particulars indicated that the engine would be competitive.

The longitudinal and cross section of the 58JS type of engine are shown in Fig. 1a and Fig. 1b, respectively. Although the design was largely based on J-engine, accessibility and overhauling of such a low Doxford engine required special considerations. Thus, it was necessary to use the Seahorse engine side connecting rod design in order to be able to fit and remove these in the crankcase. Also, the camshaft was mounted below the top level of the entablature to improve access to the fuel injectors.

**Table I Leading Particulars of 58JS3 Engine**

| | |
|---|---|
| Bore | 580 mm |
| Lower piston stroke | 880 mm |
| Upper piston stroke | 360 mm |
| Combined stroke | 1220 mm |
| M.C.R. | 4047 kW. (5500 B.H.P.) |
| R.P.M. | 220 |
| b.m.e.p. | 11.42 bar |
| Mean piston speed (lower pistons) | 6.45 m/sec. |
| Max. cylinder pressure | 85 bar |
| Overall length | 6430 mm |
| Height above crankshaft C.L. | 5535 mm |
| Width of bedplate | 2220 mm |
| Height of crankshaft C.L. above chocks | 930 mm |
| Overhauling height (lower pistons) | 6200 mm |
| Weight | 115 tonne |

More detailed descriptions of the design of individual components and systems will now be given.

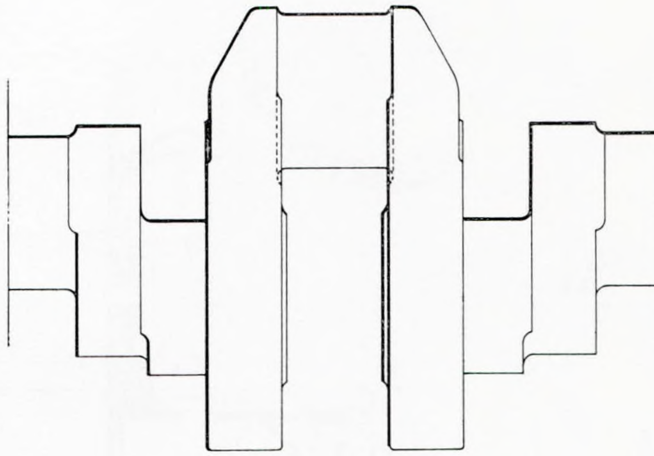


*Fig. 2a—Details of 58JS Crankshaft*

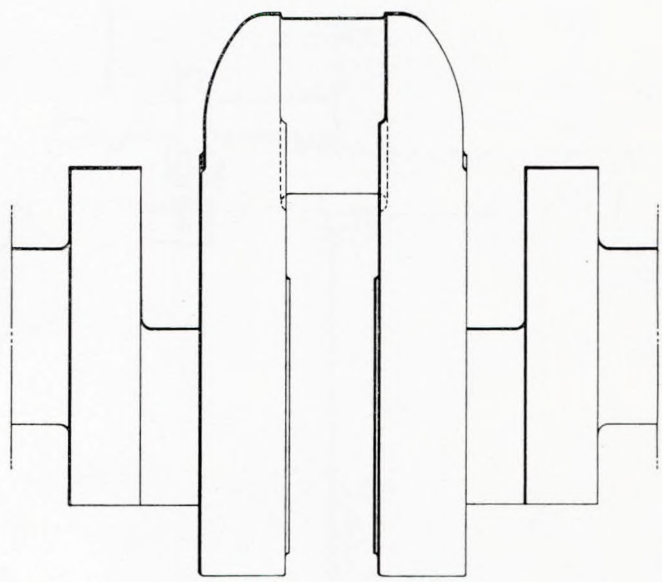


*Fig. 2b—Original J Engine Crankshaft Design*

**Crankshaft and Bedplate**

The crankshaft is of semi-built design with the side-webs and main journals combined in a cylindrical piece, as introduced with the first J-type engines. Fig. 2a shows the details of the shaft using cast centre throws. Forged centre throws have so far been used in only one case when the shorter delivery for these was required. They are, however, generally more expensive.

In most earlier J-type engines the side pin journal piece, often referred to as the 'dog-leg', was designed so that the periphery of the journal was in line with the surface of the side pin furthest away from the shaft centre line. This design is shown in Fig. 2b. For the 58J4 engines, the first of which were built in 1968, journals of smaller diameter were introduced and this practice was followed for the Doxford Seahorse and the 58JS3 engine. A method of calculating the stresses in the crankshaft with greater accuracy than before was first described by Butler and Ørbeck in 1967(1). The results of these calculations have proved useful and calculations are now carried out for all new shaft designs.

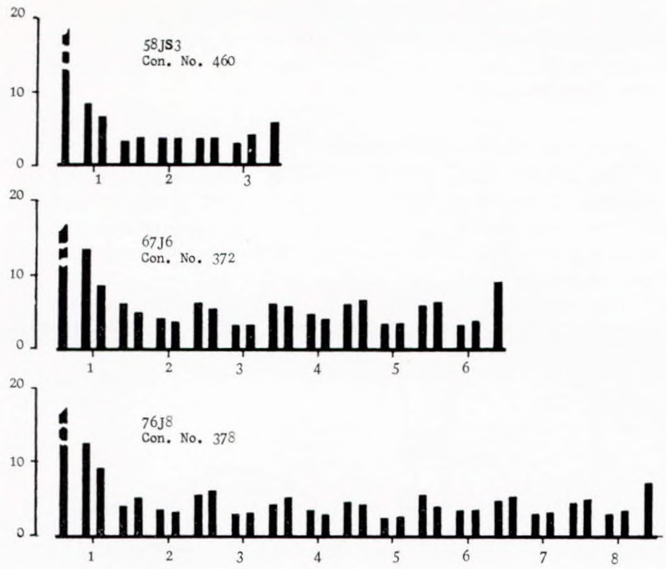



*Fig. 3—Factors of Safety in Semi-Built Crankshafts*

Fig. 3 shows a comparison between the factors of safety for a number of engine types. The calculations do not at present consider torsional and axial vibrations, nor misalignment, and the factors of safety shown are therefore higher than in service. They do, however, provide a good comparison between the 'dog-legs' and the centre throws.

In the original 'dog-leg' design the stresses were considerably lower than for the centre throws in the same shaft. The diameter of the centre pin could therefore be reduced and for the 58JS3 engine the factors of safety are nearly the same for both the journal and the centre pin fillers, the weight and cost of the 'dog-legs' were in that way reduced without any reduction in the overall strength of the shaft.

A cross section in way of the main bearings of the original J-engine bedplate design is shown in Fig. 4a. These engines were running at about 120 r.p.m. and the relatively long stroke necessitated a wide bedplate. To achieve an adequate longitudinal strength in bending an inner girder and an outer girder was used on both sides. With the large diameter main journals, which performed the combined function of earlier side webs and main journals, the cut out of the bedplate transverse girder in way of the main bearings was deep and it was difficult to secure adequate transverse stiffness.

Double rows of chocks were therefore used on both sides of the bedplate, i.e. one row supporting each longitudinal girder. Access to the inner row of chocks was through holes in the bottom plate between the longitudinal girders. However, access to these chocks was restricted and they were more difficult to fit than the chocks in the outer row.

The 58JS3 engine is the fastest running Doxford engine so far built, with the exception of the Seahorse engine. Its strokes are therefore short and the bedplate becomes correspondingly narrow. Compared to the slow running Doxford engine it is light, which was a further reason for a simplified design of bedplate. Fig. 4b shows a cross-section of the final bedplate design in way of a main bearing. One sideplate, 35 mm thick, replaces the two longitudinal girders on each side and gives very adequate longitudinal stiffness. To simplify the installation of the engine it was desirable to use only one row of chocks under each side of the bedplate. The transverse span between the two rows was relatively short, but the short strokes of the engine could also lead to a shallow bedplate, which is useful in order to reduce the engine height. The crankshaft design described above made it possible to choose as shallow a bedplate as the inside clearances permitted and at the same time achieve a very satisfactory transverse stiffness. Thus, an optimum design had been achieved.

**Engine Structure**

The opposed piston principle makes it possible to build an engine which is completely free from out-of-balance primary forces and couples. However, the single crankshaft having three throws per cylinder does not obviate secondary forces. Engines with more than two-cylinders can be so arranged that the secondary forces balance out, but engines with certain numbers of cylinders have a residual secondary couple. The three-cylinder engine is one of these. However, the couple is comparatively small and compares very favourably with the similar couples of five-cylinder and six-cylinder engines having only one piston per cylinder. Single piston engines with four or five cylinders have both primary and secondary out-of-balance couples. From the point of view of balance and vibration the three-cylinder opposed piston engines is exceptionally good.

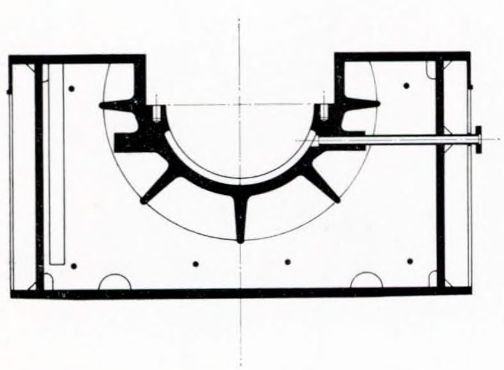


*Fig. 4a—Cross Section of 58JS Bedplate*

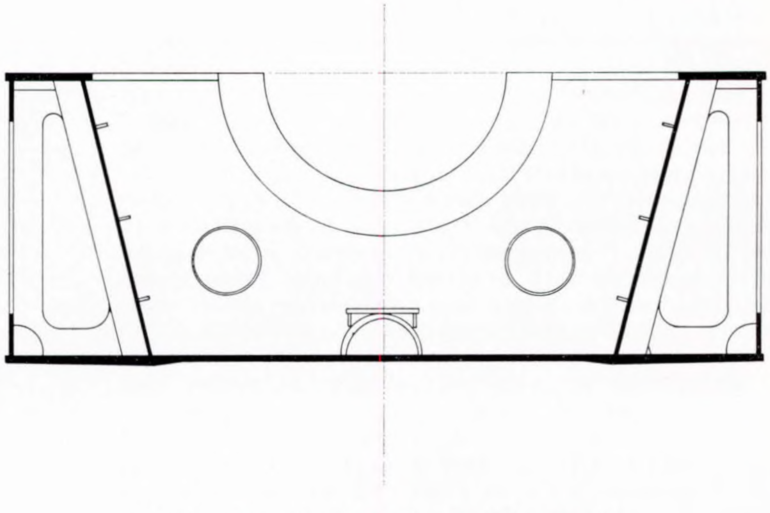


*Fig. 4b—Standard J Engine Bedplate Design*

**Table II Unbalanced External Couples (SI Units)**
Comparison of 58JS range with the Seahorse and with a single piston engine having a bore and stroke of 715 and 800 mm respectively, developing 2000 metric h.p. per cylinder at 256 rev/min.

| | Reciprocating Couple at Rated Speed (MN m) | | | | | |
|---|---|---|---|---|---|---|
| | Primary | | | | | |
| | 3 Cyl. | 4 Cyl. | 5 Cyl. | 6 Cyl. | 7 Cyl. | 8 Cyl. |
| Single piston 4-stroke | — | — | 0.60 | 0 | 0.36 | 0 |
| Single piston 2-stroke | — | — | 0.60 | 0 | 0.36 | 0.60 |
| Seahorse | — | 0 | 0 | 0 | 0 | — |
| 58JS | 0 | 0 | 0 | 0 | — | — |
| | Secondary | | | | | |
| Single piston 4-stroke | — | — | 1.79 | 0 | 0.36 | 0 |
| Single piston 2-stroke | — | — | 1.79 | 1.24 | 0.36 | 0 |
| Seahorse | — | 0 | 0.34 | 0 | 0.06 | — |
| 58JS | 0.43 | 0 | 0.15 | 0 | — | — |

Butler and Crowdy[2] presented a comparison between single piston engines and Seahorse. In Table II their presentation has been extended to include 3-cylinder engines and the 58JS range. The 58JS3 can be fitted with a Lanchester balancer, which removes the reciprocating secondary out-of-balance couple completely, as an optional extra.

As the opposed piston principle ensures that all gas forces and primary inertia forces are contained within the running gear of each cylinder unit, the bedplate has, in the vertical direction, only to carry the weight of the running gear and to cope with secondary out-of-balance forces which, as shown, are relatively small. The structure as a whole has to accept the torque reaction of the engine in the form of the horizontal thrust from the crosshead guide shoes and the equal and opposite horizontal components of the connecting rods forces at crankshaft level. To transmit this torque reaction to the engine seating and the ship, a firm connection must be achieved between the crosshead guide unit and the bedplate.

The crosshead guide unit is a small continuous welded member which extends along the length of the engine and is rigidly connected to the bedplate. To the guide unit are bolted the slipper bars which embrace the crosshead shoes and provide the running surface for both ahead and astern operation.

The entablature is mounted on the guide unit and supported at the end frames, which are double wall units; the one at the after end enclosing the chain drive to the timing valve camshaft. It forms the scavenge air receiver and manifold, accepting the charging air from the turbo-blower cooler and distributing it evenly to the inlet ports of the cylinders. It is strongly ribbed internally to provide the strength necessary to withstand the boost pressure, the ribs being carefully placed to give maximum support to the outer walls whilst in no way impeding the flow of air.

As well as containing the scavenge air pressure the entablature also supports the cylinder assemblies. For successful operation the axes of the cylinder liners must be maintained in parallel alignment with the crosshead guide surface. This has been achieved by designing the whole casing enclosing the crankshaft and running gear as a rigid box like structure. The front of this box is closed by a light but strong section; the guide plate forms the back and is firmly secured to the entablature at the top and the bedplate at the bottom, whilst the double walled end units make a significant contribution to the rigidity of the completed assembly.

**Connecting Rods and Bearings**

The centre connecting rod is of circular cross section shaped at its ends to receive the upper half shell of the bottom end bearing and the lower half shell of the top end bearing. It is thus of fixed length between the centres of the two bearings, which are of thin shell type.

The top end bearing is a shell type bearing of the inverted Tee form, as shown in Fig. 5. The lower half of the

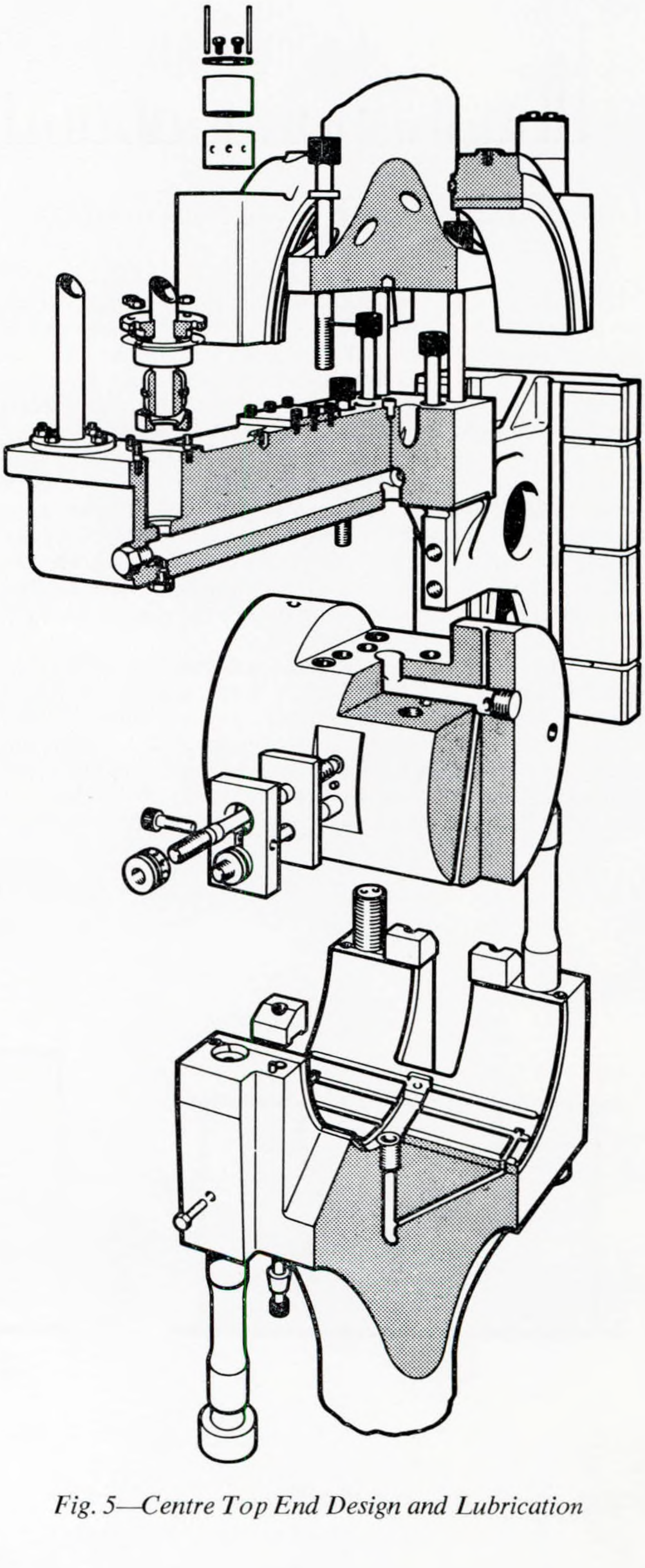


*Fig. 5—Centre Top End Design and Lubrication*

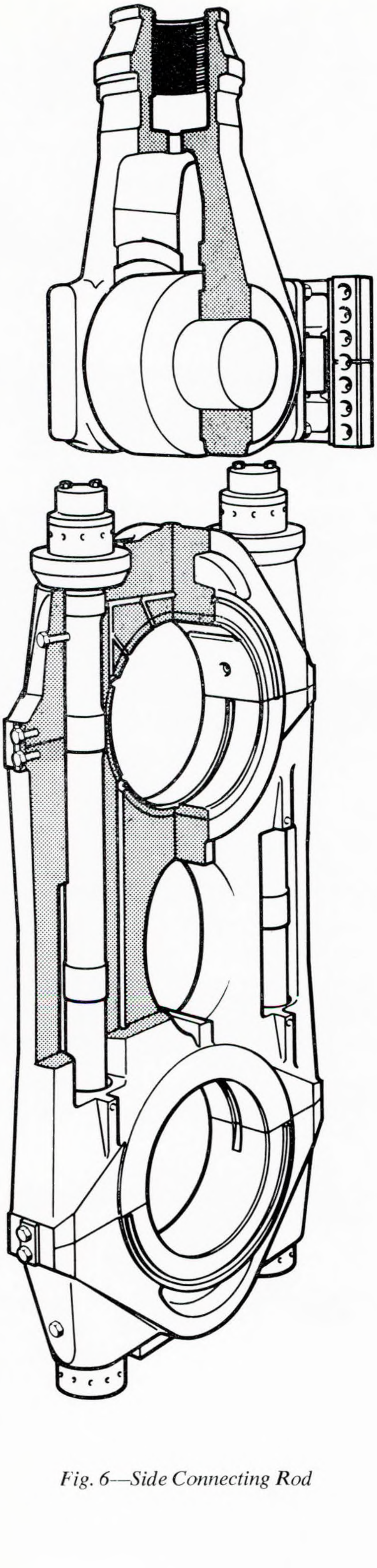


*Fig. 6—Side Connecting Rod*

shell extends for the full length of the gudgeon pin. A narrow half shell secured by a cap is fitted over the top of the crosshead pin at each end. The bearing surface over the full length of the crosshead pin, which is provided by the lower bearing shell, is broken only by the oil supply grooves. Oil is fed from the crosshead bracket through a hole in the pin into a central circumferential groove in the shell. It then flows towards the ends through axial grooves and over the bearing surface. The axial grooves are connected to one circumferential groove near each end of the bearing and from these, through holes in the connecting rod, to the bottom end bearing. The oil supply to the crosshead bracket is by means of telescopic tubes and the inherent pumping action is utilised to give a boost to the lubricating oil pressure at the top end bearing shell as it approaches the inner dead centre, thereby ensuring an adequate oil film in the bearing during the period of maximum load. The forced flow also ensures that the quantity of lubricant is sufficient to keep both bearings cool.

The telescopic tubes on the crosshead bracket are mounted at the opposite side of the crosshead pin to the guide shoes, thereby forming a balanced member free from tilting due to inertia forces.

The crosshead pin is cylindrical, with horizontal and vertical flat faces at the centre where it is located in the centre crosshead bracket. These are designed to permit firm attachment without distorting the upper surface of the bracket where it receives the palm end of the piston rod.

The loads on the side connecting rods are almost entirely tensile and the design which was successfully developed for the Seahorse engine is used for these rods. It is shown in Fig. 6. The essential parts of the design are a strong keep above the top end pin, a strong keep below the crank pin and tensile members between the two to carry the major load. Inner half bearings are needed to retain the oil and have to be separated by a relatively light member which can also carry the small compressive load. In practice this member is made as a rigid steel casting so that shell type bearings can be used and their shape maintained by the stiffness of the casting. The design provides a stiff abutment to reduce the stress range in the bolts and at the same time provides lateral flexibility, so that the bottom end bearings can accommodate to the defections of the crankshaft when running.

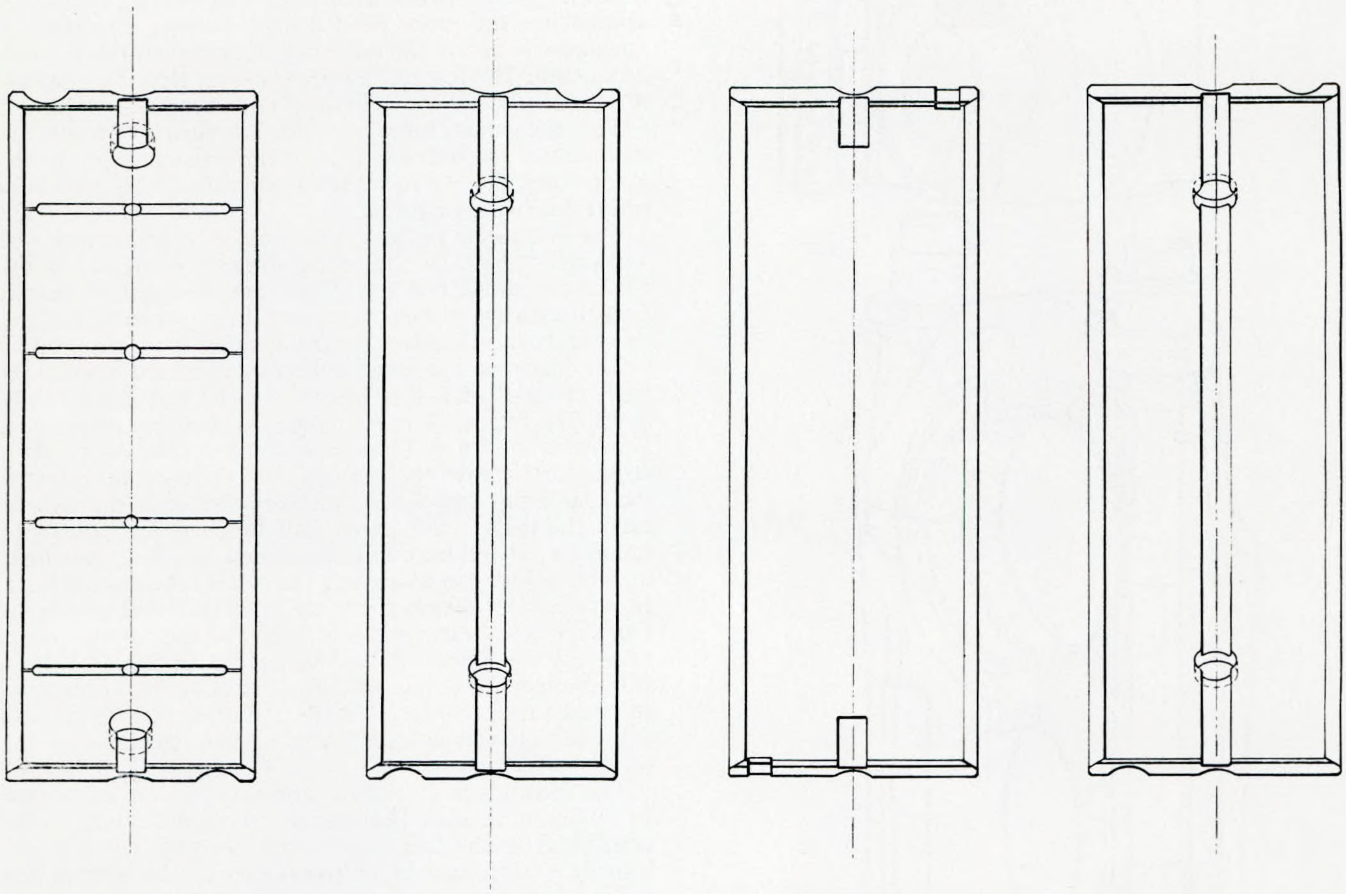
The bearings in the side connecting rod are lubricated by oil fed to the main bearings and through drillings in the crankshaft to the side bottom end bearing. The top end bearing is lubricated by oil passing round the bottom end bearing and through a passage in the rod.

Top and bottom end bearings of the side connecting rod are of the shell type and nominally of the same size. The loaded half of the top end shells is profiled in the manner shown in Fig. 7 and has formed in it axial grooves to ensure adequate lubrication under oscillating conditions. The bottom end bearing has only a circumferential central groove for conveying oil and is lubricated in the normal manner for a hydro dynamic rotating bearing, by oil passing from the central groove to the outside of the bearing. Locating buttons for the bearings ensure that they cannot inadvertently be placed in the wrong position. Both centre and side crosshead pins are made of nitrided steel and hardened.

**Piston Design**

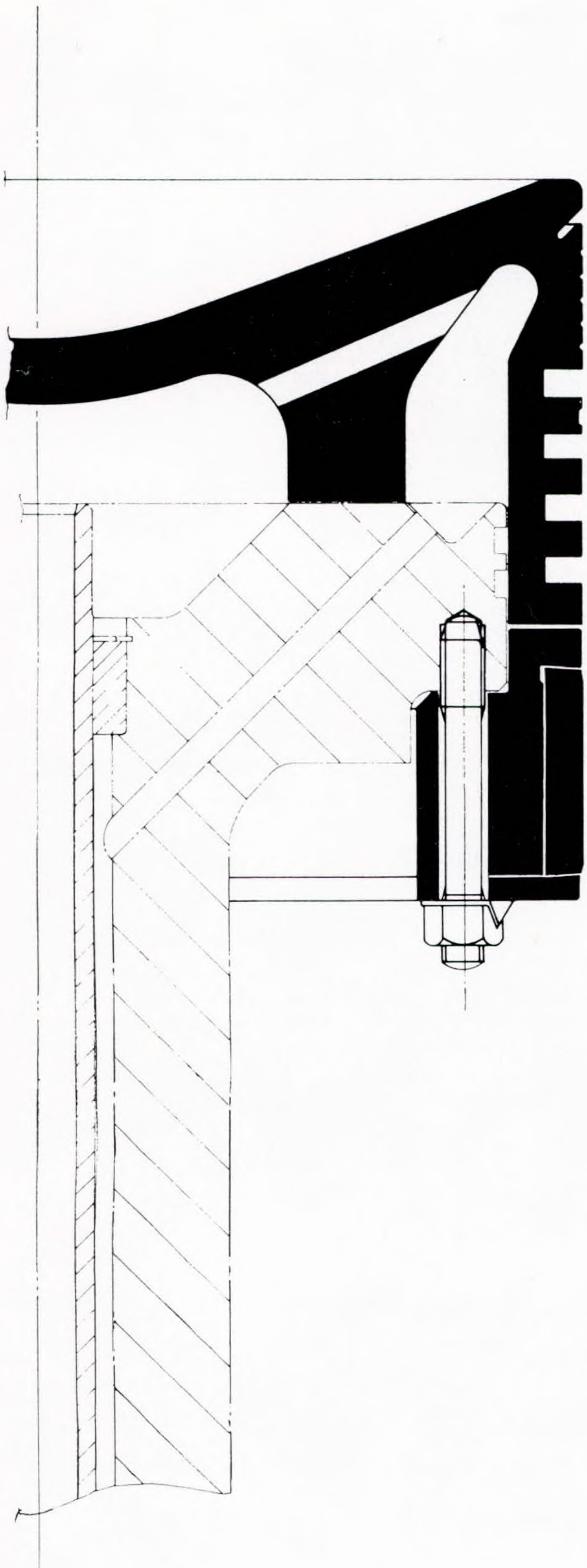
The 580 mm diameter piston, first used in the Seahorse, has undergone considerable development and progressed towards a new type now used in the 58JS3 engine. Examples of the two designs appear in Fig. 8.

On the Seahorse engine, difficulties were experienced in obtaining a rapid run-in of the firing piston rings. Ring behaviour was the subject of a good deal of critical study and Butler and Henshall described the development of a pivotted ring which could accommodate itself to changes

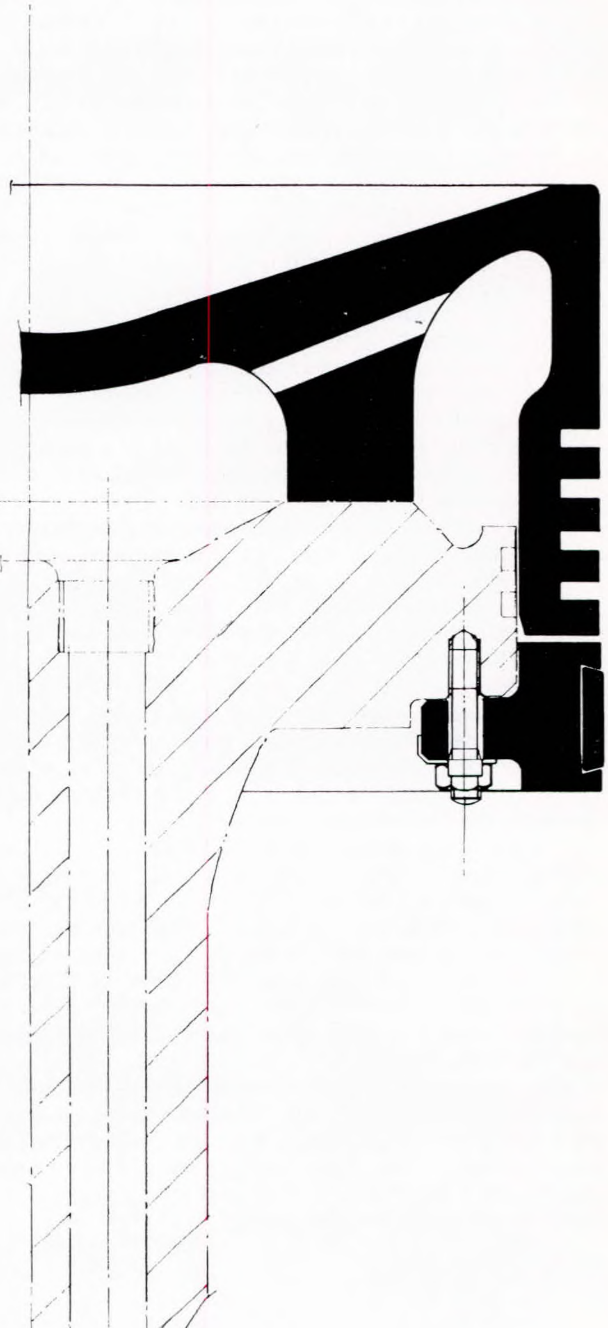


*Top End Upper Half* | *Top End Lower Half* | *Bottom End Lower Half* | *Bottom End Upper Half*

*Fig. 7—Side Connecting Rod Bearings*



Seahorse Engine



58JS Engine

*Fig. 8—Piston Design*

of ring groove inclination[3]. Simultaneously it was realised that the ring would have a much easier life if the piston could be designed so that the groove did not alter its attitude so much for various conditions of load. A number of exercises were carried out on Seahorse piston designs, using finite element methods, in order to establish the displaced shape of pistons under mechanical and thermal load and to predict the temperature and stress distribution. Because experimental work on Seahorse did not continue, practical tests of a new design of piston, based on this work, was carried out on 760 mm bore 'J' engines and proved most satisfactory.

As can be seen from Fig. 8, the new design features a relatively thin and flexible crown designed to displace the walls of the piston the minimum amount. The connection between the crown and the ring belt is also very flexible and the ring belt itself is further from the crown to accommodate this flexible connection. This increased dimension between crown and ring belt also results in a lower temperature for the first ring groove. The desired object of bringing about less change in the angle of a groove for a given change of load was attained. Pistons of this design were, therefore, used in the 58JS3 engine.

Fig. 9 shows the predicted temperature distribution and, for comparison, actual readings of piston temperature for upper and lower pistons taken by Shell templugs.

This development of the piston was sufficiently successful to avoid the need for complex ring types and the rings now used are plain cast-iron rings, functioning as gas sealing rings in the first three grooves, with a simple stepped scraper acting as an oil distribution ring in the fourth groove. The scraper action of this ring is arranged in an inward passing sense, directing the metered feed of cylinder oil towards the other rings, to lubricate them on its way to the combustion chamber.

Piston and piston ring performance have been excellent. During the very early life of the engine, because of difficulties in getting the turbocharger to self-support, there were many occasions on which the engine was grossly over-fuelled and ran with extremely high exhaust temperatures, at a time when the rings could not be expected to have bedded in. Nevertheless, whenever they were examined they were found to be in perfect condition, with no sign whatever of scuffing.

After the engine had run about 400 hours, the rings were removed and measured. Fig. 10 shows some examples of the worn profile of the rings. The wear rates, which might be expected to be high during the initial running-in period, are such that a life of over two years can confidently be predicted for top rings.

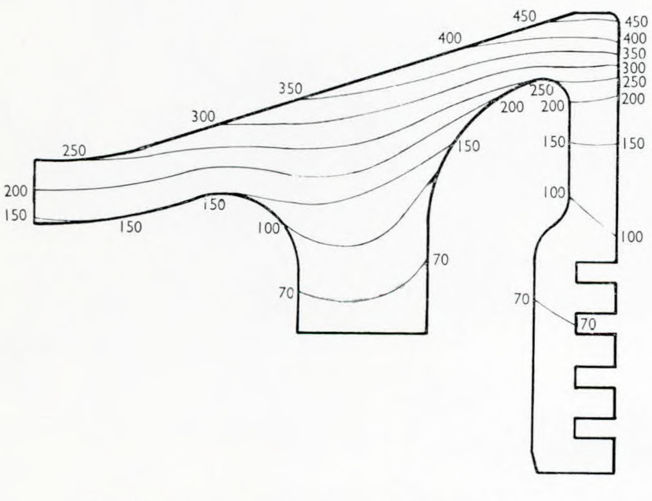



*Fig. 9—Piston Temperature Distribution*

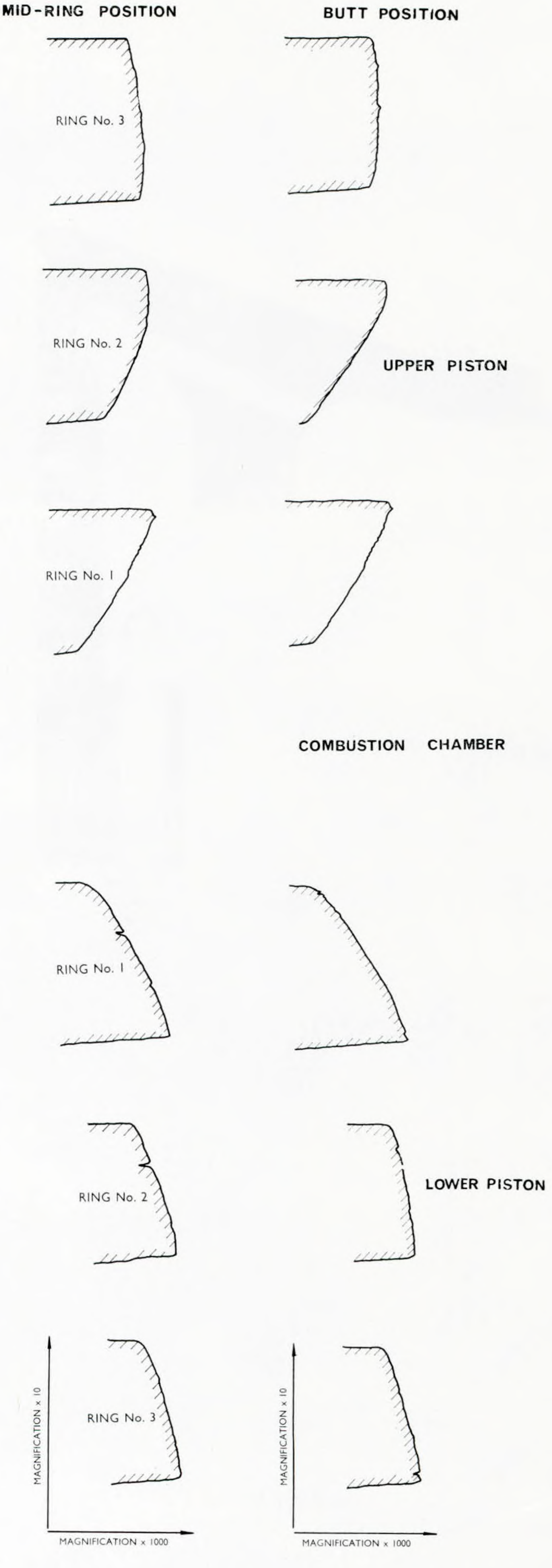



*Fig. 10—Piston Ring Profiles after 388 Hours on Various Loads*

**Timing Valve Characteristics**

In 1955 Doxford introduced their fuel injection system based on hydraulically operated injectors and the injection timing controlled by timing valves. The timing valves were operated by cams on the camshaft, which rotated at engine speed; the cams were made symmetrical for simplicity and this gave satisfactory injection timing, both for the ahead and the astern direction, without resetting the camshaft between these two directions.

Various timing valve designs were produced and Doxford introduced their present design in the late sixties(4). It was of more rigid construction than earlier designs and used a circular arc cam instead of the tangential cam used previously. The geometry of this timing valve was later calculated accurately and a computer programme prepared for subsequent studies. It was found that the roller and lever mechanism could be used to improve the characteristics and, by comparing injector valve lift diagrams and the static settings with the calculated characteristic, good agreement was achieved. Several possible improvements were also discovered.

Fig. 11a gives the calculated timing value characteristics of the 76J4 constant pressure turbocharged engine. The shape of the curve is determined by the cam and the roller-lever mechanism. The horizontal position of the curve is determined by the cam peak clearance, which should be at least 0.15 mm for safety. The vertical position is determined by the setting of the cams relative to inner dead centre of the corresponding cylinder. This position is again controlled by the maximum pressure in the cylinder, as well as the required injection timing for starting and slow running. For the 76J4 engine it was necessary to use a timing valve setting of 7 degrees before inner dead centre to 18½ degrees after inner dead centre in order to obtain a satisfactory maximum cylinder pressure at full power. This resulted in 0 degrees b.i.d.c. to 8 degrees a.i.d.c. at the starting position and a tipping position for dead slow of 1-3 degrees a.i.d.c. These latter two conditions were satisfactory, but it would have been of advantage to advance the injection by up to 3 degrees for light load. The 76J4 engine was designed for pmax=85 bar b.m.e.p.=10.87 bar and a scavenge pressure of 2.1 bar (abs). Even when constant pressure turbocharging is used there will be a certain build up of pressure in the exhaust manifold, due to the blow-down of one cylinder, when another cylinder closes its exhaust ports. This results in a relatively high pressure in the latter cylinder at the beginning of compression and a relatively late fuel injection must be used in order to achieve a normal maximum pressure for the power.

Traditionally the Doxford engines were built with the camshaft on the starboard side of the engine and this practice was retained for the 76J4 constant pressure turbocharged engines. For the 58JS3 engine it was, however, decided to position the camshaft on the port side, in accordance with the practice adopted by most builders of two-stroke marine engines. Thus the timing valves were mounted on the opposite side of the camshaft and the characteristics shown in Fig. 11b were obtained. With these characteristics the start of injection is retarded less when the power is reduced from 100% to slow running and it is therefore potentially better. In a three cylinder engine there is little back-charging of the cylinders from the exhaust manifold. The first 58JS3 engines were set for a power of 5000 b.h.p., b.m.e.p.=10.38 bar and pmax=80 bar, which gave a static timing valve setting of 11 de-

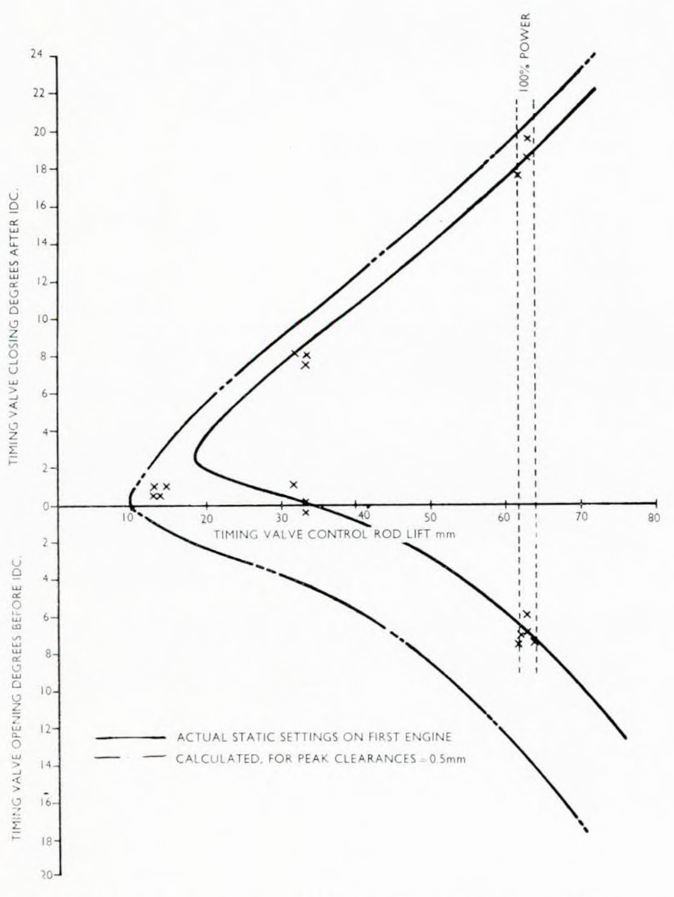



*Fig. 11(a)—Timing Valve Characteristics of 76J4 Constant Pressure Engine*

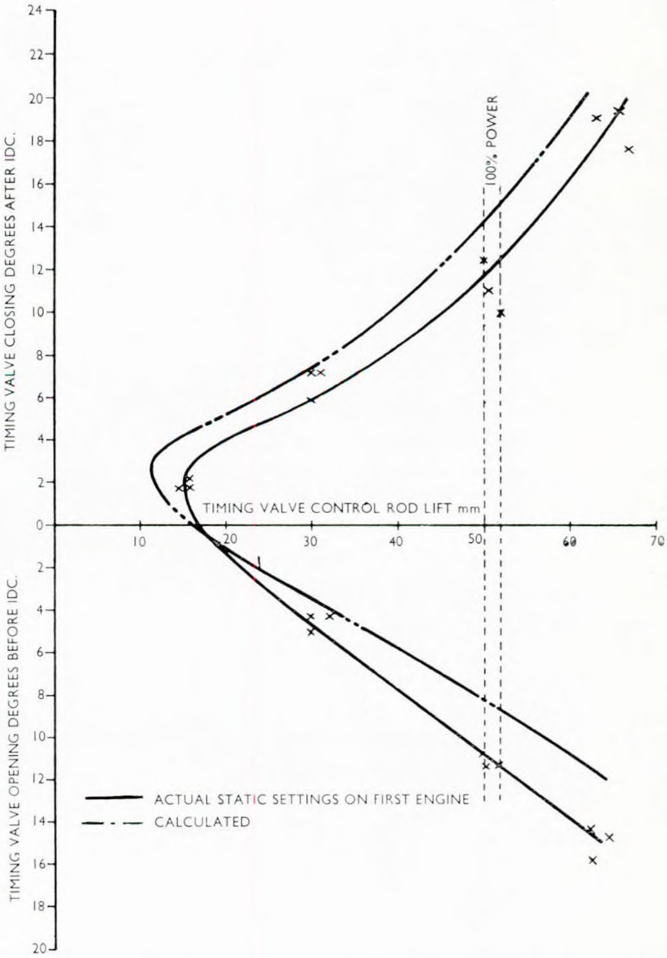



*Fig. 11(b)—Timing Valve Characteristics of 58JS3 Engine*

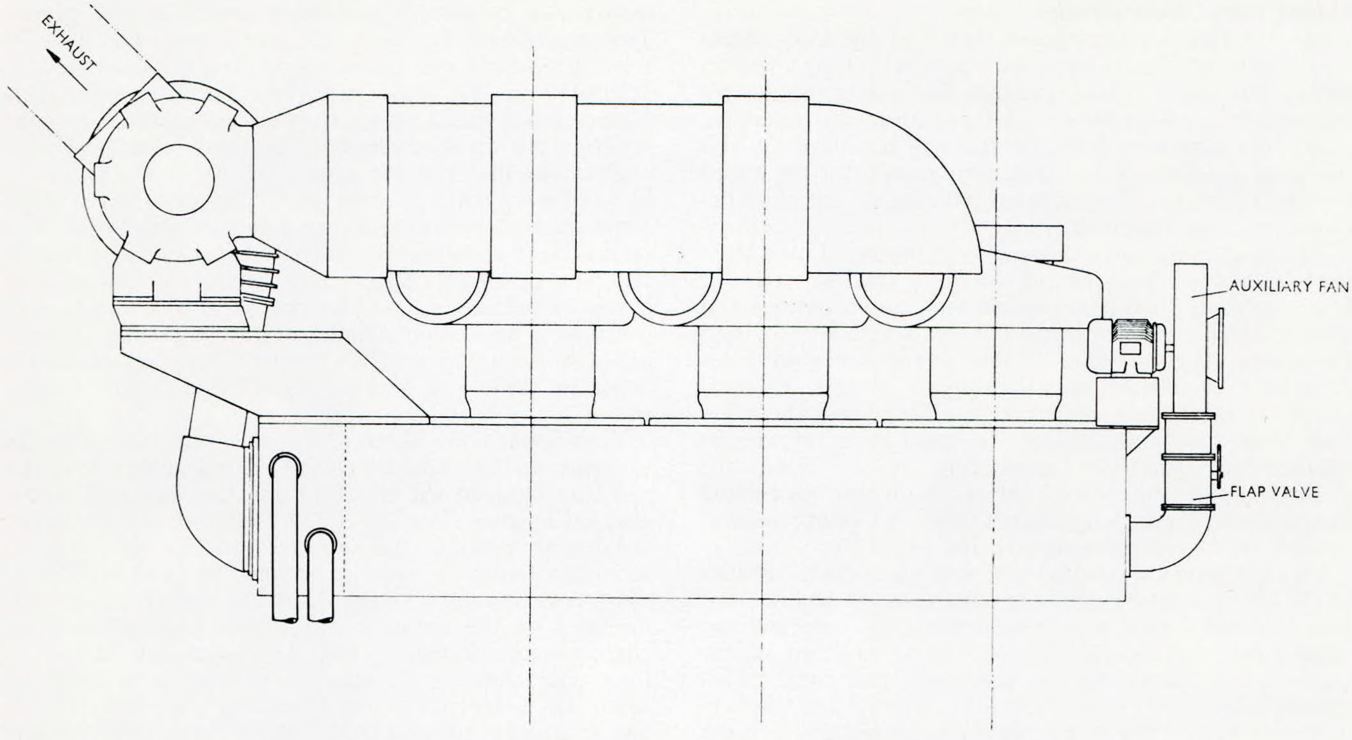



*Fig. 12a—Turbocharger System with Auxiliary Fan in Parallel*

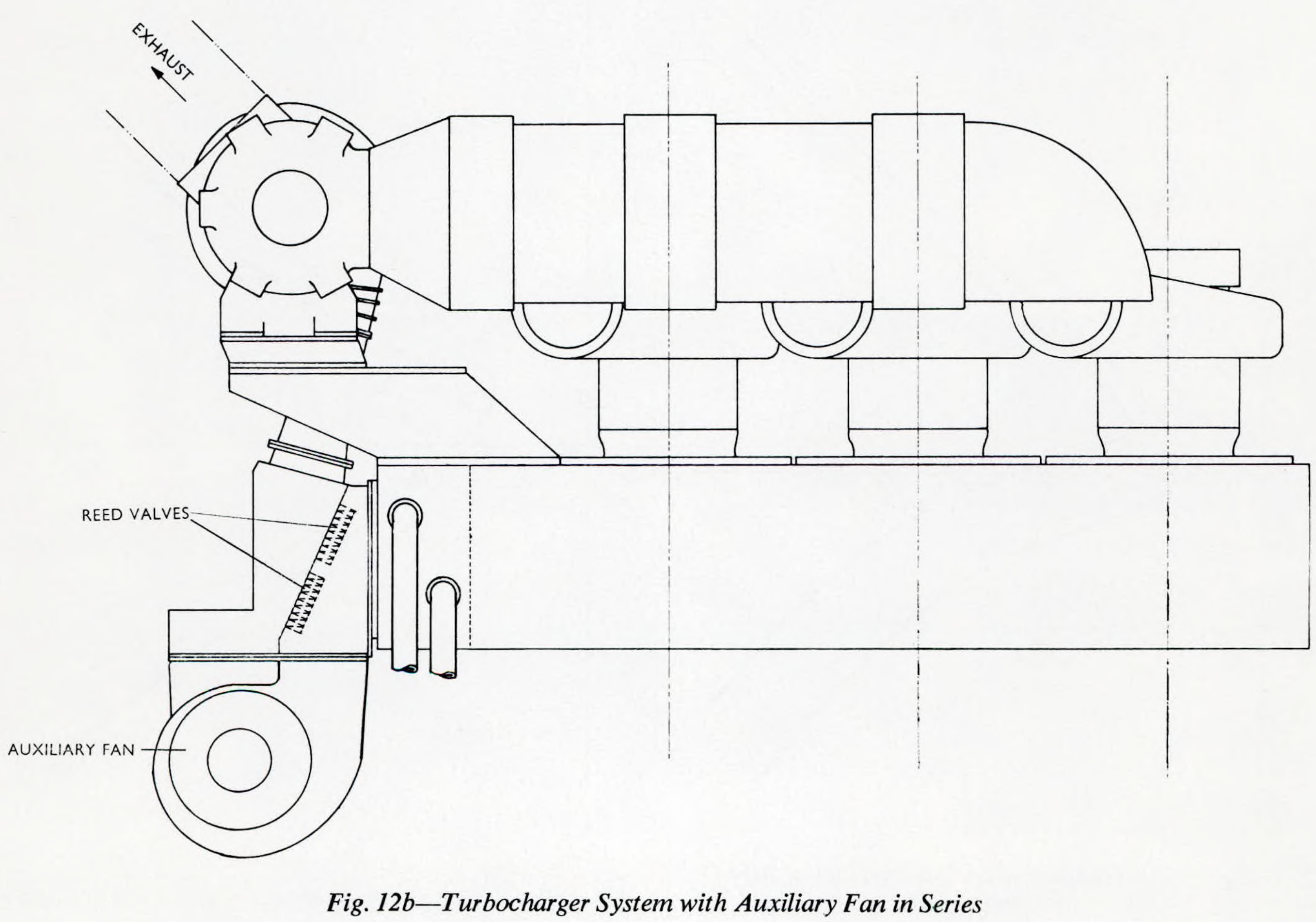



*Fig. 12b—Turbocharger System with Auxiliary Fan in Series*

grees b.i.d.c. to 10.5 a.i.d.c. At the starting position the timing is $4\frac{1}{2}$ degrees b.i.d.c. to 6 degrees a.i.d.c. Unfortunately, the actual settings on the first engine were not as good as the calculated curve, probably due to inaccuracies in the cams.

**Turbocharging System**

The turbocharging system of the first 58JS3 engine is shown in Fig. 12. A single Napier NA550 turbocharger is used, driven by exhaust gas from the large volume constant pressure exhaust manifold. A turbine protection grid is located at the end of the manifold in the large diameter section; its passage area being about eleven times the turbine nozzle area.

In the engine as originally built, air was delivered by the turbocharger compressor to a collecting box, from which it passed through a charge cooler to the scavenge air chest formed inside the entablature. For starting and running at low powers an auxiliary fan was provided. This was situated at the opposite end of the entablature and delivered air into the scavenge air chest through a controlled flap valve, as shown in Fig. 12a. Later the fan was

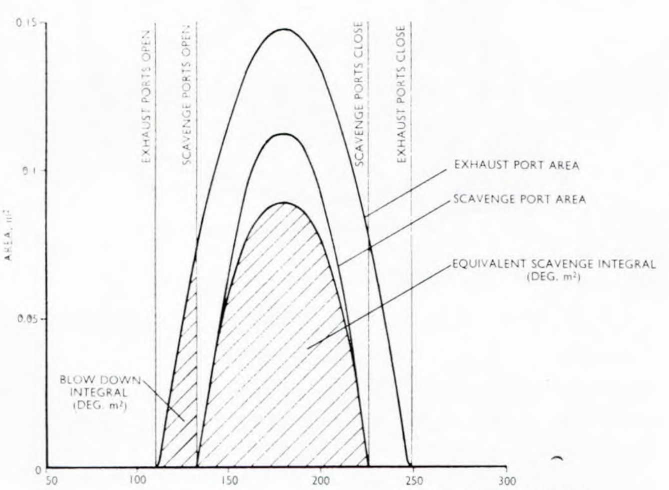



*Fig. 13a—Port Area Diagram*

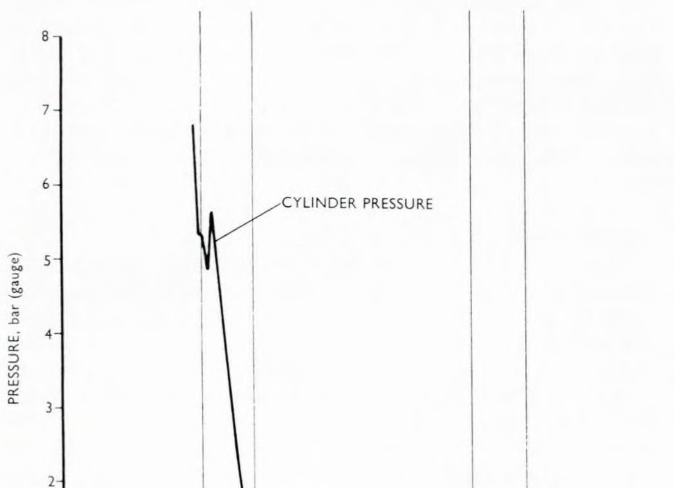



*Fig. 13b—Light Spring Diagram*

moved to the after end of the engine and arranged to work in series with the turbocharger, as shown in Fig. 12b.

The reasons for changing from pulse turbocharging to constant pressure turbocharging for the J-engines was described by Ørbeck[5]. To achieve satisfactory scavenge and combustion the Doxford engines require an air flow of approximately 9.5 kg/kW.hr., which in the 58JS3 engine at 5000 b.h.p. (3680 kW.) is pumped up to a pressure of 1.1 bar gauge. The corresponding shaft power in the turbocharger is approximately 1000 kW., or in other words one quarter of the engine output. The power to drive the turbine is taken from the exhaust gas. If, therefore, less pumping power for some reason would be required, more power could be removed from the exhaust gas in the cylinders and a lower specific fuel consumption would be achieved.

The port area diagram for the 58JS3 engine is shown in Fig. 13a. It was anticipated that improvements in the design of the exhaust ports and belt, in conjunction with an overall turbocharger efficiency of 60%, would make it acceptable to increase the expansion stroke from 105-108 degrees used on earlier Doxford engines to 111 degrees. The blow-down area was determined from measurements on earlier engines[6]. Thus the height of the scavenge ports were fixed and the resultant equivalent scavenge area appeared to be satisfactory with regards to resistance to the flow through the cylinder.

Light spring diagrams, which give the pressure in the cylinders during blow-down and scavenge, were taken relatively early during the development trials of the first engine. These differed little between the cylinders and a representative diagram is shown in Fig. 13b. From this it will be seen that the blow-down was very satisfactory. The relatively low pressure in the cylinder at the beginning of compression, referred to under "Timing Valve Characteristics", can also be seen clearly.

During the progress of the engine tests it appeared that the air flow was lower than could be desired. A re-examination of the flow path of the air and exhaust gas through the engine showed that the velocity of the exhaust gas through the piston ring grid was 96 m/sec. A double grid was used and this was placed close to the turbine inlet. It is likely that most of the velocity head through the grid was lost to friction and it was therefore decided to change to a single grid of larger diameter and mount this as shown in Fig. 12.

The air velocity at the outlet from the compressor was 72 m/sec., which was considerably higher than past experience on the J-engines. With constant pressure turbocharging it is frequently possible to use a frame-size smaller turbocharger, rather than for a pulse turbocharged engine of the same power. Consequently, the compressor outlet will be smaller and, since the air flow is approximately the same, the velocity will be higher. This point was not appreciated during the design stage of the engine and a proper diffusor had not been fitted after the compressor delivery. When the piston ring grid was changed, the turbocharger was raised slightly to obtain a better geometry and a correctly designed diffusor was fitted. Thus at full power a pressure head of .025 bar was recovered from the compressor outlet to the collecting box.

From the collecting box the air passes through the cooler and into the entablature. The entablature serves as a reservoir which feeds air through the cylinders in turn during the scavenge period and it must be of adequate volume to keep the pressure fluctuations within acceptable limits. Too large pressure fluctuations will mean that the compressor will have to be set well away from the surge line and therefore at a low efficiency. However, space limitations set a practical limit to the volume of the entablature. Returning to Fig. 13a it will be seen that the scavenge ports are open from 134 to 226 degrees after inner dead centre. Since the crank angle between each cylinder is

120 degrees, there will be three periods during which no cylinder is open during each revolution. In a three cylinder engine there will, consequently, be a relatively uneven outflow of air from the entablature. The volume of the entablature in the 58JS3 engine is 6.3 times the swept volume of one cylinder and it was necessary to set the compressor at a slightly lower than normal efficiency to avoid surge. Nevertheless, a very satisfactory overall turbocharging performance was finally achieved. This was largely due to the new exhaust belt design, which will now be described.

**Exhaust Belt Design**

It is convenient on Doxford engines to run the exhaust pipes along the back of the engine, as this keeps them away from the camshaft side and gives good accessibility. The exhaust gas from the front of the engine must, therefore, be carried between the side rods. In the exhaust belt used on the larger bore 'J' engines between 1968 and 1977 the gas passing through the exhaust ports at the camshaft side of the engine was brought past the side rods by two passages, through separate bolted-on ducts, which embraced the side rods and were designed so that the duct from one cylinder passed over the duct from the adjacent cylinder[5].

Fig. 14a shows the arrangement adopted on the Seahorse engine. Here, no attempt was made to pass the side rods and twin exhaust manifolds were used, one at each side of the cylinders, which were intended to go to two turbochargers. This arrangement gave an excellent gas flow path, but, because of the necessity to lift the pistons over one or other of the exhaust manifolds, it led to a high

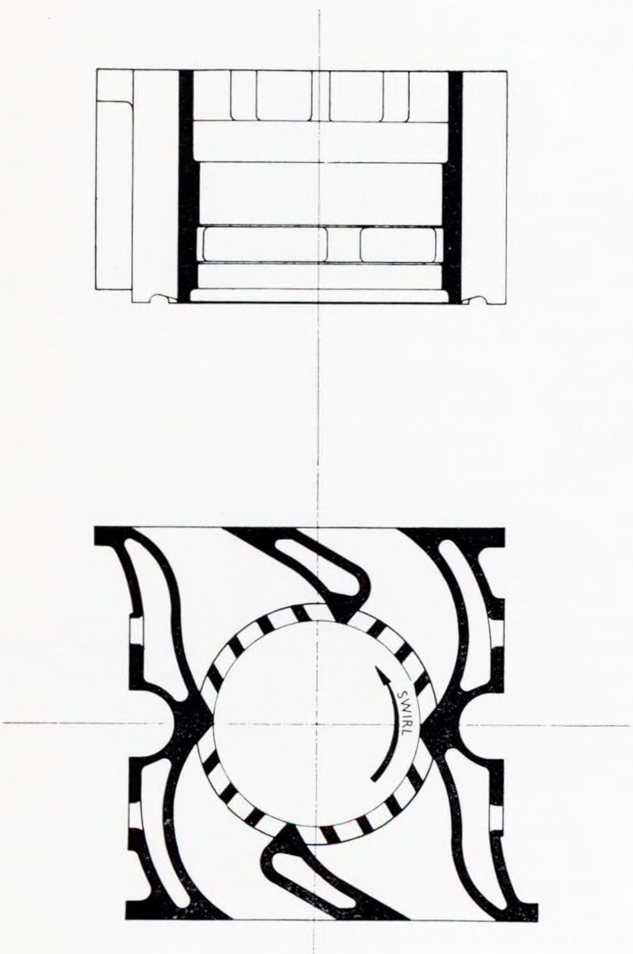



*Fig. 14(a)—Exhaust Belt Design in Seahorse Engine*

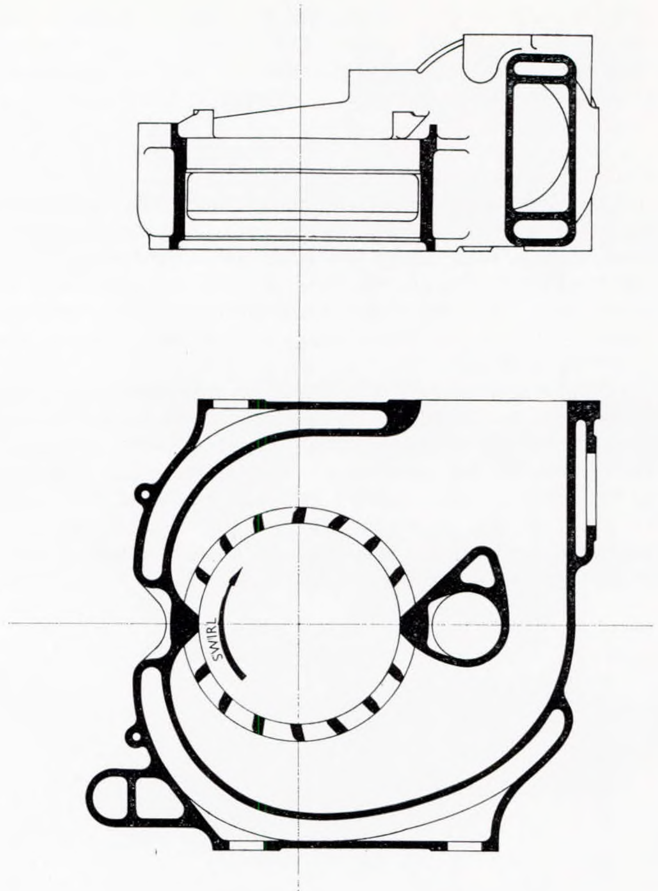



*Fig. 14(b)—Exhaust Belt Design in 58JS3 Engines*

overhauling requirement. This was not a disadvantage in Seahorse, as the engine was high powered in relation to its height and was intended for big ships.

The 58JS3 engine, with its requirement of compact dimensions and particulars of low overhead height, had to follow the standard Doxford arrangement, with the exhaust on one side. The design of exhaust belt shown in Fig. 14b was the result of studies on all sizes of engine and this form has been adopted, not only for the 58JS range, but also for the 76J range of engines. The critical section area on the centre line of the side rods is now 88% of the fully open port area, compared with 70% for the old design. The second critical section at the joint between the exhaust belt of the pipe is now 97% of the fully open port area for the new design, compared with 68% for the old design.

**Auxiliary Fan**

The Doxford J-engines, which were pulse-turbocharged, were able to start and build up to power without the assistance of an auxiliary fan. A relatively small electrically driven fan was, however, fitted in parallel with the turbocharger, to provide improved acceleration of the engine and a safeguard against fall off due to fouling and wear in service. With the loss of the pulse energy, constant pressure turbocharging systems give a poorer performance of the engine at light load. It was therefore necessary to pay more attention to the size and the arrangement of the auxiliary blower.

The 58JS3 engine was originally fitted with a 15 kW auxiliary fan arranged in parallel with the turbocharger, as shown in Fig. 12a. At the beginning of the test bed trials the motor, which was rated at 60 c/sec., was connected to the standard 50 c/sec. supply. Attempts to bring the engine up to load failed, as the required entablature pressures were above the capability of the auxiliary fan and, with the fan switched off, the turbocharger and engine combination was unable to self-support. These early attempts were made before the combustion system had received any development and gross over-fuelling often took place in an endeavour to obtain sufficient energy in the exhaust to reach a condition under which the turbocharger would respond by providing more air. At this time the engine was not run in.

In order that running-in could proceed, a temporary arrangement was rigged. A 33 kW auxiliary fan was arranged in series with an upstream of the turbocharger compressor. It delivered the air through a controlled flap valve. Downstream of this valve were two air intake apertures, which could be opened or closed by hand. When increasing load a point was reached at which the turbocharger was able to take over on its own. The apertures were then uncovered, permitting air to be drawn from the atmosphere, the flap valve closed and the auxiliary fan stopped. On reducing load, the fan was first started and allowed time to accelerate up to full speed. The flap valve was then opened and finally the atmospheric air intakes closed. The whole of this assembly was mounted well upstream of the nozzle used for measuring the air flow into the turbocharger, in order to avoid interference with its accuracy.

By this means there was no difficulty in providing assistance to the turbocharger up to engine loads of over 50%. In this region the limit of flow capacity of the fan was reached, but the turbocharger was safely into its self-supporting condition.

Work could now proceed to run the engine in, develop the combustion to a satisfactory state and match the turbocharger correctly. After this had progressed to a satisfactory state and a 60 c/sec. electrical supply had been provided, it was found that the auxiliary fan arranged in parallel as originally designed was adequate for test bed purposes. In other words the auxiliary fan could support the engine up to a high enough load for the turbocharger to take over successfully on its own. However, the characteristics of the fan did not leave any margin for operating conditions at sea and two courses of development were, therefore, pursued: one was to investigate the required characteristics of a more suitable fan for the original parallel arrangement and the other was to embody a series fan arrangement into the engine design.

The characteristic of the original auxiliary fan mounted in parallel with the turbo-compressor is shown in Fig. 15a. When the engine ran at 'dead slow' and the turbocharger contributed very little to the air flow, the fan operated approximately at Point A. With increasing load on the

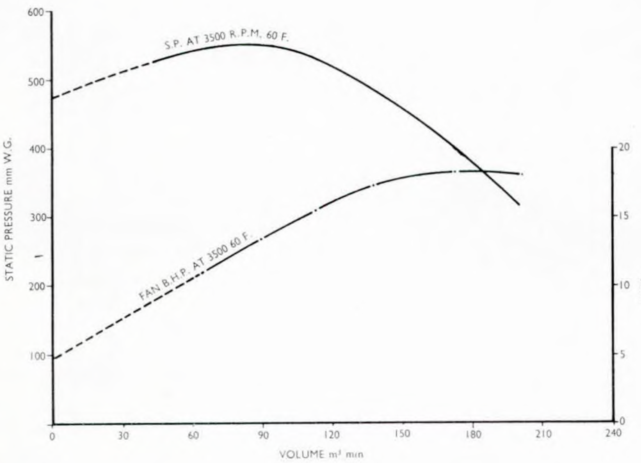



*Fig. 15a—58JS3 Characteristics of 20 h.p. Fan Mounted in Parallel*

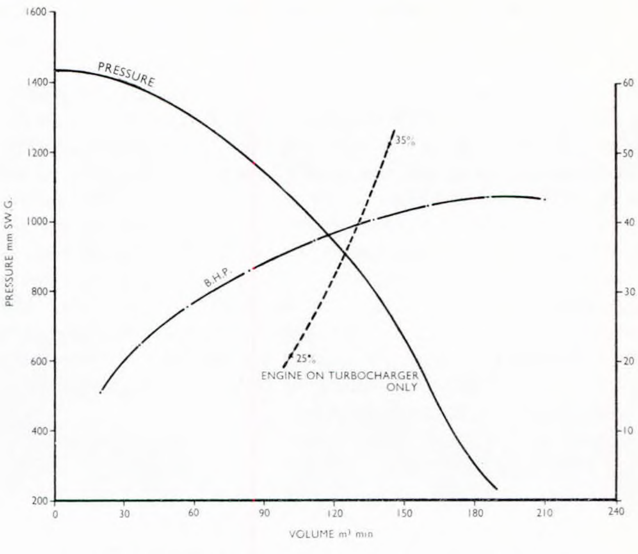



*Fig. 15b—Characteristics of Proposed Fan Mounted in Parallel*

engine the turbocharger provided more air and the pressure in the entablature increased. The auxiliary fan, which sensed a higher back pressure, delivered less air, i.e. it travelled backwards on the operating characteristic from Point A. The operation of the fan was limited by surging. Just before this took place the pressure in the entablature was given by Point B, which is near the maximum pressure on the curve. It was therefore concluded that the fan goes into surge when the back pressure becomes higher than the maximum point on the characteristics.

Tests carried out with no assistance from the auxiliary fan showed that the engine would pick up satisfactorily from entablature pressures down to 0.09 bar. The desired auxiliary blower characteristics should therefore provide for back pressures higher than this. A second requirement from the auxiliary fan was that the engine should be capable of 40% power with the turbocharger out of action. With the original fan only 15% had been achieved. Other engine builders quote 25% and the requirement of 40% was later lowered, but it was clear that a fan of over twice the power would be required. After some search a fan with the characteristics shown in Fig. 15b was ordered, but its delivery would be too late for the first engine.

There is also a chance that a parallel fan can force the turbocharger into surge. However, this fan arrangement offers no resistance to the main flow of air from the turbocharger and it will therefore be investigated further.

In the meantime the three cylinder engine was provided with a permanent arrangement of a series fan. A fan may be arranged in series either upstream or downstream of the turbocharger. Upstream the fan casing is subjected to only low pressures, but the accommodation of the necessary non-return valves and air filters leads to a bulky arrangement. Downstream the fan casing has to withstand the full delivery pressure from the turbocharger, but no additional intake filters are required and the non-return valves can be arranged neatly.

This arrangement was, therefore, adopted and Fig. 12b shows the final form.

The compressed air from the turbocharger compressor is led to a collecting box by a duct which is arranged to have a diffusing effect. In the box a bank of non-return valves of the reed type is arranged, so that the air passes through them on its way to the charge cooler and the entablature air chest. The auxiliary fan is mounted beneath the box, its intake connected to the space upstream of the bank of valves, and its discharge to the space downstream. At low loads, when the turbocharger cannot supply air at sufficiently high pressure, all the air passes through the auxiliary fan, which raises the pressure to an acceptable level. The non-return valves remain closed and prevent the high pressure air passing back to the intake of the fan. As the turbocharger air pressure rises with increase of load, the fan is called upon to pass larger quantities of air at lower pressures until there is a pressure drop through the fan and the non-return valves open. The power to the fan is then switched off and the fan left to 'windmill' at reduced speed during normal operation of the engine.

The non-return valves are entirely automatic in operation and the system requires simpler controls than the parallel arrangement. On the other hand, there is a pressure drop incurred by the air during its passage through the valves; an energy loss which is ultimately reflected in a slightly higher fuel consumption.

**Starting and Manoeuvring**

The design philosophy adopted to ensure satisfactory starting of the constant pressure turbocharged three cylinder engines was described by Ørbeck[6].

A starting positioner, earlier referred to as a starting assister[6], is provided to move the engine rapidly out of a dead band, if required. This is shown in Fig. 16. It consists of two pneumatic power cylinders (1), one for ahead and the other for astern movements. These cylinders are mounted almost vertically at the forward end of the engine and pivoted at the top, so that a roller at the end of the piston rod can be engaged to a wheel (2) at the forward end of the crankshaft, to turn the shaft the required amount. Each power cylinder is brought into engagement by means of an engagement cylinder (3) acting on the guide (4) for the piston rod in the power cylinder. The power cylinders

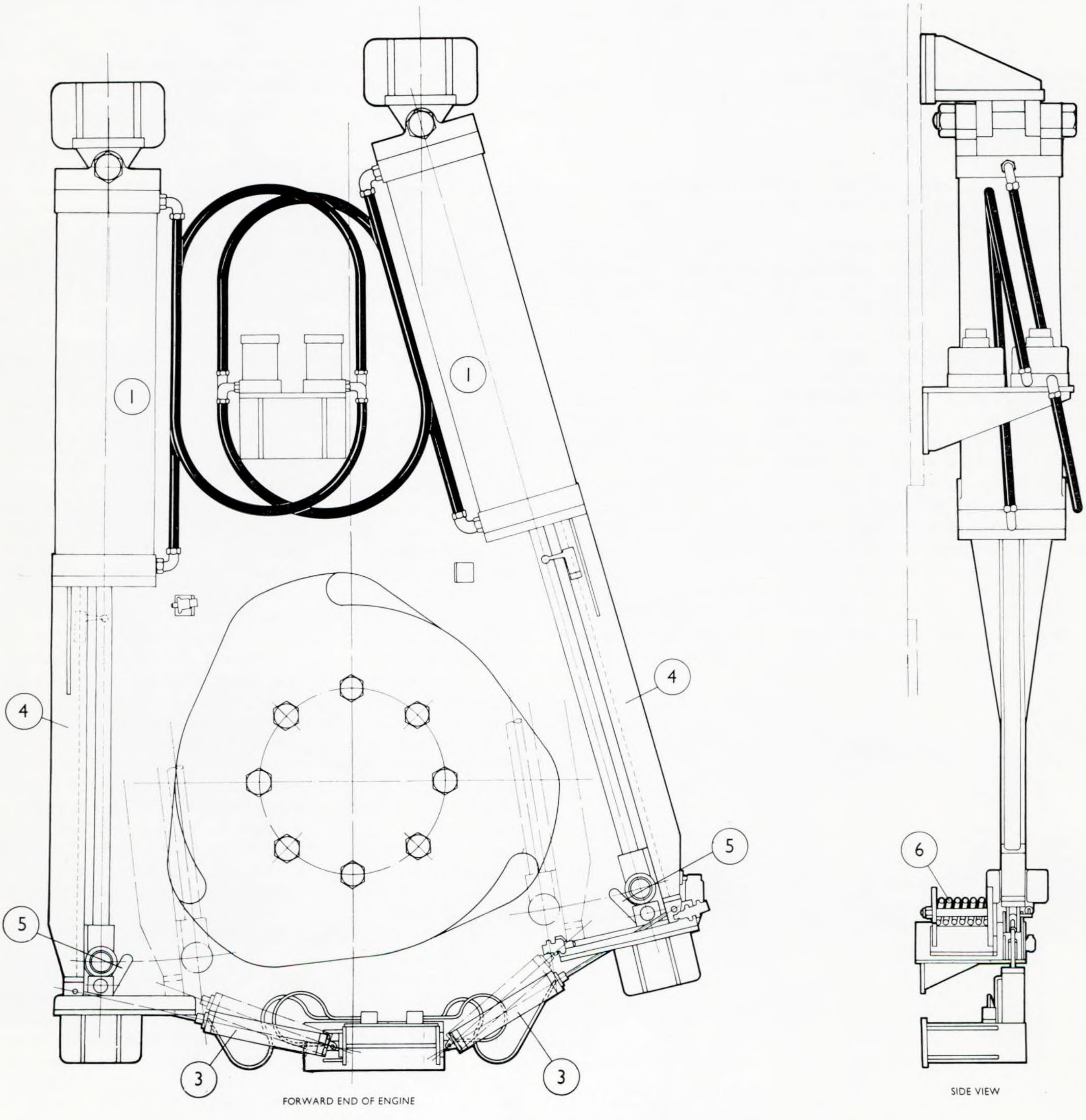



*Fig. 16—Starting Positioner*

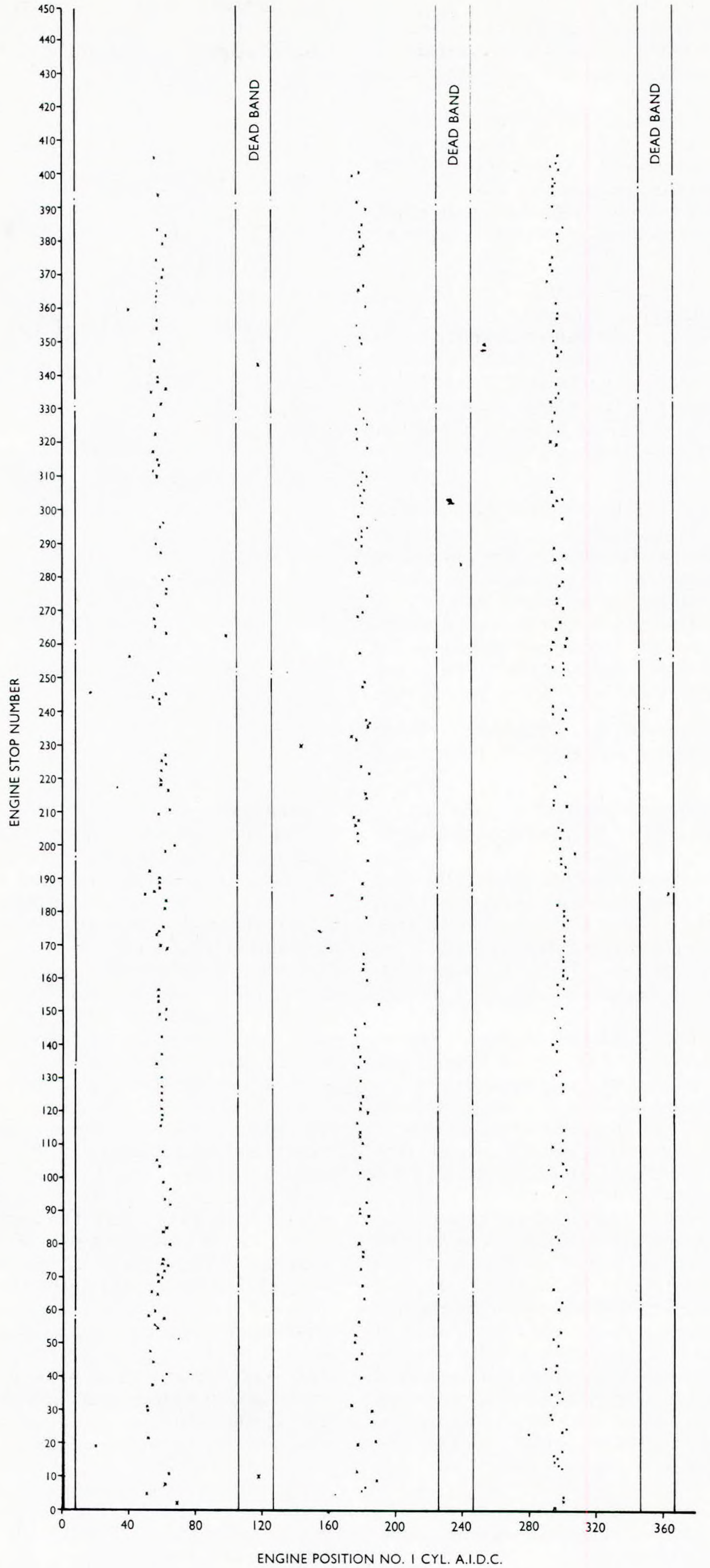



*Fig. 17—Results of Stopping and Starting Tests*

are kept ready for action in the positions shown in Fig. 16. Latches (5), which are operated by springs (6), keep the power cylinders securely in their storage positions. Special plates on the wheel are fitted as a second safety precaution to prevent the ahead power cylinder from engaging in the astern grooves and vice versa.

The 'ahead' or 'astern' mechanism is selected by the air starting lever in the control box. Also, an electrical interlock prevents the starting assister from operating if the engine rotates at more than 5 r.p.m. An electrical distributor operated by the engine camshaft passes the starting signal to an arrangement of solenoid valves to give the following action:

(a) In the intervals from 32° before to 8° after I.D.C. for each cylinder the engagement cylinder for the selected direction will move the power cylinder to its engagement position. The power cylinder will then be given air at a moderate rate until its roller engages with the groove in the wheel. Thereafter the rate of air supply is increased to enable the power cylinder to pull the crankshaft round and out of the dead band.

(b) When the power cylinder has reached the end of its stroke the groove in the wheel has turned round so that it will release the roller easily. The air will be shut off the engagement cylinder and the power cylinder will be returned to its storage position by the return spring. The air will also be turned off the power cylinder after the piston has moved to the bottom end of its stroke pneumatically. The power cylinder is therefore stored ready for its next action.

(c) As soon as the crankshaft has been turned off the dead band the engine will start in the normal manner.

A purely pneumatic system is provided to carry out the above functions, independent of the electrical system, as back-up.

When applied to the 58JS3 engine these predictions were on the pessimistic side. It was shown on the test bed that the engine will start from a position 15 degrees before inner dead centre of any cylinder and also 5 degrees a.i.d.c., but not from positions between these. Thus, the engine has three dead bands from 15 degrees b.i.d.c. to 5 degrees a.i.d.c.

During the test bed trials of the first engine the crankshaft position was noted each time the engine stopped and a record was kept of whether the engine succeeded or failed to start again from that position. The results of these tests are plotted in Fig. 17 and it will be seen that only in about one stop out of a hundred did the engine stop within a deadband. On average it took the starting positioner about 6 sec. to go through its full cycle.

If the above results from the test bed are repeated in board the ship, some considerable simplifications become possible. The starting positioner can be made to act within deadbands of 20 degrees b.i.d.c. to 20 degrees a.i.d.c., i.e. a total of 40 degrees, as before, to provide some overlap for safety. Outside these dead bands the engine will start in either direction. The starting positioner can therefore be made to operate as soon as the engine stops in a dead band and, further, only one side of the present starting positioner will be required, since it is immaterial which way the engine is turned off the dead band. Thus, while stationary, the engine will always be ready to start on one of the main cylinders.

In addition to the conventional pilot air system, the 58JS3 engine is fitted with an electrical system incorporating solenoid valves mounted close to the starting air valves. Tests were carried out to compare the two systems, during the test bed trials of the first engine. The crankshaft was set to the same angular position in both tests and the engine was turned on air. Fig. 18 shows records of the crankshaft speed as a function of the time from the first movement. The superior performance of the electrical system is clearly demonstrated.

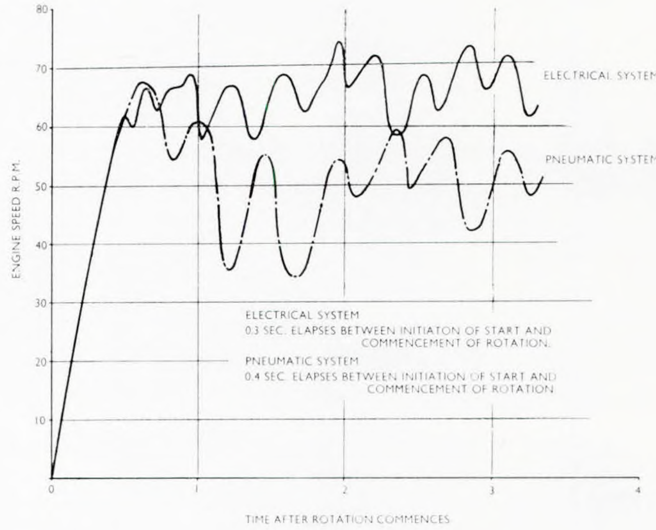



*Fig. 18—Engine Speed on Starting Air*

**Combustion**

The fuel injection system follows standard Doxford practice. Fuel in a common rail is maintained at a constant pressure, which can be controlled according to the running conditions of the engine. A timing valve for each cylinder admits this fuel to injectors and controls both the period and time of injection.

Swirl is imparted to the air when it enters the cylinder through the scavenge ports. A ring of vanes is clamped to the liner round the scavenge ports. The vanes are angled to match the sides of the ports and guide the air into the desired motion, as shown in Fig. 19. This air movement helps to mix the fuel and air for combustion.

The large bore 'J' type engines have two injectors situated diametrically opposite to each other. The Seahorse engine had four injectors at 90° spacing round the combustion chamber. The four injectors were necessary because of the higher crankshaft speed of the Seahorse relative to the air swirl speed. The speed of the 58JS3 engine, at 220 r.p.m., is lower than that of the Seahorse, but higher than that of any other 'J' engine. At the design stage it was realised that better mixing of the fuel and air could be obtained most easily by using four injectors, but, on the other hand, it was regarded as highly desirable to make an attempt to obtain good combustion with two injectors, as there were obvious gains to be made in simplicity of the system and in maintenance. Sections through the middle of the combustion chamber, showing the two injector arrangements (a) and the four injector arrangements (b), are given in Fig. 20.

The engine was, therefore, first built with a two-injector per cylinder fuel system. The injectors used were of Bryce manufacture, U size, but quite short in length, weighing only 6 kgs.

For the two injector system each nozzle had to have a comparatively large total hole area and drilling patterns, having 6 or 8 holes, were chosen.

*(a) From Seahorse Engine. Used initially with two injectors and finally with four injectors.*

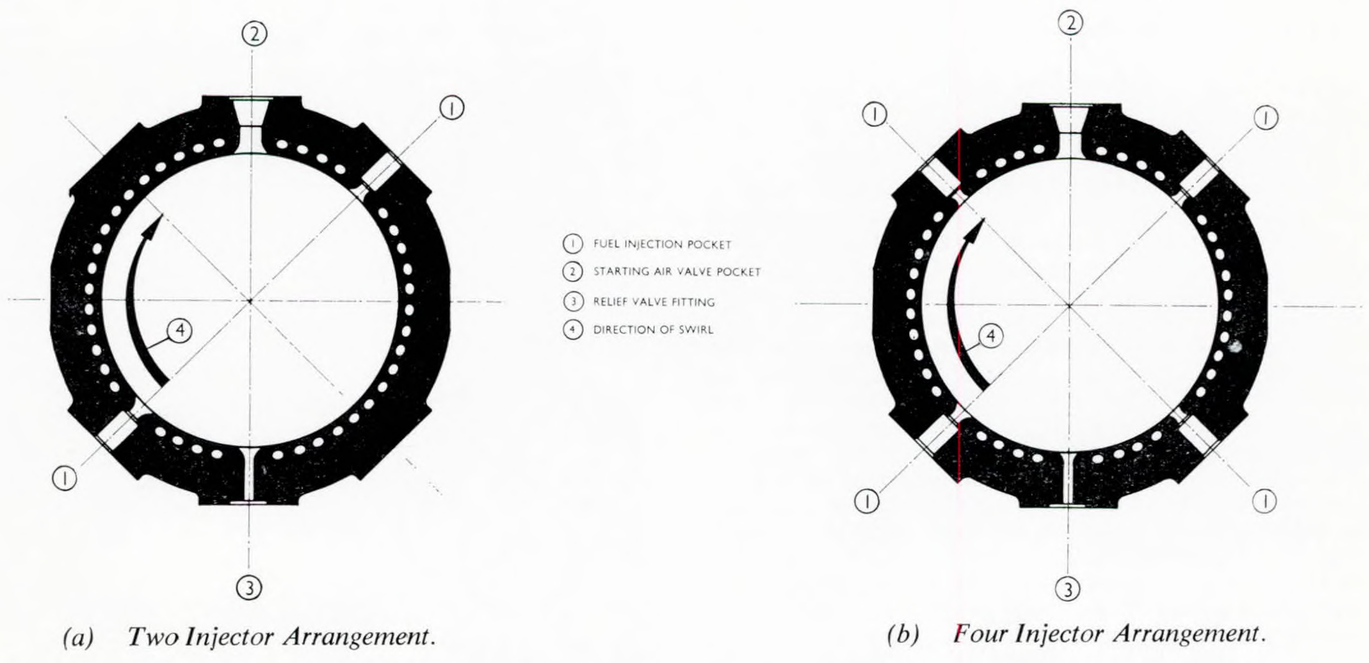
*(b) Modified Scavenge Ports with no Swirl Vanes. This gave best results with two injectors.*

*(c) Modified Scavenge Ports with Swirl Vanes.*

*Fig. 19—Scavenge Ports and Swirl Vanes*

With Doxford engines in the past holes evenly spaced in a regular cone pattern had been very successful, accordingly sets of experimental nozzles were produced having holes similarly positioned. Because of the relatively flat combustion chamber resulting from the short strokes, the included angle of the cones had to be small and values of 30° and 26° were chosen. On the drawing board these narrow regular cone patterns appeared to leave certain zones of the chamber without fuel and alternative nozzle designs were made with an arrangement of holes giving an approximation to an elliptical cone.

The first runs with the two injector system gave rather poor fuel consumption. This was not altogether surprising in view of the fact that the shape of the combustion chamber and the speed of the engine were different from any previous design. Of the nozzles which had been prepared, those with six holes evenly spaced round a regular cone proved to be the best.




*(a) Two Injector Arrangement.*

*(b) Four Injector Arrangement.*

*Fig. 20—Section through the Middle of the Combustion Chamber*

With all the nozzles tried there were signs of impingement of some of the sprays on the crown of the piston towards the outer edge of the combustion chamber. In an endeavour to investigate whether the impingement was an important factor, pistons having a different shape for the combustion chamber face were tried. With these experimental pistons the combustion chamber was changed from the familiar double round nosed cone shape to a lenticular shape, each dish being part of a sphere. There was no impingement, but the fuel consumption deteriorated by a considerable amount.

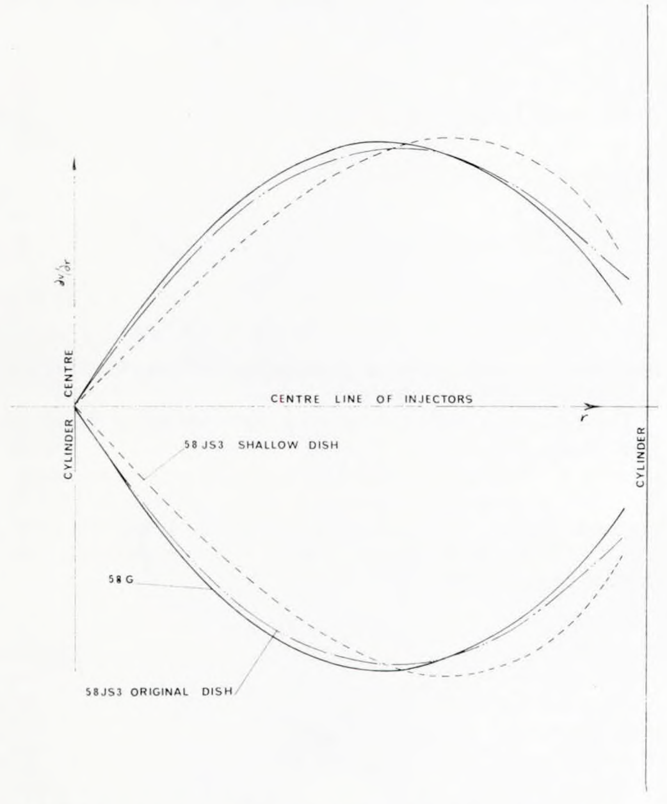



*Fig. 21—Distribution of Combustion Chamber Volume*

Fig. 21 shows the distribution of volume with respect to radius for each of the two pistons, together with Seahorse for comparison, and it is seen that the standard cone shape has much more volume, towards the centre of the combustion chamber, than the lenticular shape. The original pistons were rapidly restored to the engine.

The scavenge port and guide vane configuration originally built into the engine is shown in Fig. 19a. This was almost identical to that which had given the good results on the Seahorse. In view of the discouraging results being obtained with two injectors per cylinder, it was decided to investigate the benefit of changes in the air swirl. A set of liners was modified by opening out the scavenge ports and, at the same time, increasing the swirl angle as shown in Fig. 19b. These liners were tried first without guide vanes and showed a significant improvement in combustion and fuel consumption. Guide vanes were manufactured to match the new swirl angle as shown in Fig. 19c, but when fitted the results were markedly worse than without the swirl vanes.

Estimations of swirl ratio made later indicated that without the guide vanes the swirl was less than the original swirl, as very little guidance was given by the edges of the ports to the air walls and the velocity was lowered. With the guide vanes the swirl was greater than the original. Apparently, when using two injectors per cylinder, the lower swirl ratio was beneficial.

At this stage, it was decided to move to the four-injectors-per-liner configuration and the production of suitable nozzles was immediately put in hand. These nozzles were produced very quickly by a means which is discussed later in this paper and, because of the novelty, it was decided to try a set of them in the two-injector type liners during the time that the four-injector type liners were being prepared.

Each of these injectors had a four hole spray pattern, based on one which had given very good results in the Seahorse. When only two of them were used in each liner the total hole area was, of course, far too small, but they nevertheless gave a surprisingly good performance. Accordingly, the holes were opened out to give a total hole area more nearly that required for the two injector configuration. This gave the best consumption that had been obtained with two injectors so far, i.e. 157 g/h.p.hr. on diesel oil.

An explanation of this can be seen in the distribution of fuel illustrated by the theoretical diagrams on Fig. 22 and

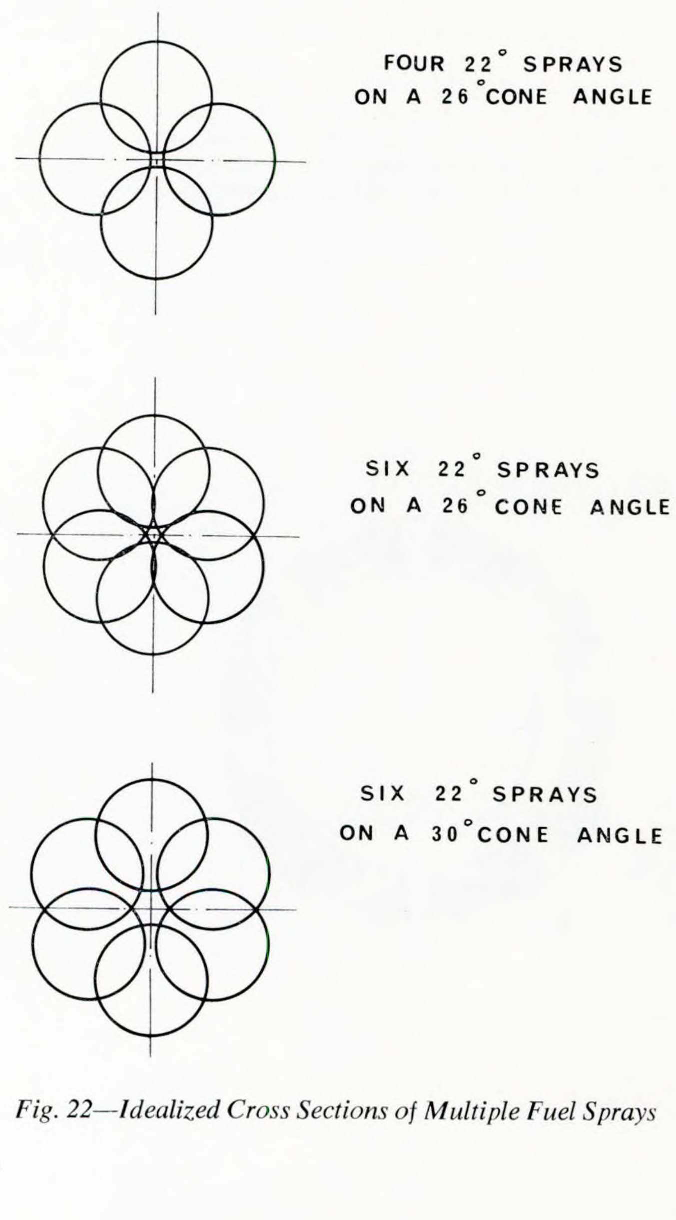



*Fig. 22—Idealized Cross Sections of Multiple Fuel Sprays*

which corresponded closely to actual target tests taken from the various hole patterns. The four hole nozzle has a clear area in between the separate sprays and air can enter the centre of the cone and become entrained with each of the separate sprays all round. The six hole cone does not permit this to happen easily in the case of the 26° included angle cone. There was a tendency, borne out by the practical target test, to form a concentration of fuel in the middle of the cluster with virtually no access for the air.

When liners with the four injector system were fitted to the engine the injector hole pattern chosen had the benefit of the tests taken with the two injector system. Immediately excellent results were obtained; the fuel consumption on diesel oil being well below 148 gr/h.p.hr. at full load and 90% load and remaining below 150 gr/h.p.hr. down to 65% load.

Some effort was put into the construction of a mathematical model that would account for the observed effects due to changes in nozzle configuration, fuel pressure and air swirl. The results did not always explain the effect of change in swirl, but this could well be due to the fact that swirl affects scavenging as well as combustion and the

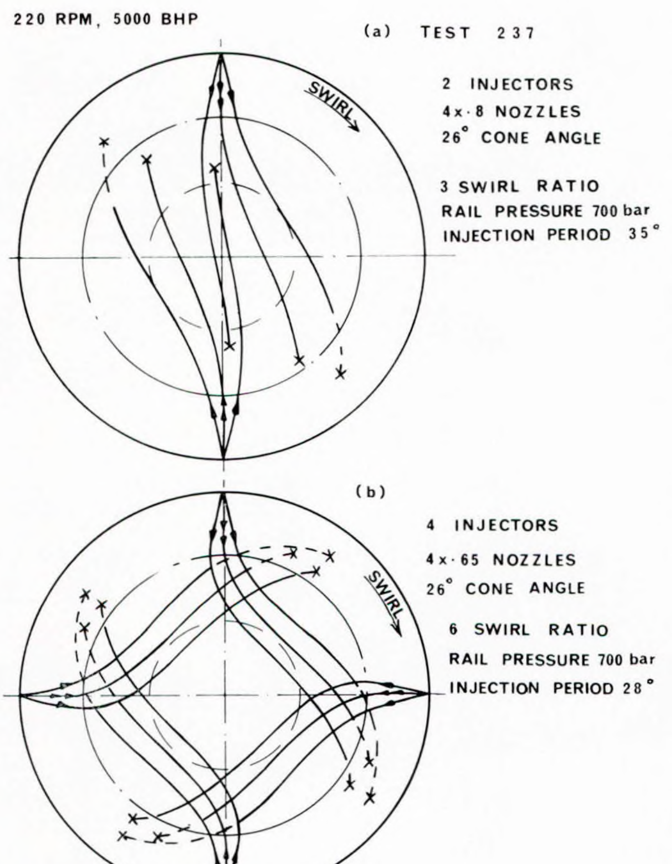



*Fig. 23—Estimated Fuel Spray Paths—Plan View*

model did not take account of this. However, considerable light was thrown upon the various factors in the combustion process and typical results are shown in Figures 23a and 23b, which illustrate the marked difference observed between the best results using two injectors and the much better results using four injectors.

The other diagrams represent a plan view of the combustion chamber and the centre of the spray path is shown by the full lines terminating in a cross at the position reached by the tip of the spray at the end of the injection. When looking at this diagram it is essential to remember the air distribution in the combustion chamber: at inner dead centre, half the air lies outside the radius of 200 mm shown chain-dotted in the figures and only 10% lies within a radius of 105 mm, shown dashed in the figures.

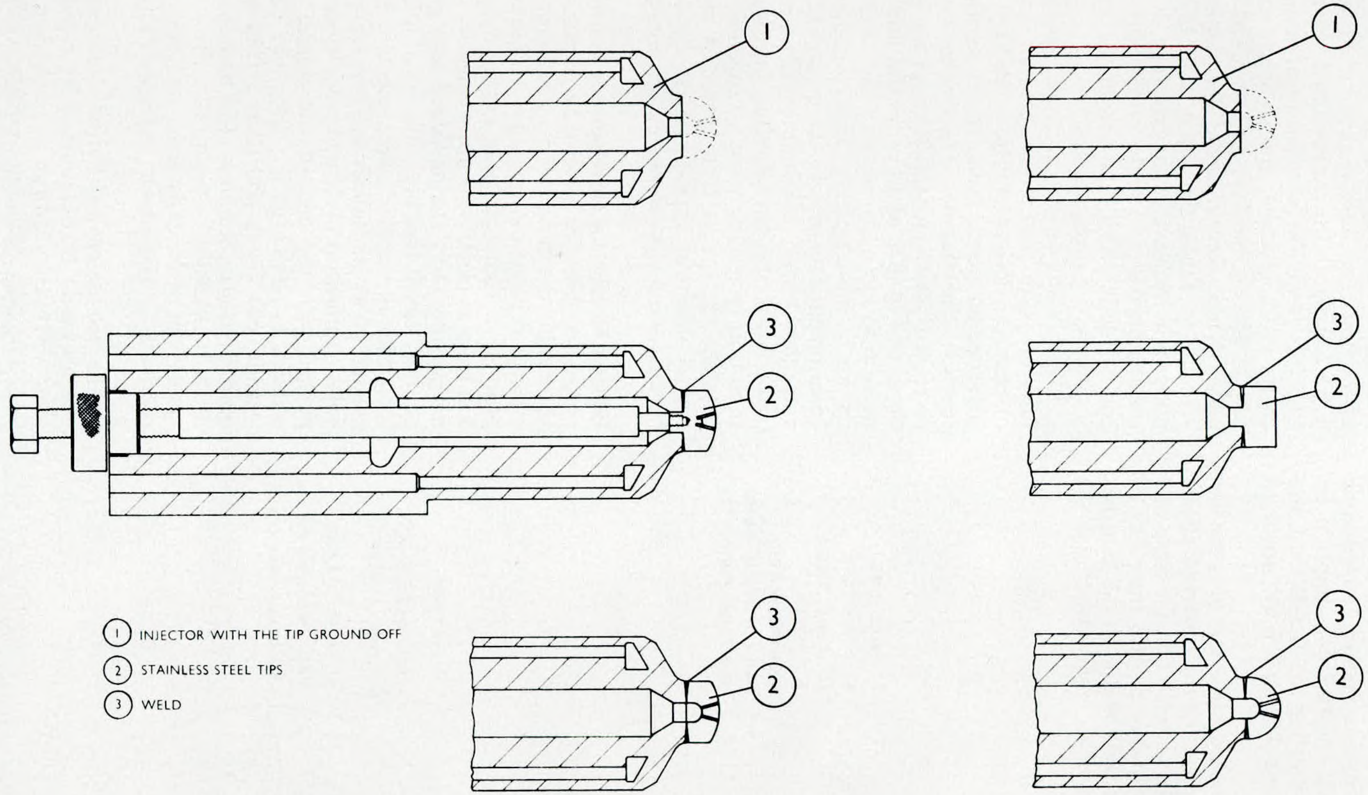
Fig. 23a shows the predicted spray paths for two injectors, each having four holes at a 26° cone angle. Two of the holes are vertical above each other. The swirl ratio is the low figure for the modified scavenge ports shown in Fig. 19b. It will be seen that upstream sprays collide in the centre of the chamber and so are probably not fully effective in mixing the fuel and air. The central sprays pass close to the central area, but apparently with enough velocity to reach the other side of the chamber. It is noticeable, however, that coverage of the outside area of the chamber is poor in the initial stages, although after the end of the injection, the tips of the sprays will be carried round this region by the swirl.

Fig. 23b shows the calculated spray paths for the four injectors with four holes .65 mm diameter in the same orientation. The swirl ratio is the high ratio of the original scavenge air ports. None of the sprays now arrive at the centre of the chamber. The smaller hole diameter and the increased swirl deflect the sprays more rapidly than those of Fig. 23a and the central and downstream sprays move into the outer parts of the chamber, where most of the air is, more quickly. This, together with the increased entrainment rate, due to the greater total number of holes and the better circumferential coverage, presumably account for the considerable improvement in fuel consumption given by this arrangement. The interference between the sprays from adjacent injectors towards the end of injection appears not to be detrimental, perhaps because the fuel in the spray tips has already burned.

**Injector Nozzles for Development Testing**

Anyone who has experience of engine development will appreciate the difficulty of obtaining, quickly, modified injector drilling patterns. To overcome this development problem a technique was explored in which a relatively soft tip was electron beam welded to the remainder of the nozzle, including the hardened and ground seat. Injector nozzles, which had been discarded after experiment as no longer being required, were selected. Each one had the tip ground off as far back as was necessary to entirely clear the holes. A mushroom shaped piece of stainless steel was then pushed into the end of the nozzle and electron beam welded, as shown in Fig. 24a. The tip was then profiled outside and inside and then drilled according to the required pattern. Stainless steel was chosen mainly because of its 'weldability'. It is sufficiently soft to drill if care is taken, but sufficiently hard and resistant to combustion products to last under test conditions for 400 hours or more.

This technique was entirely successful and satisfactory nozzles could be obtained quite rapidly. They proved to wear well even when run on unpurified homogenised heavy fuel. The only difficulties experienced were in drilling the stainless steel, as it was liable to work harden. Because of this difficulty a further development was embarked upon, as shown in Fig. 24b. Nozzle tips were preformed externally and pre-drilled to the required pattern, but the internal hole was made smaller than that required in the final condition. This hole was tapped to receive a rod which passed through the nozzle and was used to hold the tip tightly against the mating face. Electron beam welding of the tip to the nozzle was then carried out, the rod was unscrewed and the internal sac formed to the finished dimensions by drilling using the existing hole below the




*(b) Revised Method*

*(a) Original Method*

*Fig. 24—Electron Beam Welding of Fuel Injector Tips*

seat as a guide. This meant that any accident in the drilling occurred on the relatively cheap stainless tip before it was welded on to form a complete nozzle. The most important saving was in time, as electron beam welding was carried out only on tips that had been correctly drilled.

These development nozzles have been perfectly satisfactory and have lasted for over 100 hours without showing any signs of deterioration. It may be that they would be perfectly satisfactory for use in service, but this has not been attempted and is not intended at the present time.

**Test Bed Results**

Engine tests have been carried out on marine diesel fuel, fuel with viscosity of 1100 seconds Redwood No. 1 and fuel of viscosity 3,000 sec. Red. No. 1. When corrected to a standard value the tests all give the same result. Results for the final four-injector configuration used on the first engine is shown in Fig. 25a. For comparison the best results obtained with two injectors so far are shown dotted.

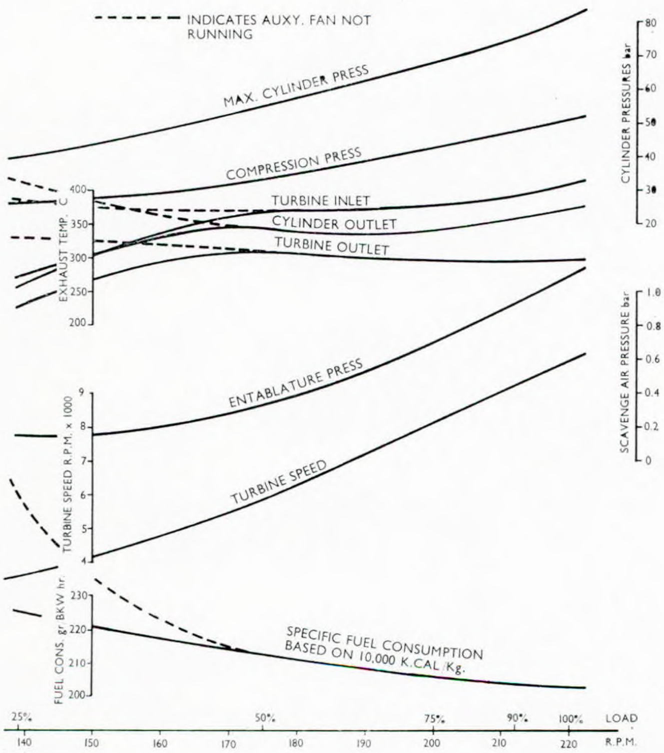



*Fig. 25(a)—Engine Performance Curves 58JS3*

During fuel consumption tests fuel was measured by means of a tank mounted on load cells. These load cells had previously been calibrated by dead weights. The engine was loaded by a Heenan & Froud dynamometer with pneumatic indication of the torque. As a check against the pneumatic measuring system one of the links in the dynamometer load balancing gear was fitted with a load cell, the output being displayed on a digital volt meter. Both the original pneumatic gear and the load cell measurement of torque were calibrated by dead weights hung on an arm attached to the carcass of the dynamometer. In addition to the foregoing, the assistance of B.S.R.A. was enlisted and strain gauge torque measuring equipment was fitted by them to the shaft between the engine and the dynamometer. These three means of measuring the torque gave closely similar readings; throughout the tests the maximum overall error being of the order of 0.5%.

The test bed results obtained for the first constant pressure turbocharged 76J4 engine are given in Fig. 25b for comparison purposes. This engine was turbocharged by a Napier 650 turbocharger fitted at the after end. The

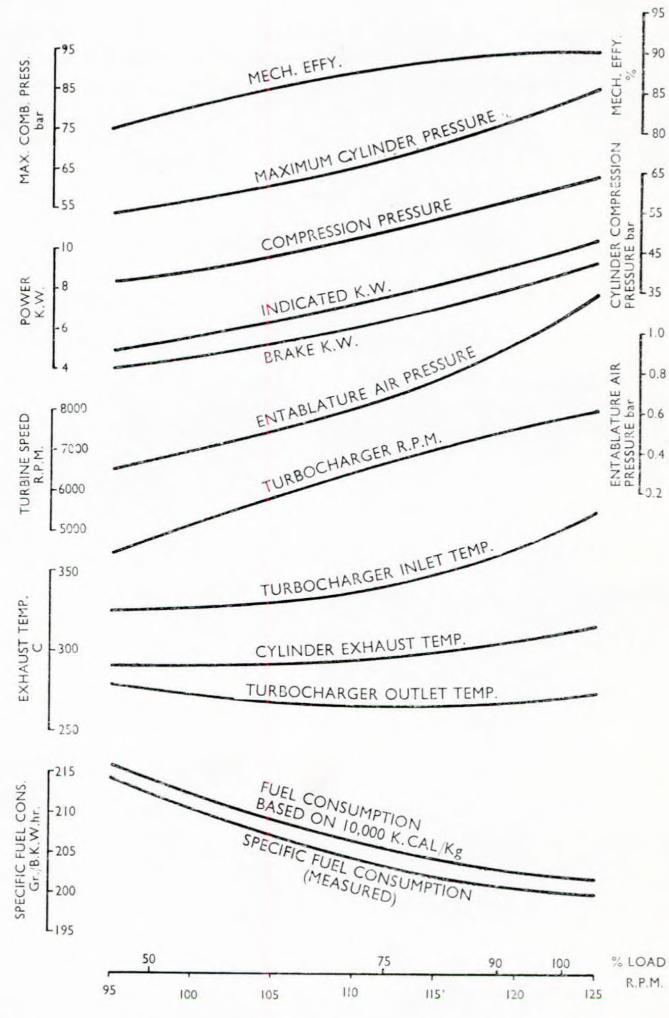



*Fig. 25(b)—Engine Performance Curves 76J4 Engine*

scavenge and cylinder pressure were about the same for the 76J4 and the 58JS3 engine, but the exhaust temperatures were 20-30°C lower for the former engine. This indicates a higher specific air flow for the 76J4 engine, but the results were very satisfactory for both engines.

The specific fuel consumption was marginally lower for the 76J4 engine, which was fitted with two injectors per cylinder. It is probably the higher rotational speed of the 58JS3 which makes the four injector arrangement preferable. However, as the load on the engines is reduced, the specific fuel consumption for the 76J4 engine rises more. The timing valve characteristics given by Figs. 11a and 11b partly explain this result, as the start of injection is retarded slightly more when the load is reduced from full power for the 76J4 engine. Further improvements in the timing valve characteristics are considered for both engines to lower the part load fuel consumption.

## Acknowledgements

The authors wish to thank Doxford Engines Limited for their permission to publish this paper and they extend their thanks to their colleagues within the company for their cooperation and support. Many useful suggestions were made by Ellerman City Liners and other friends in the industry. Ricardo & Co. Engineers (1927) Ltd. should be mentioned in particular, as they provided the results presented in Fig. 23. The authors are grateful for all the support they have received.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# 'The Doxford Three-Cylinder Engine'

## The London Meeting

**Mr. A. J. S. BAKER,** *Fellow of The Institute of Marine Engineers*

Henry Henshall and Finn Ørbeck are to be congratulated on producing the most outstanding and down-to-earth British paper I have heard at any venue for several years. True, it is an exceptionally successful story which they have told, but it just goes to show what a very small team, such as Doxford's, can do by grinding intellectual effort, in a surprisingly short time.

The happy concept of revising the much loved '3 legged' Doxford in turbocharged form is so close to the obvious in these awful industrial times as to be sheer inspiration. As any experienced engineer knows 3-cylinder Doxfords established the fame and reputation of the marque and played no small part in winning the Second World War.

Opening speakers frequently look for possible points to criticise and carp on. If any of our excellent audience tonight are looking for such fireworks I must disappoint them entirely. Firstly, due to the unfortunate printing delay of the paper, I saw it for the first time just before the meeting. Secondly, from an admittedly frantic first reading I can find nothing worthwhile to criticise. Instead I will attempt to emphasise a few of the elegant features which the paper describes.

In Table II you will note the total absence of unbalanced couples. This feature is already taken advantage of in the bedplate design and chocking arrangement. However, I would suggest that its full advantage will be seen (and felt) when the engine is installed in a vessel with a decent afterbody form. I believe many mariners may come to bless Henry and Finn for this feature alone.

Figs. 5 and 6 show again the elegant top end and side rod arrangement that we saw in the much lamented Seahorse. However, in this case I believe that the rather lumpy looking extension, to convey oil to the piston coolant trombone, also has a balancing function to perform. It is this ability, to make one part do several distinct functions, which marks a brilliant piece of design work and divides the "master from the lad". Moreover, because this design is so simple in concept it is also one of the most difficult to design in detail. Similarly, although I have seen various pictures of the top end and side rod practice in this and the fabulous Seahorse, they never cease to thrill me with their inspiration. Mr. Henshall has already described the elegant top end in some detail, which needs no emphasis beyond admiration, but the side rod concept of carrying the principal tensile loads in long steel studs and the compressive stresses in the forged or cast members, is so correct that it amazes me that we see its application in so few push–pull functions, even today. In 40 years I have seen several people try to apply repetitive reversing stress cycles to cast aluminium cylinder blocks on five separate occasions, only to meet various forms of disaster every time, although it must be clear to anyone blessed with reason and eyesight that if the casting is placed under compression by suitable tensile members, the casting will be stabilised and the assembly will have a long and happy life.

The piston design (depicted in Fig. 8) is again typical of brilliance and commonsense, such a rare combination today. It must be obvious to all that combustion heat is bound to produce piston distortion; Doxfords have recognised this fully by designing a piston which does not attempt to restrain thermal strain, but to take advantage of it by the careful use of thin sections. It is therefore not surprising that the 1st and 2nd ring groove temperatures are maintained to moderate levels (Fig. 9). This is, of course, just what is needed to successfully lubricate the cylinders and piston rings. However, the real pay-off comes as shown in the low regularity of both forms of twist in each ring. First, the long cycle effect of each total running period indicated covers many cycles, as witness the pronounced slope to the ring profiles. Secondly, the smaller order of in-cycle ring twisting is demonstrated by the large radii of curvature of the peripheral profile. As is known, my Company has worked with Leeds University on rings and ring dynamics for some years, but Doxford are probably the first engine builders to understand and apply what has been freely published for all to read.

The present arrangement of auxiliary fan has the attraction of being a fool proof and fail safe device, once the engine has been started and run-up to its constant speed for the fan. That the performance of this fan, in series with the turbocharger, is excellent is borne out by the performance data of Fig. 25a. Many alternate arrangements are possible, but do practical seagoing engineers really want the complication of electronic sensors and such boxes of mystery, if they can be avoided without penalty? I think emphatically not. They would infinitely prefer a spare fan motor, shaft and bearings in the unlikely event of fan failure at sea. I cannot imagine a Classification Society demanding anything else in the way of fan spares. Even if some "belt and braces" man demanded a complete spare fan, its cost, stowage, and improbable fitting effort, would be so slight as to render it not too significant, compared with the problems of a new complex system.

Turning now to combustion, the rapid achievements of the small Doxford team deserve to be held up as an example to most of the people 'beavering away' on similar problems, but in much larger research groups. The enormous effort of experimenting with various numbers of injectors shows just what Britain can do if she really tries, even under awful times and financial pressure. So they used some specialist ability from Ricardo, as they acknowledge honourably, but the results shown in Fig. 23, which incidentally show evidence of research methods evolved by the late and great J. F. Alcock, suggest that none of the latest sophisticated and costly methods of modern anemometry have been used here. However, the final results shown in Figs. 25(a) and (b), and arrived at so quickly, fully justify the steps and methods used for this research.

The problem of accurately matching a fuel injector to any new engine is common knowledge, as, too, is the difficulty of procuring experimental nozzles needed as research proceeds. Almost casually Fig. 24 shows how Doxfords have eradicated these problems totally, cheaply and in one tremendous stride. It also demonstrates Doxfords awareness of the potential of the latest manufacturing techniques. This they have demonstrated by the electron beam welding of new and fully drilled stainless tops to unsuitable, and even used, injector bodies.

The present paper is superb in itself, but far more than this it shows clearly that enthusiastic and gifted British engineers can still beat the World if they have sufficient dedication and half a chance in the way of finances and time.

Truly, a magnificent paper, beautifully presented about an outstanding British achievement in these dark times.

**Mr. D. A. WIGHT,** *Fellow of The Institute of Marine Engineers:*

This is a wide ranging and very interesting paper and not the least remarkable aspect of the work it describes is that it all took place within less than 2 years. I would like to select just one aspect for comment, namely the injection/combustion. I recall my own involvement several years ago in optimising the injector spray angles, etc. and the combustion chamber shape on two types of medium speed four stroke engines.

Naturally there are the obvious differences, but what struck me about these cases was that both showed a final solution which involved unequivocal impingement on the piston surface. The primary problem in each case was, in fact, degradation of the cylinder oil, and hence, with such engines, the sump oil by insolubles, in both cases with some visible smoke. At the time of the first example there was considerable reluctance to risk impingement, because of fears for the metal temperature in the target area of an uncooled aluminium piston. Yet, when this condition was included in the range of options, it was found that the rise in piston temperature was scarcely measureable, but oil degradation ceased to be a problem, the exhaust became invisible again and a useful gain in fuel consumption was achieved, especially when a relatively large holed nozzle was used. Of course, the outer edge of the bowl of a Hesselmann piston crown affords a ready target to a central injector and it is where most of the air is. Moreover, wetting the piston, rather than the flameplate, guarantees air movement over the fuel during the combustion/expansion phase.

The second example, a larger engine, did not have, initially, a true Hesselmann shape but a simple convex piston bowl floor, against which, obviously, the individual fuel sprays did not fit very snugly. In fact, initially impingement took place at the underside of each spray cone close to the injector and a substantial temperature rise was measured (some 50–60°C). Closer consideration suggested that this was due, not to burning, but, more likely, to a transfer of the heat of the compressed cylinder charge to the fuel spray and thence to the piston, the essential conditions for efficient heat transfer, *viz* pressure, velocity and the liquid phase, all being amply fulfilled.

Restoring the Hesselmann shape allowed the same technique used in the earlier case to be applied with equally good results, including a fuel consumption of 145 gm/bhp/hr. In fact, the second case was the English Electric "C" engine, of which, for reasons which are now part of history, only a dozen were ever built.

One should be wary, in matters where science is not yet established enough to supplant intuition, of ascribing one's success to what may, for all one knows, have been handicaps, but I support the authors' *prima facie* finding that impingement was a beneficial ingredient in the outcome. Do they intend to explore this avenue further?

Many more years ago I was involved with an opposed piston two stroke, intended for locomotive application, but never brought to production and I remember particularly the complexity of trying to optimise the combustion, when altering one variable inevitably altered two others. The engine was phased with an exhaust lead. The philosophy at that time was to use fan shaped sprays, each nozzle hole being individually specified, in a flat cylindrical combustion space.

The authors have, in their running to date, used sprays in a cone formation, and no doubt the orientation about the diametral plane is important. Have they considered experimenting with asymmetric spray patterns, including fan arrangements?

Finally, much of the advance discussion of this engine has centred, not just on the "aggressive" fuels mentioned, but on its ability to burn what might be looked upon as the outrageous, for example water oil mixtures, or even mixtures including coal. Is it too early to ask the authors to comment on such a topic?

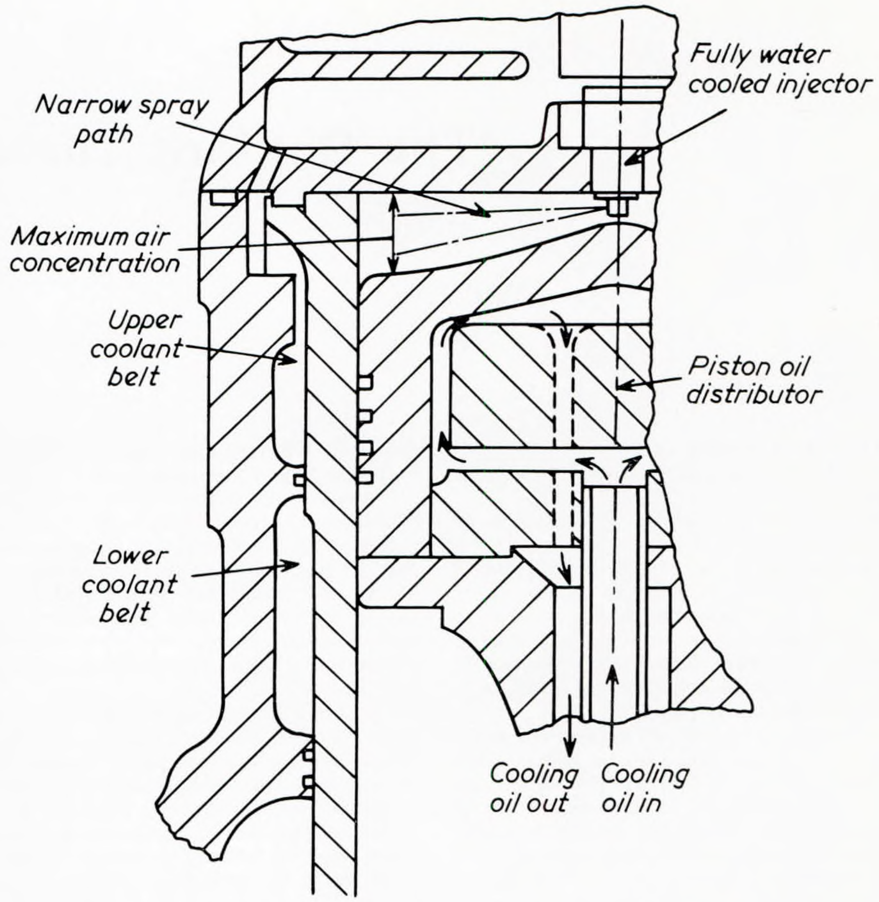


*Fig. 26—Abingdon B—1 Combustion Chamber and Part Cylinder, at Top Dead Centre*

**Mr. A. J. S. BAKER** (*further contribution*): All diesel engineers must know that it is a basic fact that fuel cannot burn efficiently unless it meets and mixes with sufficient oxygen at the appropriate time in the combustion cycle. Some smaller engines, with higher total fuel to air ratios and swirl, may be able to augment this basic requirement by such devices as deliberate impingement of the fuel spray on the walls of the combustion chamber, but if this route is attempted in a large, low-speed two-cycle engine such problems as local hot spots and piston crown burning inevitably follow. Fig. 25 of the paper shows the relative volumes of air in the chamber at different radii of combustion volume concentration. This pattern occurs simply because the combustion volume at any radius is dependent directly upon its vertical depth, but the square of the radius. The same diagram also shows the low swirl ratio inherent to this type of chamber, but somewhat augmented by optimised fuel injection. I demonstrated this to my own satisfaction in a 203 mm base experimental two-stroke, as shown in Fig. 26 (one half of a concentric section). Knowing that only low swirl ratios are possible with a full scale low speed engine, I did all I could in the way of porting, etc. to minimize swirl in this small chamber (an Alcock swirl number of only 8 was achieved). In reaching the strange combustion chamber shown in Fig. 26[7] I must acknowledge Dr. H. Watson, Ing E. R. Groschel and, of course, the late Mr. J. F. Alcock. The rest of the diesel establishment thought it very funny and not to be taken seriously. Nevertheless, within only 6 weeks the engine had operated, smoke-free, up to 18 kg/cm$^2$ and has stood the test of time as a cylinder lubricant performance predictor to this day.

That engine, the Abingdon B–1, has a single central injector and heightens the problem of combustion chamber volume by its very short stroke of 273 mm, unlike the long combined stroke used by the authors. However, I was fortunate in having the enthusiasm of Ing. Groschel, and the acceptance of Dr. Watson, who eventually produced a fully cooled nozzle, which worked with a sufficiently narrow spray pattern to miss the chamber walls, albeit with the then unprecedented maximum line pressure of over 900 Ats.

Since that time the engine has continued to run many thousands[8] of research hours with heavy fuel of all available types, including the sort of fuel which marine applications may be forced to burn in future. I believe, therefore, in a modest way, that I am able to confirm that the new Doxford should eventually be able to digest the worst fuel characteristics offered.

As regards the reliability of the semi-built crank, this is not going to prove any less reliable than a single piece forged crank. Two significant changes lead me to this conclusion. Firstly, Doxfords have eliminated the water cooled injector sleeve, which had been known to allow cooling water to leak into a cylinder if improperly fitted. Unfortunately, the paper does not show, completely, the clever and idiot proof arrangement of the injection valve where it passes through the combustion space wall. By this advance, hydraulic locking of a cylinder is reduced to an act of sabotage or other event too unlikely to contemplate in practice.

Secondly, with the latest practice of shrink/force at Doxford, it is normal to scrupulously degrease the surfaces before they are shrunk together. Thus any shrink junction liable to 'let go' in the old days will today start a mechanism of friction welding between pin and web. My experience of friction welding occurring adventitiously has been for it to produce local welded areas, which not only will not slip, but, if it is ever required to separate them, a choice must be made between the component of the junction to be preserved and that to be sacrificed, because try as one may the only way to dismantle the junction is to trepan away one or other component.

Another speaker enquired about the applicability of the new 3-cylinder Doxford to waste heat recovery by fluid bed. I am fortunate to have Dr. G. Moss as a colleague and friend. It was he who conceived the Chemically Active Fluid Bed (C.A.F.B.). From Dr. Moss' researches it would indeed be very advantageous to extract the heat of exhaust from the 3-cylinder turbocharged Doxford in view of the high turbo-charger outlet temperature. By this means it is possible to recover up to 85% of the exhaust heat. This would probably mean two beds in series, or cascade, not to produce excessive back pressure on the turbine. I cannot say how best this recovered heat might be utilised at sea, but I am sure someone cleverer than myself, like Mr. Crowdy, whom I note is with us tonight, may start to do some sums, bearing in mind that a properly designed fluid bed should give a coefficient of heat transfer of at least 50 Btu/Ft$^2$/°F/Hr to the piped fluid to be heated. This compares with about 12 Btu/Ft$^2$/°F/Hr in a conventional static heat exchanger.

Furthermore, the figures of performance quoted by the authors, together with the high turbine outlet temperature, could make such a potentially reliable means of initial energy conversion into a beautiful unit for total energy supply for such applications as remote communities. With its perfect dynamic balance and total absence of rocking couples, foundations could be simple and cheap, while its simplicity, potential reliability and ease of operation and maintenance could easily sell it for this purpose. These days it is electrically practical to produce regular 440 v 50 cycle supply at virtually any alternator speed.

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**The Newcastle upon Tyne Meeting**

**Mr. A. J. WICKENS,** *Member of Council:*
First of all I would like to say how much I appreciate being invited to open this discussion. I have had a long association with both authors and have been privileged to work alongside the Doxford Engine Design Team, which has generated an admiration for their innovative attitude and approach to engineering problems, and this has been fortified by reading and listening to the paper. We all delight in an engineering success story and I am sure the tale we have heard tonight will rank as a milestone in two stroke engine development.

The adoption of an exhaust piston crank lead of 180° takes us back to the days of the naturally aspirated Doxford Engine, which, of course, was also a constant pressure machine. The complication of having a lead in excess of 180° was understandable in the case of the pulse turbocharged engine. A figure of 188° became enshrined in turbocharged opposed piston engine thinking and was consequently adopted for the Seahorse engine, although it was constant pressure turbocharged. To start, therefore, with, perhaps, a hypothetical question: Did the authors have any misgivings about adopting the 180° exhaust piston crank lead and will its use now become universal Doxford practice?

There will be general agreement about the fuel situation outlined and its influence on engine development. However, some of the trunk piston four stroke engine manufacturers would take issue with the suggestion that they operate their valves at above 700°C. One of the leading manufacturers adopts a design criterion of not exceeding 450°C in the hottest region of his exhaust valves.

There is an obvious transposition between the text and the figure numbers in the case of Figs. 4a and 4b. The point of mentioning this is that it would be interesting to see a detail of the main bearing keep. Is this component designed to form part of the strength section of the main bearing girder? Does it abut or is it attached to the two vertical cheeks shown above the crankshaft centre line on Fig. 4a.

Table II compares the unbalance external couples of the 58JS range with Seahorse and two single piston engine contenders. The comparison is not quite valid, as the 58JS occupies a rather lower power range than the other designs shown. It does, however, serve to illustrate the very modest unbalance figures experienced with the 58JS engine and for interest, since the four cylinder single piston two stroke machine has now made its debut, on the same basis as the other figures, its primary couple is 1.89 meganewton metres and its secondary couple 1.44 meganewton metres.

There is a comprehensive description in the paper of the running gear bearings, but it omits a reference to the material. Could the authors say if all shell bearings are of white metal, or whether it has been necessary to go to aluminium tin or a similar high load bearing alloy?

In Fig. 1a there is apparent swelling towards the lower end of each side rod. It is believed that this is part of an improved tightening arrangement for these rods. If so, I am sure operators will appreciate such a development. Perhaps the authors could amplify what they have done in this region?

Fig. 8, comparing the piston design of the Seahorse with the 58JS engine, conceals a feature on the Seahorse version which involved a relief of about 0.5 millimetres on the supporting surface between the piston crown and the rod, outside the pitch circle diameter of the studs. This allowed for flexure of the crown without overstressing the studs, which hitherto had been subject to occasional breakage. Could the authors say if this feature has been perpetuated?

Fig. 10 presented an invitation, which I could not resist, to calculate the slopes and radii of curvature of the ring profiles shown. Comparing the mid-ring profile of Number 1 ring of the lower piston with an early Seahorse result one gets the following interesting comparison. The 58JS3 shows a radius of curvature of 2.22 metres and a slope of 4.5 times $10^{-3}$. This compares with the Seahorse figure of 0.34 metres radius of curvature and a slope of 8.5 times $10^{-3}$. The substantial increase in radius of curvature and the halving of the slope is indicative of the improvement resulting from the new design of piston and the remarkably low top ring groove temperature, which must be of the order of 90°C, is a major achievement.

The description of the timing valve characteristics, and the considerations involved, is most interesting. Later in the paper there is a hint of further improvements to reduce the part load fuel consumption and I am sure the authors will recall discussions about schemes to give adjustable injection start timing. Is it possible to say anything further about these possibilities?

One of the major novelties of the engine is the starting positioner and I am sure other contributors will have questions to ask about this which I would not wish to pre-empt. It would, however, be interesting to know if the conventional Doxford two lever control system is used for the engine or whether a pre-selection direction lever is incorporated.

The section dealing with combustion is extremely interesting and is an object lesson in how to make sense out of a series of apparently inconsistent development results. Could the authors confirm, however, that the engine as it is now running has the system of swirl vanes shown in Fig. 19a, and were these the ones used on Seahorse throughout its 500 hours trial? The authors will probably recall that, as part of the subsequent development programme, alternative swirl systems were to be investigated.

Understandably, because of the excellent condition of the piston rings, the paper does not mention cylinder lubrication. It would, however, be interesting to know if the system adopted is identical with the timed injection arrangement used on the J engines, and whether the injection points for the lower piston are below or above the scavenge ports.

The 58JS engine has made a major impact on the marine propulsion world. In the few moments granted to them, when they were not actually pre-occupied with the day-to-day development problems of launching such a project, the authors would only be human if they indulged themselves on speculating about the next stage along the road. Perhaps they could lift the curtain a little bit to give us some idea of their further thinking.

**Mr. B. TAYLOR,** *Member of Council:*
Mr. Taylor took the opportunity of adding to the information provided in the paper. He said that results which had just become available from the sea trials of the first ship to be fitted with the Doxford 58JS3 engine were very promising. During these trials, which lasted for a total of five days, the control equipment had been most thoroughly tested. After setting and adjusting the controls it was shown that manoeuvring of the engine was entirely satisfactory when operating on fuel with a viscosity of 3,500 secs. Almost double the number of starts required by the Classification Society from one compressed air receiver were obtained when the engine was operated remotely from the bridge.

In order to test the starting positioner the engine had to be set to a dead-band position by means of the turning gear. He was not sure whether the engine had stopped normally in a dead-band as the operation of the positioner is entirely automatic when on remote control and the only indication of its functioning is the momentary illumination of a warning light in the control room. From a dead-band position the starting sequence takes a total of seven seconds, five of which are occupied by the barring-round of the crankshaft by the positioner before starting air is admitted to the cylinders.

The engine room is designed for unmanned operation and very comprehensive monitoring equipment is provided for the main engine to indicate any deviation from normal operating conditions. It was pleasing to note that during the six hours U.M.S. test at full power revolutions there was not a single alarm, even though the trials were carried out under adverse weather conditions.

**Mr. J. R. B. ROBERTSON,** *Associate Member:*
Any new engine type from a well known engine builder is of great interest. However, apart from the advance copy of the paper I have not had an opportunity to see this engine at the builder's works, nor have I seen any details previously.

Having said this there are a couple of points I wish to raise:

1. It appears from the drawings included in the paper, in particular Fig. 5, that the centre crosshead pin is penetrated by about 10 holes for attachments, oilways, etc.

The bolts which secure the piston rod flange and crosshead oil supply bracket result in four landing faces which in service could be subject to pulsation; this also applies to the faces of the guide shoe attachment.

From dealings in the past with 2 cycle engines I have had experience of fractures being propagated from the attachment bolt holes, requiring renewal of the crosshead pin.

Are special precautions taken during manufacture in respect of internal finish, counterboring, radiusing, etc. of the attachment holes.

2. My second query is in respect of the auxiliary fan, which, when not in use, 'windmills' all the time the engine is in operation: at what R.P.M.? Also, is there no tendency for the impeller and casing to foul up?

In this connection what is the fouling situation with regard to the reed valves and what access is there to the reed valves?

It is clear from the contents of the paper that extensive research has been carried out during the development of the engine and these matters may well have been investigated. The authors are to be congratulated on their most informative paper.

**Mr. J. B. STEWART,** *Member:*
Would the authors explain the 'blip' on the light spring diagram, Fig. 13b, as this brought the blow down exhaust pressure perilously near the scavenge pressure at the port opening.

**Mr. A. JEMMETT,** *Member of The Institute of Marine Engineers:*
My initial reaction is to congratulate both the authors Messrs. Henshall and Ørbeck for a paper well written, and well presented.

To achieve good combustion and fuel consumption it was necessary to use a four injector arrangement. One of the disadvantages seen with this system is that, in the event of an injector failing, it would be extremely difficult to locate, consequently all four would have to be removed for testing, which presents some difficulty, considering all pipes are fitted with anti-spray tubes and are in a position of limited access.

Are the designers still considering a two injector system and, if so, could not an angled nozzle be designed to ensure spray penetration into the outer parts of the combustion chamber, avoiding the central area, and thus achieving similar results to the four injector systems?

Two means are provided for starting the engine, a conventional pneumatic rotary valve pilot air system and a pneumatic, electrically operated, pilot air system; the latter proving to be most efficient in respect of higher starting revolutions, but the former being considered most robust. Will both systems be retained and, if not, what is the preference of the designers?

In conclusion, I would like to congratulate Doxford Engines for their courage in designing and producing a prototype engine, which is, in fact, the first of seven production engines being built to meet a rigid shipbuilding schedule.

## Correspondence

**Dr. P. A. MILNE,** *Vice-President:*
The paper presents an account of how an existing engine range has been adapted to meet market conditions by capitalising on its best features through the application of research and development results. The authors are to be congratulated on their work, particularly as it offers the prospect of improved sales for the only remaining British design of slow speed diesel. Continued effort in this direction was a feature of British Shipbuilders report and accounts, 1977/78, and underlines the importance the Corporation attaches to this particular work and the future of the Doxford Engine design. The progress is also a tribute to the team responsible for the Seahorse Project, because not only have the test results been used in the new design, but a number of the most successful features have been incorporated directly.

As the authors stress in the paper the Doxford Engine range has a number of features of particular advantage in both current and future market conditions. The ability to operate without valves will be a substantial benefit when dealing with the poorer grades of fuel predicted in future ship operation. The opposed piston arrangement minimises balance problems and associated noise generation external to the engine. Could the authors say something of the design approach taken to minimise noise generation within the engine and their ideas on how further improvements might be progressively obtained? This feature, together with engine balance, will become increasingly important in the future and here again the Doxford Engine starts with an initial advantage. It has also been noticeable on other engine ranges that, as the number of cylinders has been reduced, it has become increasingly difficult to accommodate torsional and axial vibration in the shafting system design. Could the authors say something of their experience in this area and offer views on how the problem may be accommodated on the new engine range? This may become more difficult as engines are uprated and a given power delivered from a reduced number of cylinders. In fact, this question is of particular interest to the Doxford designers, as they, more than any other team, are moving into a range of designs which could offer competitive installations in ships that previously have used medium speed engines.

Reference is made to the re-design of the bedplate to simplify its construction and take full advantage of the shorter stroke and reduced size of main bearing journal. This is also an example of a production engineering approach to the design process and the resulting structure certainly looks as though it would be very much cheaper to produce, both in terms of the material element and also labour content. Could the authors give other examples of this type of approach to the design of other main features? This must, of course, always be reconciled with the technical requirements, so that a satisfactory structural analysis is achieved at the same time as minimising production costs. This will not only be first cost, but also total operational through life costings, to ensure that the customers requirements were being met. A further example of this is given in the section of the paper dealing with combustion, where, ultimately, it was found necessary to use four injectors. No doubt some further work on air flow and injector nozzle design might make it possible to revert to the earlier arrangement of two injectors per cylinder. This is not only obviously cheaper from a first cost point of view, but offers the prospect of a reduced maintenance burden. When commenting on the performance of the experimental nozzles, reference is made to running on unpurified homogenised heavy fuel. Could the authors say something on this part of the experimental programme, as other engine designers had concluded that a homogeniser can be an advantage, but they are not confident enough at this stage to dispense with purifiers or other forms of fuel treatment?

There are a number of aspects the authors have not been able to cover within the confines of the paper. For example, little is said of condition monitoring and control equipment, although this must have taken some proportion of the development effort.

In a similar vein, only occasional reference is made to 'maintainability' and, here again, this aspect must have received some attention, even allowing for the fact that the opposed piston design makes the cylinder and combustion space rather more accessible than in alternative ranges.

**Mr. P. MANSON,** *Member:*
I would like to congratulate the authors on the presentation of a most interesting paper, at a time when many owners must be giving serious thought to the type of machinery they plan to use for future new tonnage. As pointed out in the paper, marine plants of the future will be required to use poor grades of residual fuels and the so called slow speed diesel engine, under which heading the 58JS type qualifies, will, in my opinion, be the most suitable to cope with these conditions, apart, of course, from the steam turbine plants, which will also have a part to play in future years.

It is also pleasing to note that the authors have devoted the greater part of the paper to research and development on the combustion side of the engine, as against that of the crankcase running gear.

In my experience of late, the crankcase running gear of most of the modern slow speed diesel engines operates, on the whole, comparatively free of troubles, provided proper micro-filters are fitted in the lubricating oil system and competent engineers are in charge of the machinery.

Many marine engineers can well recall problems with the centre crossheads of the older design of Doxford Engine, and whilst it is known that the crankcase running gear of the modern J Engines, including crankshaft and bearings, runs trouble free, it would be very interesting if the authors could indicate, from their feed back information, what their records prove in this respect. It would be enlightening to many engineers, whose minds can only recall the old design of Doxford Engines.

I can recall the crankshaft deflections of the Sun Doxford Engine (cylinder bore 813 mm), where deflections of anything between 0.6 mm to 0.8 mm were the normal. There are still many engineers who visualise the deflections of the modern J Engine still being away above those obtained in the modern Sulzer or B. & W. engines. It would, therefore, make a very useful addition to the paper if the authors could show the old and new design of crankshaft, together with their respective deflection readings, and thus clarify the situation.

I personally like the arrangement of the four fuel injectors, as employed on each cylinder, and from the results shown in the paper it should prove most successful in service.

I would, however, like to ask the authors views on a very important matter, namely, the total running hours after which all injectors should be changed, irrespective of their condition. In my opinion, many troubles experienced with cylinder liners, pistons and rings, etc. are due to neglect of maintenance of the injectors, which, in turn, results in heavier engine maintenance costs, plus, of course, increasing fuel consumption. One of the features of the fuel injectors referred to in the paper is that they are small and comparatively lightweight and easy to manhandle. If the authors could also indicate the cost of the injectors this would be helpful, to enable an owner to decide whether re-conditioning, or renewing, was the best course. The practical skills, as obtainable from the marine engineers of past years, are not so readily available to shipowners today.

In the latter part of the paper it is quite evident the importance the authors have placed on good combustion, thus keeping the thermal loading to good, acceptable limits. Over the years this is one aspect that has been glossed over and the authors are to be congratulated in the thoroughness with which they tackled this matter, and, in my opinion, ensured the success of the J Engine for the future, when there will be much greater variation in the grades of fuel for use in the diesel engine.

## Authors' Reply

*To Mr. Baker:* The authors express their thanks to Mr. Baker for his complimentary remarks. His complete understanding of so many of the problems and their solutions has enabled him to highlight several points in a very interesting way, for which the authors are most grateful.

The matter of the auxiliary fan is one which may well be a subject both for discussion and serious investigation for some time. The advice to continue with a simple arrangement that works well cannot lightly be disregarded; nevertheless, the parallel fan has attractions. It is likely to yield a small, but worth-while, saving in fuel consumption and a better chance of easier starting. This latter feature results from the fact that the parallel fan delivers air through a flap valve, which is shut while the engine is stationary. It opens when the main control lever is moved off its stop position and is closed again when this lever is moved beyond its position for approximately 40% power. The air from the series fan is blown through the engine while this is stopped. It also goes through the cooler just after the fan and is, therefore, colder through the engine than at the fan delivery. Of course, the air delivery passages from the series fan could be arranged to by-pass the cooler, but the arrangement cannot be made as neat as the present one and, unless a valve is incorporated, the loss of cooler effectiveness at full output would be further detrimental to performance.

In his subsequent comments at the end of the discussion, Mr. Baker referred to the arrangement of the fuel injection valve and for completeness the method of fitting is shown in Fig. 27.

On the subject of total energy units the Doxford Engine has an advantage that Mr. Baker has been quick to spot. Because the high specific output is obtained with relatively low cycle pressures there is a correspondingly lower pressure drop across the turbine, resulting in the exhaust energy being available at a more useful temperature.

*To Mr. Wight:* The authors thank Mr. Wight for describing his interesting experiences with combustion in some other engines. In the case of the Doxford 58JS3 it is not thought that impingement on the crown is essential to smoke free combustion or low fuel consumption, but simply that there are worse locations in which to distribute the fuel. The primary consideration is to spray the fuel where the air is and if this means a small amount of impingement then it is better to tolerate this, rather than attempt to avoid it at a cost of inferior air and fuel mixing. However, any condition which leads to local high temperatures and 'burning' of the piston crowns cannot be accepted.

It is confirmed that it is the intention of the Doxford team to explore all facets of combustion as far as possible. Asymmetric sprays, including fan shaped patterns, are not overlooked. It is interesting to recall the Doxford engines of an earlier vintage, having long strokes and correspondingly deep piston bowls, had fan shaped sprays, in which the plane of the fan was arranged vertically.

There are very good reasons for believing that the Doxford engine should be able to cope with 'outrageous' fuels better than most other engines. Some preliminary work has been done, but it is, as yet, too early to report and the story must wait to be told in some future paper.

*To Mr. Wickens:* The authors thank Mr. Wickens for the kind remarks with which he opened the discussion at the Newcastle upon Tyne meeting and are grateful for his penetrating questions, the replies to which they hope will provide an opportunity to explain some further details of the design.

The first constant pressure engine which Doxford built was the Seahorse and careful analysis of the practical measurements of pressure fluctuations in the cylinders and the exhaust system had led to a confident conclusion that an exhaust crank lead of 180° would

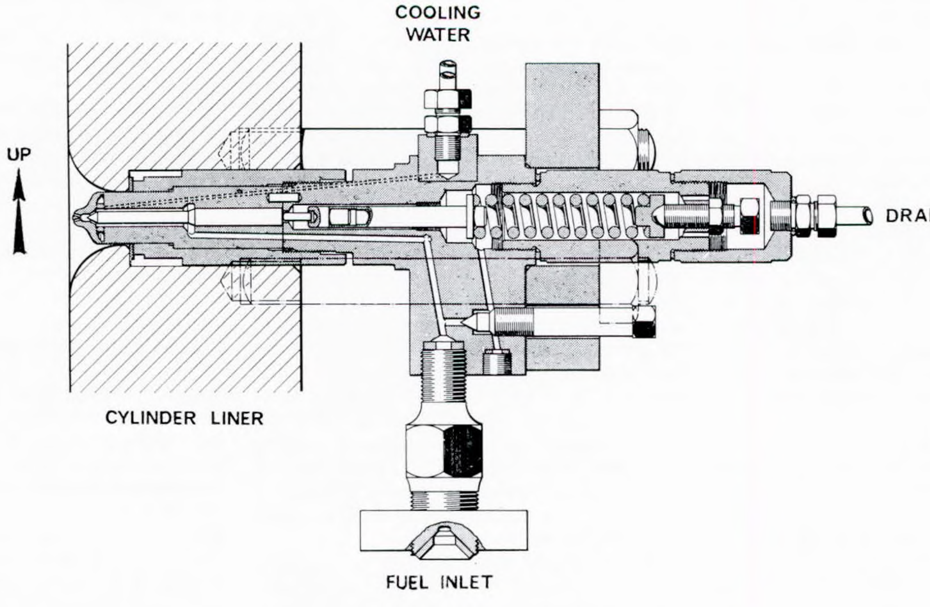


*Fig. 27—Assembly of Injector into Cylinder Liner*

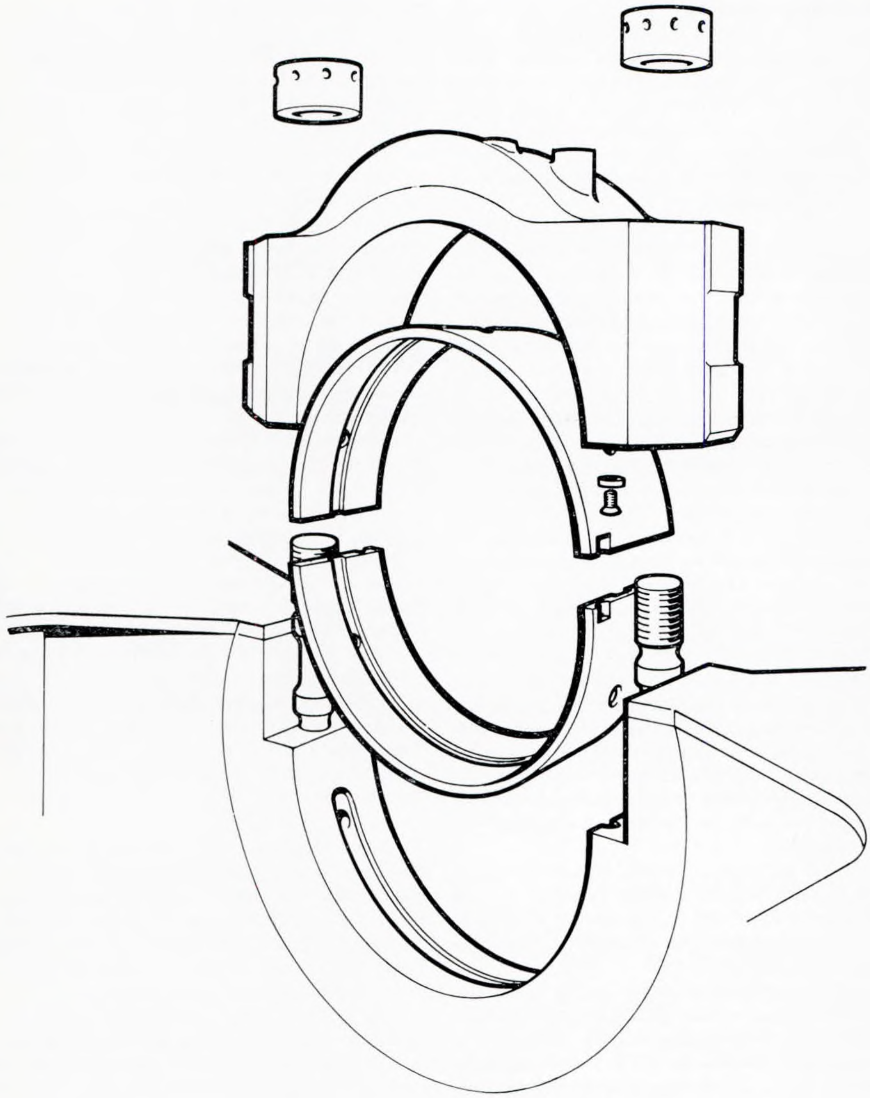

*Fig. 28—Main Bearing Assembly*

present little difficulty. It was, therefore, adopted without misgivings and it has lived up to expectations. It will be universally adopted for all Doxford constant pressure turbocharged engines.

The authors concede that not all trunk piston four stroke engines have exhaust valves with temperatures as high as 700°C, but considerable efforts in the design have to be made to keep the valve temperature down.

Fig. 28 shows the details of the main bearing keep which Mr. Wickens requests. It is conventional in form and is not called upon to act as a strength member to aid any support of the guide plate, as was the case with the Seahorse. It contributes to the strength section of the main bearing girder in much the same way as any other conventional main bearing keep does. It abuts the two vertical cheeks, but is not attached to them.

The authors feel quite justified in including the 58JS engine in Table II as it stands. The release rating of the engine is below its design potential and given a little finance and a little time for development, 2,000 horse power per cylinder is within its capability, without any modification to the out-of-balance forces or couples.

All the shell bearings on the crankshaft are of white metal and there has been no thought of going to aluminium tin, or any similar high load bearing alloy, for the initial release of this engine. The swelling at the lower end of each side rod denotes a hexagon, which is formed so as not to interfere with the stress flow. This is connected with an improved tightening arrangement, whereby the tightening torque and its reaction on the crosshead bracket, can be closely coupled by the tightening gear, with obvious merits in torque control, reliability and safety.

The design of the piston in the 58JS engine is directed towards greater flexibility, not only in the connection between the crown and the ring belt, but also the crown itself and the supporting ring. The diametral expansion of the crown is reduced to almost half that of the Seahorse and it has been considered that the need for the relief which Mr. Wickens mentions has disappeared. Without the relief the piston is simple to manufacture and there is obviously a better bedding surface to the rod.

Mr. Wickens comments on the piston ring profile underlining the success of the new and more flexible design of piston.

At this stage it is not possible to say much more about timing valve characteristics other than it is one of the items in the development programme for this and other Doxford engines.

The controls for the engines which have been built so far are integrated into a total bridge control system with an electronic governor. The emergency manual control box has the standard Doxford system with two levers, but incorporates, also, two push buttons for ahead and astern operation of the starting positioner by pneumatic means.

The engine is now running with the system of swirl vanes shown in Fig. 19a.

These are almost an identical mirror image of the ones used on Seahorse, as the direction of swirl on this engine corresponds with other Doxford engines, which is the opposite way round to that used on the Seahorse. Cylinder lubrication has the timed injection arrangement used on all Doxford engines, incorporating the developments for higher speed that originated on the Seahorse. The injection points for the lower piston are above the scavenge ports.

The engine has obvious development potential and, as the authors have indicated earlier, 2,000 horse power per cylinder should be well within reach. The opposed piston principle and the way it is engineered in the Doxford engine is a very good system for dealing with aggressive fuels. It is the intention to develop the engine to exploit these particular advantages.

*To Mr. Taylor:* The authors express their gratitude to Mr. Taylor for his contribution, which added the important information that had just become available to him from the sea trials of the first ship to be fitted with a Doxford 58JS3.

*To Mr. Robertson:* In reply to Mr. Robertson, the authors point out that the design of the crosshead pin has evolved from earlier designs on the larger Doxford engines, where the mating surfaces between the pin and the bracket were curved. The clamping forces necessary to avoid fretting were high and were apt to distort the bracket, because of the fundamental need for working tolerances on the curved surfaces. The present bracket, with its flat surfaces, is much easier to bed correctly and these require lower clamping loads to achieve reliability. Good finish on the internal surfaces of all the holes is carried out during manufacture to avoid stress raising irregularities.

The auxiliary fan 'windmills' at approximately 800 r.p.m. No tendency to foul has been observed.

Access to the reed valves is by a large door covering the aft faces of the valve box. The valves are within very easy reach; the operating conditions for the valves are not arduous, as they are not near to the cylinders and, therefore, not subject to fouling influences, neither are they subject to cyclic impact at their seats. In short, their operating conditions are far removed from those imposed on similar valves used in engines with under piston displacement scavenge systems.

*To Mr. Stewart:* Mr. Stewart has raised an important point concerning the 'blip' on the light spring diagram. This feature occurs after the blow down of the exhaust. If the diagram is examined closely it will be seen that during the blow down period the momentum which the exhaust gas has acquired continues until the cylinder pressure has dropped well below the scavenge air pressure. It is a natural result of the gas dynamics that this is followed by a swing to the situation that Mr. Stewart has noticed. However, the period during which

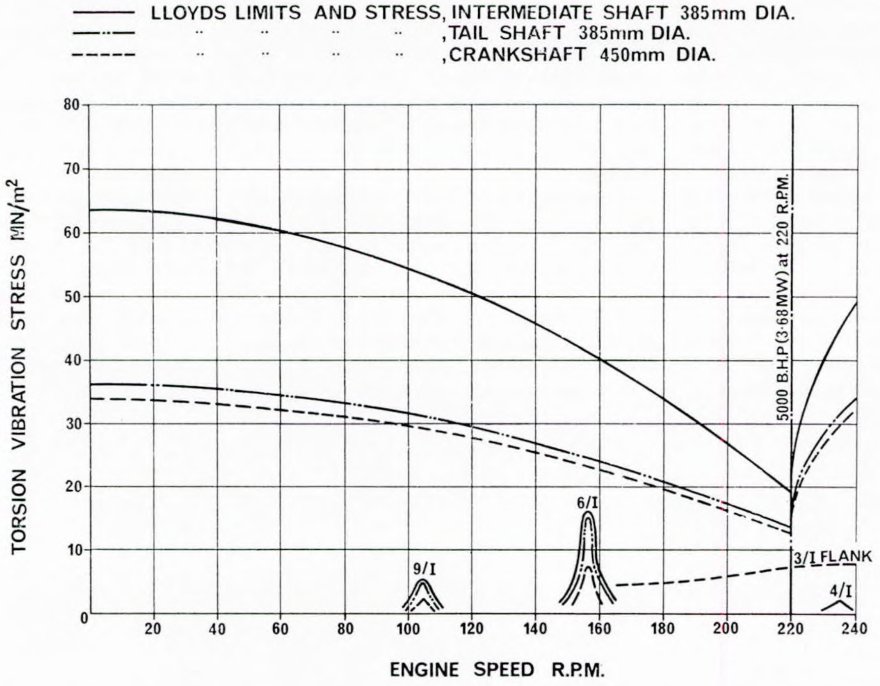


*Fig. 29—Aft End Installation Torsional Vibration Characteristics*

the pressure in the cylinder is higher than that in the scavenge manifold is very short and it is doubtful whether it is sufficient to reverse the flow. If a small reverse flow does occur then it is likely merely to return a little of the air that has just entered the scavenge ports.

*To Mr. Jemmett:* In his contribution Mr. Jemmett has put his finger on two interesting points. On the subject of injectors the authors must agree that it is a little more difficult to trace one injector that has failed out of four, but on the other side of the balance sheet it should be pointed out that the failure of one injector is not so detrimental to the overall performance of the engine. The two injector system is to be the subject of further investigation and development and angled spray patterns are high on the list of alternatives to be examined.

The electrically operated pilot air system is neat and is easily matched to electric control systems, which are becoming commonplace. On the present series of engines the electric system is regarded as somewhat experimental and for this reason the conventional pneumatic system is provided as well. It is the authors view that the future system is likely to be electrical when it has been developed to an adequate stage of proven reliability.

*To Dr. Milne:* The authors are grateful to Dr. Milne for underlining many of the fundamental advantages of the Doxford engine design. He has raised the subject of design for the least possible noise levels and many investigators in this field have shown that a major factor is to isolate the members carrying cyclic loads from the large surfaces enclosing the engine. In this respect the Doxford engine design is well placed, because its opposed piston single crankshaft concept does just this. Future improvements will be incorporated in the design of the filter-silencer and the compressor delivery pipe. Most of the noise at present experienced comes from auxiliary sources and a programme of research and development is being drawn up to include an attack on these details. Some of them are proprietary equipment, for example turbochargers, and the makers are undoubtedly at pains to reduce noise levels.

Another example is the valves in the telescopic pipes in the top end lubrication, for which the development programme must ensure that the correct pressure characteristics are attained, without high frequency valve flutter or similar conditions which cause noise.

Fig. 29 shows the torsional vibration system for this engine. In the authors' view, the direct drive systems are not likely to pose problems any more difficult to solve than those of medium speed geared engines. It should be borne in mind that the equivalent inertia of a medium speed engine operating through a relatively high reduction gear is quite substantial and the number of critical speeds for multi-cylinder four stroke engines is relatively high.

Literature on value analysis and value engineering reiterate that one can always improve one's first design. The design team at Doxford would not disagree with this, and, having designed what they believe to be an

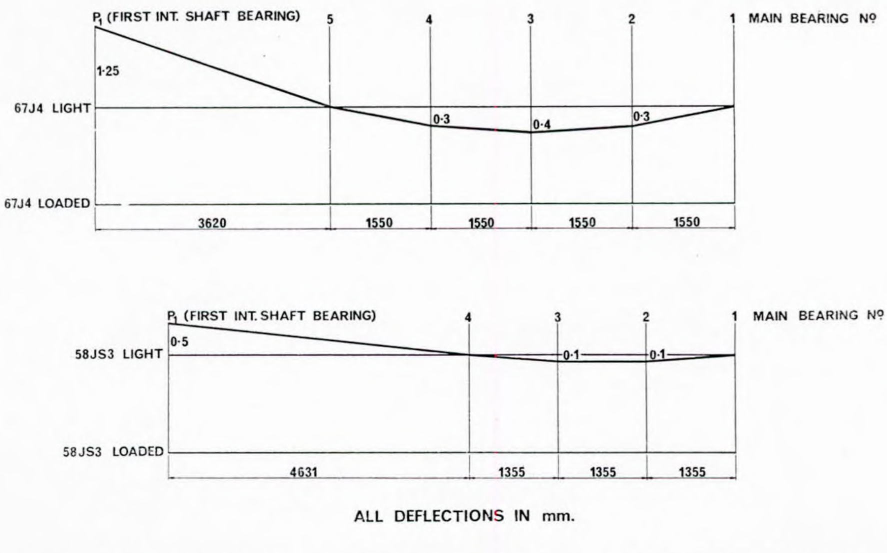


*Fig. 30—Comparison between Alignment Instructions for 67J4 and 58JS3 Engines*

excellent engine, are now only too keen to make it even better. It is difficult to single out components for particular attention, but obviously the main structural parts represents a large fraction of the engine cost and will probably receive prior attention. On the subject of running on unpurified homogenised heavy fuel, this has been practised on the Doxford engine test berths for the last four years. However, engines are on test for only a short time and the progress of the system into the service conditions will be watched intensely and with great interest. The authors are only too aware that there are many aspects which they have not covered and which must be most interesting. These must wait for another paper in the future.

*To Mr. Manson:* The authors thank Mr. Manson for his contribution. Centre crossheads were certainly problematical on many earlier design of engines, not only Doxford. The present Doxford design of bearings and their lubrication has given excellent results and the same can be said of the crankshaft bearings.

As the engines have been developed to give more power from smaller masses of metal they have also become stiffer and it is interesting to note that the 58JS3 simple bed plate is far stiffer than that of any previous Doxford engine. Fig. 30 is offered to compare deflection readings of new and old designs of crankshafts, as Mr. Manson suggests.

The authors are interested to see Mr. Manson is a protagonist of four fuel injectors. The main objections centre on maintenance work and he is right to highlight this point. The recommended time between overhauls is 2,000 running hours. The injectors are proprietary 'U' size injectors, which is the same size used on many four stroke medium speed engines. In the form used for the Doxford engine they are much shorter and, therefore, much lighter. The weight of one injector is 6 kg. The authors ask to be excused from publishing a cost for the injectors as, firstly, it would involve the manufacturers policy and, secondly, any figure would rapidly become out of date and meaningless. However, Doxford would be happy to quote for individual cases.

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