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THE DOXFORD SEAHORSE ENGINE

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The paper describes an engine designed to develop 2500 hp per cylinder at 300 rev/min combining many of the advantages of slow speed direct drive and medium speed geared marine engines. In four to seven cylinder configurations it will cover a range of powers from 10 000 to 70 000 hp as a geared marine propulsion plant, and will also be suitable for stationary generating applications for station loads from 7–100 MW, all with a single size of cylinder.

The unique advantages of opposed piston engines which enable such a range of engines to be designed with low stresses, good balance and four point support are examined. The design considerations relating to air flow, combustion, thermal stresses, gas and inertia loads, crankshaft and crosshead bearings, alignment and frame rigidity, torsional vibrations, control and monitoring are all discussed.

The constructional details of the engine are described and illustrated both by drawings and photographs with particular reference to the novel features such as the auxiliary blower drive, piston and liner cooling arrangement, engine frame, connecting rod, crosshead and crankshaft construction, detuner and expansion chocks. The production and transport facilities, quality control and service potentialities are explored. The presentation of the paper will be supported by test results from the 4-cylinder prototype engine.

INTRODUCTION

The Economics of the Geared Crosshead Engine

No apology is given for the repeated use of financial yardsticks in a technical paper presented to a learned engineering society. Engineering unrelated to the hard economic facts of life is a sterile abstract pursuit, totally incompatible with the well being of the profession.

Rotational Speed

Ever since the advent of screw propelled ships there has been a natural tendency to adopt direct drive. In this connexion it is interesting to recall that the s.s. *Great Britain* incorporated a chain drive between the steam reciprocating engine and the propeller, though in this instance the propeller ran at some three times the engine speed. The recent evolutionary trend towards much larger vessels calls for correspondingly lower propeller revolutions. The relationship between the absorbed power and the optimum speed of the propeller no longer matches the natural relationship between the speed and power of Diesel engines, the design of the direct drive Diesel engine increasingly being compromised by the conflicting claims of propulsive efficiency and low first cost. The adoption of reduction gearing allows the optimum speed of revolution to be chosen for both the prime mover and the propeller, a step taken many years ago and now completely accepted as being a *sine qua non* of turbine propulsion machinery.

In Fig. 2 a statistical analysis is attempted to show the relationship between vessel size and the economic liability of retaining a shaft speed matched to the slow speed Diesel engines currently being marketed. There is, of course, no point in adopting geared transmission unless the gains from so doing more than offset the extra maintenance demand and power loss, but such gains originate not only from higher propulsive efficiency but also from the lower first cost of higher speed prime movers and greater reliability due to divided power systems.

Referring to Fig. 2 and taking a 150 000 ton dw vessel as an example, it can be seen that the probability is that it will have a length of 280 m, a draught of 17·8 m and a propeller diameter of 9·9 m which would have an optimum rotational speed of 67 rev/min. On the other hand, 102 rev/min is suggested as being representative of the service speed of the currently available direct drive Diesel engines. The difference in q.p.c. arising from the variation in propeller speed for a vessel of this size is about 9 per cent. In other words, where current practice results in the use of a direct drive Diesel engine of some 29 400 hp (22 000 kW) a geared drive engine would only require to develop 27 500 hp (20 500 kW) when due allowance has been made for the loss of power in the gearing. It will thus be seen that there is a power saving of some 6½ per cent which is equivalent to an improvement in the first cost of the machinery of £1·50/hp (£2/kW). The fuel saving arising out of the lower propulsive power, when capitalized over the life of the vessel at going interest rates, is equivalent to a further £6/hp (£8/kW) giving a

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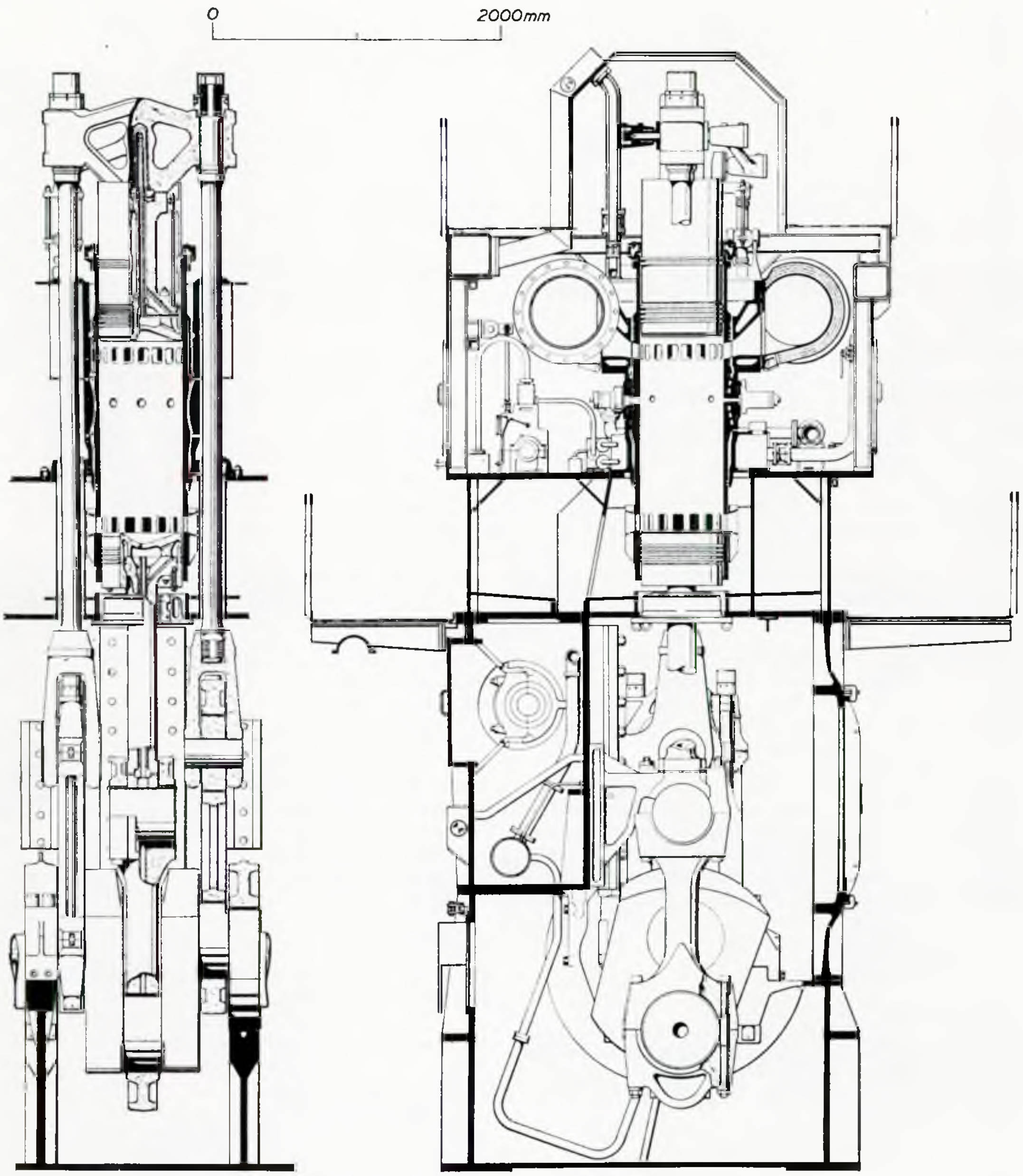


Fig. 1—Sectional arrangements.

total of £7·50/hp (£10/kW). In other words, it could be said that it would be advantageous to install geared machinery with a first cost some £7·50/hp (£10/kW) higher than that of direct drive machinery. The data used in the preparation of Fig. 2 is all taken from published information relating to motor ships presently under construction in the U.K. It will be noted that there is a cost saving in excess of £5 per installed horse power (£6·70/kW) for all ships above approximately 20 000 tonnes dw.

The design features which minimize the complications inherent with the adoption of reduction gearing are discussed in a later section of this paper.

Crosshead and Diaphragm

Though the advantages outlined above favour the adoption of geared drive in the majority of tonnage presently under construction, it must not be overlooked that the slow speed direct drive Diesel owes its prominence in the main to its high power per cylinder and its suitability for burning high viscosity fuel. Geared marine Diesel installations have been available for many years, but they have hitherto never succeeded in obtaining more than a small proportion of the market, confined in the main to specialized tonnage, due to the fact that they cannot take full advantage of the economies possible when operating on high viscosity fuel. The slow speed direct drive engine is, however, eminently suitable for operation on high viscosity fuel due to its low speed of revolution, its large size and its crosshead and diaphragm construction. It is, therefore, important that those features which allow full advantage to be taken of operation on high viscosity residual fuels are not sacrificed when an engine of this type is designed for geared drive.

There is a limit to the degree to which high viscosity fuels can be atomized. Individual particles of fuel entering the cylinder at high velocity must travel considerable distances before they are completely combusted. Clearly, the larger the combustion space the greater the chance of combustion being completed without impingement on the combustion chamber walls. In an opposed piston engine with a combined stroke/bore ratio of 2·24 the combustion chamber space is substantially nearer the spherical ideal than is possible with a single piston engine with half this stroke/bore ratio, a feature which further reduces the danger of fuel impingement.

Combustion must also be completed in the appropriate part of the cycle if the expansive potential of the engine is to be fully utilized and in an engine operating at 300 rev/min the time available is limited to about 1/100th of a second.

However good the combustion, it is nevertheless essential that the crankcase be separated from the combustion zone by a diaphragm to completely obviate the possibility of any products computed on the current differential of £4/ton between the price of marine Diesel and fuel oil.

Market Orientation

In order to ensure good service coverage, spares availability and quality control, it is essential that the potential demand shall be such as offers the best possible chance of quantity production of component parts, if not complete engines. The major part of the marine propulsion field must therefore be satisfied by the minimum number of frame and cylinder sizes.

It is in any case desirable that any installation should have the minimum possible number of cylinders, to facilitate maintenance and monitoring. Even were it practicable to have single, twin, or three cylinder engines, it would be impossible to provide a comprehensive range of powers with a single cylinder bore due to the large steps in power between the individual engines. As it was considered that the maximum step in power should be limited to 25 per cent, an examination of

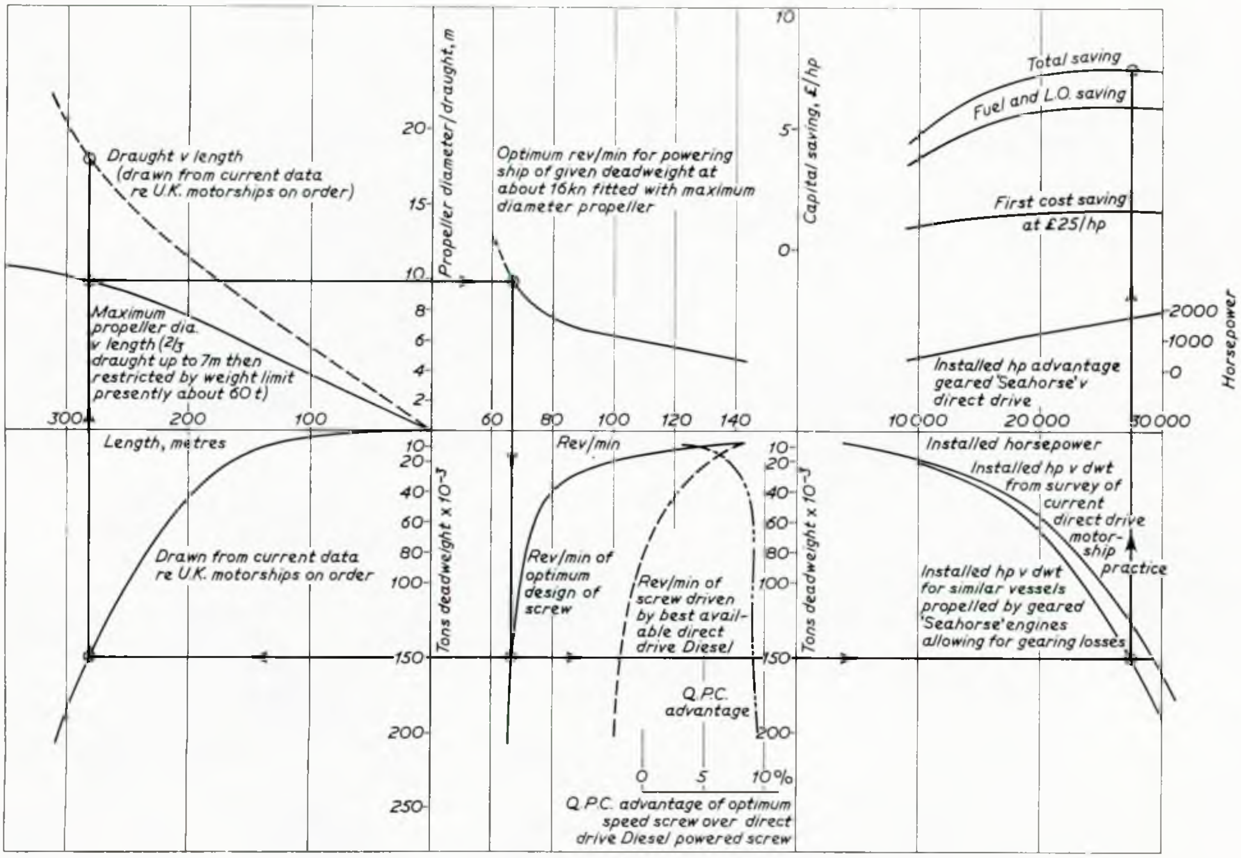


FIG. 2—*Engine first cost premium v ship size (propeller rev/min optimization)*

of combustion finding their way into the crank chamber. The inclusion of a diaphragm also allows the use of special purpose high alkalinity oils for cylinder lubrication and the crosshead construction in conjunction with a completely symmetrical liner ensures even wear and controlled consumption of cylinder lubricating oil independent of liner and piston ring condition.

Since the crankcase oil is of relatively inexpensive straight run mineral type formulated for lubrication rather than neutralization it is eminently suitable for use in the main reduction gears, thus obviating the necessity to have two completely separate forced lubrication systems.

The full economic advantages of crosshead and diaphragm construction in terms of fuel and lubricating oil saving are equivalent to no less than £40/hp (£53·60/kW) in first cost when possible engines led to the conclusion that the only practicable arrangements were those listed in Table I. The notional single piston engines listed with an output of 0·5 hp/cm² (3·73 MW/m²) are envisaged as uniflow scavenge two-stroke engines—they could of course be four-stroke engines with a B.M.E.P. twice that shown.

With the B.M.E.P. and low stroke/bore ratio listed, the latter being necessary to keep down engine weight and first cost, there would be a considerable overlap between the crankshaft pins and journals. Overlap is highly desirable in small engines since it allows the crank web thickness to be reduced, or even eliminated, but it prevents the use of built-up crankshafts in large engines. The single piston engines would therefore have to have solid forged crankshafts, each of which would be unique

TABLE I—PROPORTIONS AND PERFORMANCE OF HIGH OUTPUT CYLINDER UNITS FORMING THE BASIS OF A RANGE OF DIESEL ENGINES—*to cover any power requirement from* 10 000 *to* 70 000 *hp* (7·5 *to* 52·5 *MW*) *with the minimum number of frame sizes consistent with adequate rating at intermediate powers*

Type	*Power/cyl. hp*	*Eng. sizes*	*Power/cm² hp*	*Bore mm*	*B.M.E.P. kgf/cm²*	*Piston speed m/s*	*Stroke/bore ratio*	*Rev/min*
Single Piston	2000	5–6–7–8	0·5	715	11	6·8	1·12	256
Single Piston	2500	4–5–6–7	0·5	800	11	6·8	1·12	227
Opposed Piston	2500	4–5–6–7	0·95	580	10·9	13·0	2·24	300

to its own engine size and, incidentally, for which no forging facilities are available in this country. The single piston engines would also give rise to very considerable design problems in respect of balance, internal moments and torsional vibration as mentioned elsewhere. On the other hand, the opposed piston engine is eminently capable of covering a power range from 10 000 to 70 000 hp (7·5 to 52·5 MW) with a single cylinder size and only four frame sizes. Due to the combined stroke/bore ratio of twice that of the single piston engines, the crankshaft can be of the semi-built type in which the majority of forgings are interchangeable throughout the range, thus easing stock control problems.

With a rotational speed of 300 rev/min the opposed piston engine is suitable for electric generating duties at either 50 or 60 Hz. In this connexion it is interesting to note that 300 rev/min is the only rotational speed between 200 and 600 which is exactly suitable for both frequencies.

Production Costs

Though the specific weight of a geometrically similar series of Diesel engines is proportional to the cylinder bore, and as a consequence, the small high speed engine has an intrinsically low material cost, there is good reason to believe that a 580 mm bore opposed piston engine can be manufactured at a figure competitive in first cost with any other Diesel engines incorporating crosshead and diaphragm construction. The current world wide marine market could be satisfied by the production of twenty engines per week, and even if all these engines were manufactured in one plant, such numbers would not justify mass production equipment of the type used in the automobile industry. The specific costs associated with the handling, setting and sizing operations, increase rapidly as the bore is reduced. So long as the components are of a size that require mechanical handling equipment, little can be done to reverse this trend if the engines are required in relatively small numbers. On the other hand, if the size of the bore is increased to reduce the number of parts per unit power, the stage is soon reached where the handling of individual components becomes a major problem due to their extreme size and weight. The same problem of outsize components is manifested in material costs, the rate for which rises steeply when the number of potential suppliers is sharply diminished due to limited demand. The minimum cost per unit power for crosshead engines manufactured in traditional premises coincides with bores of about 750 mm. It should be appreciated that the actual shape of the specific cost/cylinder bore characteristic is unique to any particular establishment. The characteristic shown in Fig. 3 is based on cost figures relating to the manufacture of slow speed direct drive Diesel engines at a maximum rate of twelve per annum. It is considered that the increased production that should be achieved with a rationalized engine range with a unified bore, should enable the minimum point of the characteristic to be shifted to a smaller bore—it is hoped to exactly 580 mm.

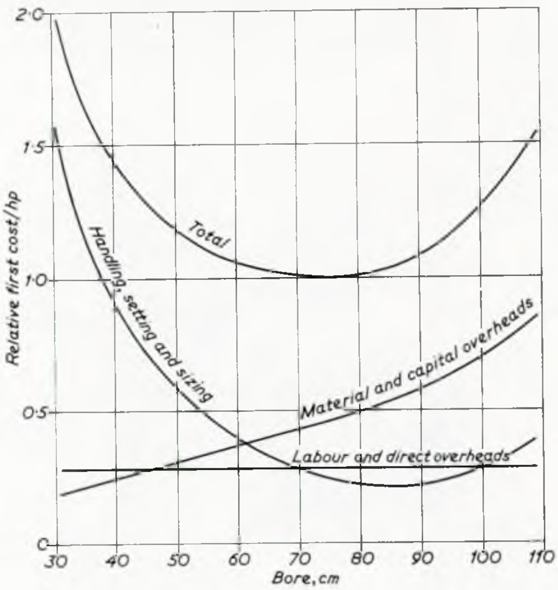


FIG. 3—*Specific cost v bore (crosshead engines)*

DESIGN PROBLEMS

Air flow

In a reversible two-cycle engine the gas exchange process consists of three phases. In the first phase, known as the blow-down period, the exhaust ports are open but the scavenge ports closed, exhaust gas escapes from the cylinder to the exhaust system and the cylinder pressure will fall roughly exponentially to that of the exhaust system. The scavenge ports should be opened during this process as soon as the cylinder pressure has fallen to the level of the scavenge trunk pressure. In the second phase, which is the scavenging period proper, both scavenge and exhaust ports are open, air passes into the cylinder and exhaust gas and some air pass from the cylinder to the exhaust system. The weight of air passing through the engine is dependent on the pressure difference between scavenge trunk and exhaust system, the absolute pressure in the scavenge trunk, and the time effective area of scavenge and exhaust ports. In the third phase, in which the exhaust ports are open but the scavenge ports closed, some of the air in the cylinder at scavenge port closure is lost to the exhaust system.

The power output can be increased by burning more fuel in the same air quantity at the expense of permitting higher gas temperatures and within the limitations of good combustion, or by increasing air throughput and maintaining the proportion of this throughput which is retained in the cylinder.

In a ported engine the available port area is proportional to the product of cylinder circumference and piston stroke multiplied by a factor dependent on the period in crankshaft degrees for which the ports are open. It is necessary to provide bars between the ports of suitable size to support the piston rings; so the only way in which port area can be increased in a given engine is by increasing their height by widening the angle of port opening.

In a direct drive marine engine, with crankshaft revolutions restricted to artificially low levels by propeller considerations, there is no great problem in providing sufficient port area to attain a satisfactory air flow with normal turbocharging, although some large bore slow speed engines require fairly high scavenge trunk pressures.

However, with a geared engine running at higher crankshaft speed, the bore and stroke are smaller in relation to horse power, and it becomes necessary to provide relatively larger ports or higher pressure drop across the cylinder, or a combination of both.

Some advantage in relative port area has been gained on the Seahorse engine compared with the 76J engine, which also produces 2500 hp/cylinder but at 119 rev/min, by making the exhaust piston stroke longer in relation to the main piston stroke. It can be shown that for an opposed piston engine the optimum ratio of exhaust piston to main piston stroke to give maximum air flow should be between 0·4 and 0·5, and in this engine the ratio is 0·478. In addition, the port opening period has been increased for both scavenge and exhaust ports by ten degrees of crank angle, giving exhaust opening 100 degrees after scavenge piston inner dead centre, closing 244 degrees after, and scavenge opening 132 degrees after inner dead centre, closing 228 degrees after. The exhaust crank lead is eight degrees as on the J-engine. This timing gives an equivalent orifice area for blow down of 54 cm^2 compared with 81 cm^2 for the 76J engine, and an equivalent combined scavenge orifice area of 168 cm^2 compared with 262 cm^2. Whilst these areas are smaller by approximately one-third in absolute value, the equivalent scavenge area divided by the product of bore and stroke is 0·0222 for the new engine, compared with 0·0158, an increase of 40 per cent in relative area.

The equivalent orifice area for blow-down is computed by integrating the product of exhaust port area and number of degrees of opening and dividing by 360, to obtain an area which would pass the same weight of gas for the same pressure difference if open continuously. The equivalent scavenging area is found by the same process but in this case the instantaneous effective area is calculated by the formula

$$A = \frac{1}{\sqrt{\frac{1}{A_s^2} + \frac{1}{A_e^2}}}$$

where A_s and A_e are respectively the scavenge port and exhaust port areas, and A is the effective area of the two openings in series.

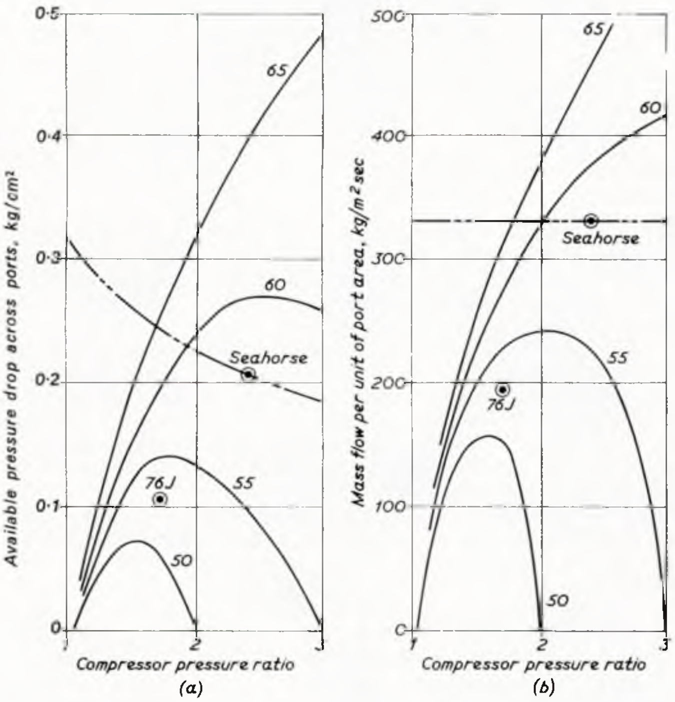


FIG. 4—*Port flow characteristics*
(a) available pressure drop (b) specific air flow

The smaller blow-down area for the same horse power is acceptable on the new engine because of the adoption of constant pressure turbocharging instead of pulse charging. Pulse charging has the effect of producing back pressure at the exhaust ports during the blow-down period, which retards the discharge of the exhaust gas.

On the other hand, the smaller effective scavenge area for the same horse power means that it is necessary to have about twice the pressure difference between scavenge entablature and exhaust system on the new engine to obtain the same total air flow.

Fig. 4(a) shows the pressure difference available across the ports in a turbocharged engine for varying scavenge trunk pressures up to three atmospheres absolute, and for overall turbocharging efficiencies from 50–65 per cent. Fig. 4(b) shows the corresponding mass flow attainable per square metre of port area at varying pressure ratios and turbocharging efficiencies. The overall turbocharging efficiency shown is the effective efficiency from discharge from exhaust ports to inlet to scavenge ports and is, therefore, lower than the product of compressor and turbine adiabatic efficiencies by an amount proportional to the sum of pressure losses in the air filter silencer, the trunking and air cooler, and the exhaust system before and after turbochargers, as well as by the power absorbed in the turbocharger bearings. The overall effect of these losses amounts to about 10 per cent of the turbine work so that an overall turbocharging efficiency of 50 per cent would correspond to a product of compressor and turbine efficiencies of 55 per cent.

On the 76J6 engine, pressure records taken at full load show that the mean pressure difference across the ports during the scavenge period is 0·105 kg/cm^2 at a scavenge pressure of 1·7 atmosphere absolute. This gives a specific air flow of 7·3 kg/hp hr. A specific flow of 8 kg/hp hr for the Seahorse engine requires the pressure drop shown by the chain dotted line on Fig. 4(a) passing through the point giving a pressure drop of 0·205 kg/cm^2 at a boost pressure of 2·4 atmospheres absolute. It will be seen that at this point the engine needs an overall turbocharging efficiency of approximately 58 per cent compared with 53 per cent for the current engine.

Some improvement in turbocharger efficiency may be expected from the more even gas pressure and full admission which obtain with constant pressure charging, but it is doubtful if any available equipment in the near future could produce the necessary internal efficiency of 64 per cent, except, perhaps, over a narrow flow range and with the compressor operating very close to its surge limit. Such conditions would make the engine unduly sensitive to fouling; even if full load operation were possible with turbochargers alone they would not be able to supply sufficient air at part load when exhaust temperatures would be much lower.

For this reason the engine has been designed with an engine-driven auxiliary compressor providing 10 per cent of the total air required. The additional air provided by the auxiliary compressor passes through the turbocharger turbines as exhaust gas and therefore enhances their power output, but because it is compressed separately it does not increase the power absorbed by the turbocharger compressors. The overall effect, therefore, is to give a flow characteristic through the engine similar to that which would obtain with a straight turbocharger system having 10 per cent higher overall efficiency. Besides ensuring that at service power the engine will be much less sensitive to fouling, the auxiliary blower ensures an adequate air supply during prolonged slow running and would provide emergency scavenging in the event of failure of one or both turbochargers.

To avoid the possibility of surging resulting from the parallel operation of the turbochargers and auxiliary compressor, the auxiliary compressor is arranged to supply air to the trunking between the aft turbocharger and aft end air coolers through the flap valve shown in Fig. 5. The piston of the pressure sensor moves an hydraulic pilot valve admitting lubricating oil at 70 lb/in^2 to one end or the other of the cylinder operating the main valve. If the pressure from the auxiliary blower is higher

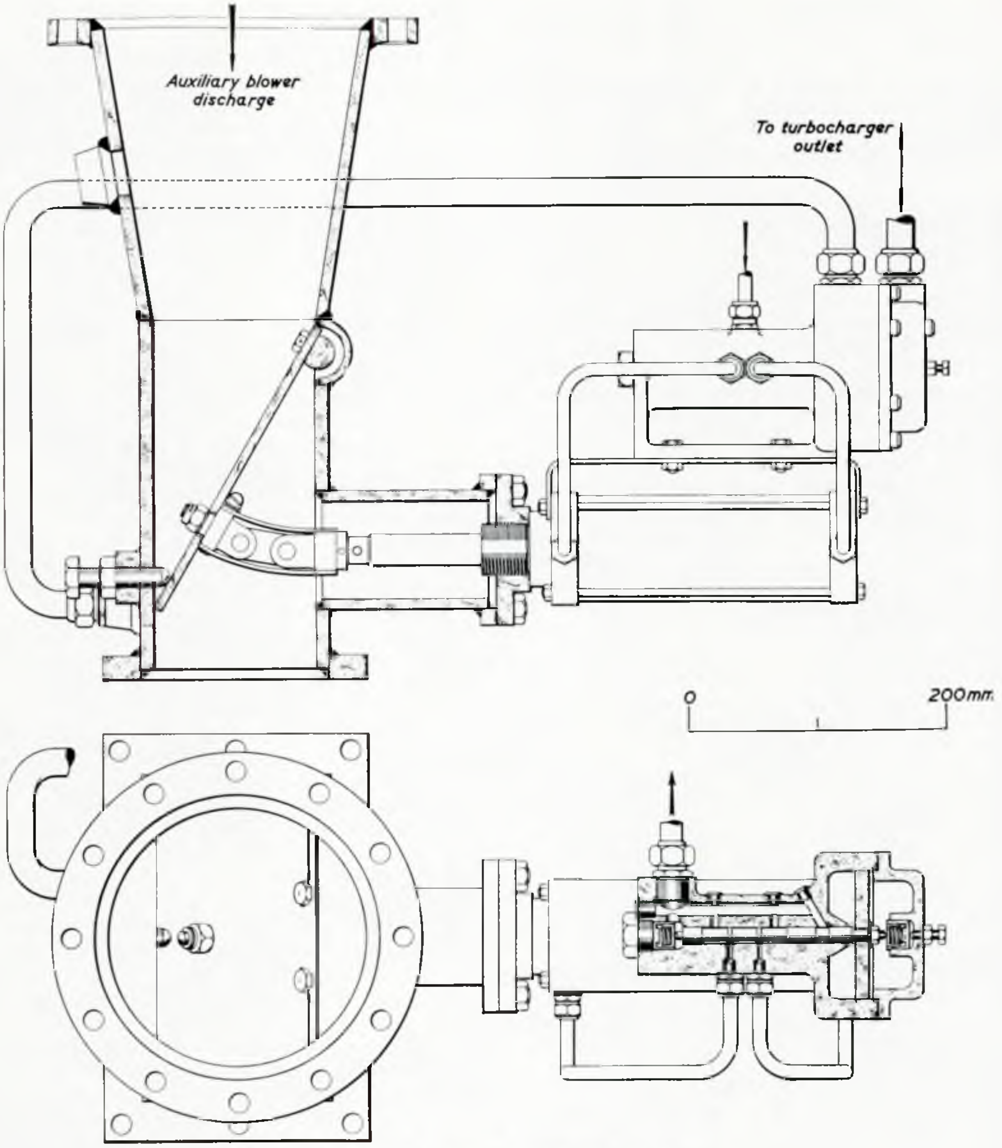


FIG. 5—*Auxiliary blower discharge control valve*

than that from the turbochargers, the flap valve reduces the outlet area from the auxiliary blower. The resulting increase in outlet velocity reduces the static pressure to the same level as that from the main turbocharger.

To keep the pressure drop through the engine as low as possible the exhaust ports slope upwards from the cylinder and the exhaust belts discharge at both back and front of the engine into two exhaust pipes, of which one carries exhaust gas to a turbocharger at the forward end of the engine, and the other to the second turbocharger at the aft end of the engine. The passages in the exhaust belts, shown in Fig. 6, are carefully streamlined to discharge the gas in the direction of flow along the pipe.

On the air side, discharge from the turbocharger compressors is through straight diffusers to the air trunk feeding the large air coolers. The diffusers enable some of the velocity head at the outlet from the compressor volute to be converted to additional pressure in the scavenge entablature. The cylinder inlet ports have aluminium inlet vanes to control swirl in the cylinder and to provide a streamlined entrance to the ports.

Combustion Efficiency and Fuel Consumption

Compared with a slow speed engine the mechanical efficiency of a geared engine must be lower because of extra friction losses due to the higher rubbing speed of the bearings and the power taken in this case by the auxiliary blower, which amounts to about 3 per cent of engine output. To attain equal specific brake consumption it is therefore necessary to have a higher indicated thermal efficiency.

Thermal efficiency in an internal combustion engine depends on the completeness of combustion and the useful expansion ratio after combustion. These two requirements are contradictory since perfect combustion requires a relatively long time, but fuel burned before piston inner dead centre results in negative work, and fuel burned after piston inner dead centre results in loss of useful expansion stroke.

To attain the most complete combustion of fuel possible in an acceptably short time it is necessary to limit the fuel/air ratio to the lowest practical level, to inject the fuel rapidly in finely atomized form, and to ensure that each fuel droplet finds the air it requires for combustion with minimal delay.

Fig. 7 shows for various compressor pressure ratios the total air flow (curve a), the mixture retained (b), the fresh air retained (c), and the fuel to air ratio (d). Curve (a) is derived from Fig. 4(a). Curve (b) is the product of the volume available in the cylinder at closure of the exhaust ports and the number of revolutions per hour; curve (c) shows the values in curve (b)

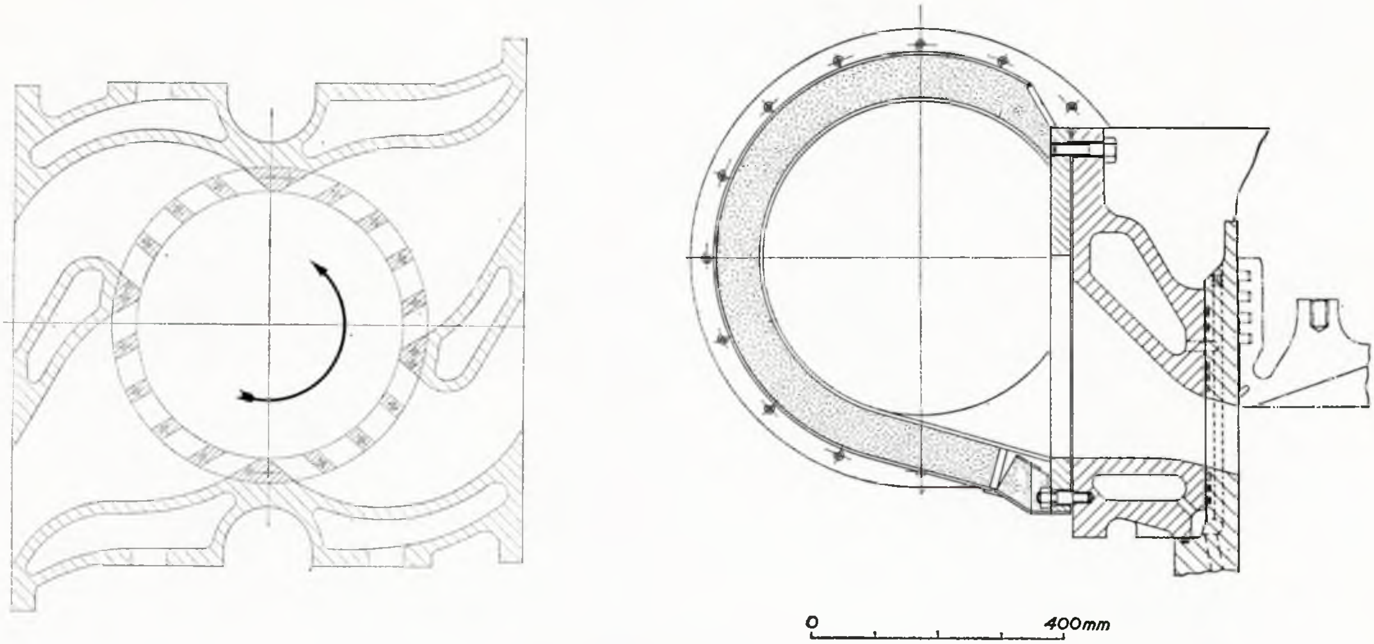


FIG. 6—*Exhaust belt*

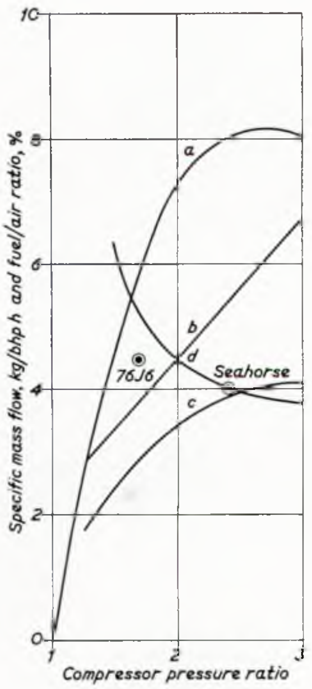


FIG. 7—*Air flow*

reduced to allow for the effect of mixing between incoming air and remaining combustion gas, and loss of density due to the remaining gas heating the incoming air; finally, curve (d) shows the fuel/air ratio calculated for a specific brake fuel consumption of 155 grammes/bhp hr.

It will be seen that the fuel/air ratio for the Seahorse engine is approximately 12 per cent lower than for the current 76J6 engine. Besides assisting quick combustion, lowering the fuel/air ratio reduces the tendency to dissociation and formation of carbon monoxide in the combustion gasses, both of which have an adverse effect on thermal efficiency.

Quick combustion is bound to result in not only an increased rate of pressure rise but also in a higher maximum cylinder pressure, so the engine has been designed to accept a cylinder pressure of 106 kg/cm² (1500 lb/in²) instead of the normal 70. The value of maximum/mean pressure is thus increased from 6·5 to 9.

Fig. 8(a) shows a cross section of the combustion space available at inner dead centre, (b) shows a plan view of the space, and (c) shows the radial distribution of available air and a typical distribution of fuel from a nozzle. The air distribution in (c) is obtained from the cross section (a) by multiplying the height of the space by the circumference at the radius concerned. The match between fuel and air distribution is good and the use of four equally spaced fuel injectors reduces the degree of air swirl needed to provide circumferential matching of fuel and air.

The use of four fuel valves is also necessary to provide sufficient nozzle orifice area backed by adequate flow area through the fuel valves. It has been found that attempts to extend the total orifice area of any fuel valve beyond about 50 per cent of the area at the needle valve seat results in the pressure drop through the nozzle holes themselves being unacceptably low in relation to the total pressure drop through the complete fuel valve, with consequent loss of atomizing performance.

The target of minimizing parasitic pressure drop has been kept in mind throughout the injection system. All high pressure pipes have been sized to suit a flow rate corresponding to injection in twenty degrees of crank angle, and the timing valve shown in Fig. 9 has a minimum through flow area of 120 mm² compared with 75 mm² for current engines. The high pressure pipes have flanged connexions for reliability at pressures up to 10 000 lb/in² and the main fuel pump has plain cylindrical plungers and no means of adjusting output. This permits adequate clearance between plunger and barrel to suit hot heavy fuel, without excessive leakage. Since there is no means of adjusting fuel pump output a third spill valve is fitted in the system to act as a safety valve should the air loaded spill valve in use become choked. The third spill valve has a diaphragm loaded by lubricating oil and is set to release fuel at a slightly higher pressure than that used for normal full load running.

The traditional Doxford fuel system employing an accumulator pump and timing valves has many merits. Not the least of these is its quiet operation and the absence of any rephasing device needed for reverse running. In the case of the Seahorse

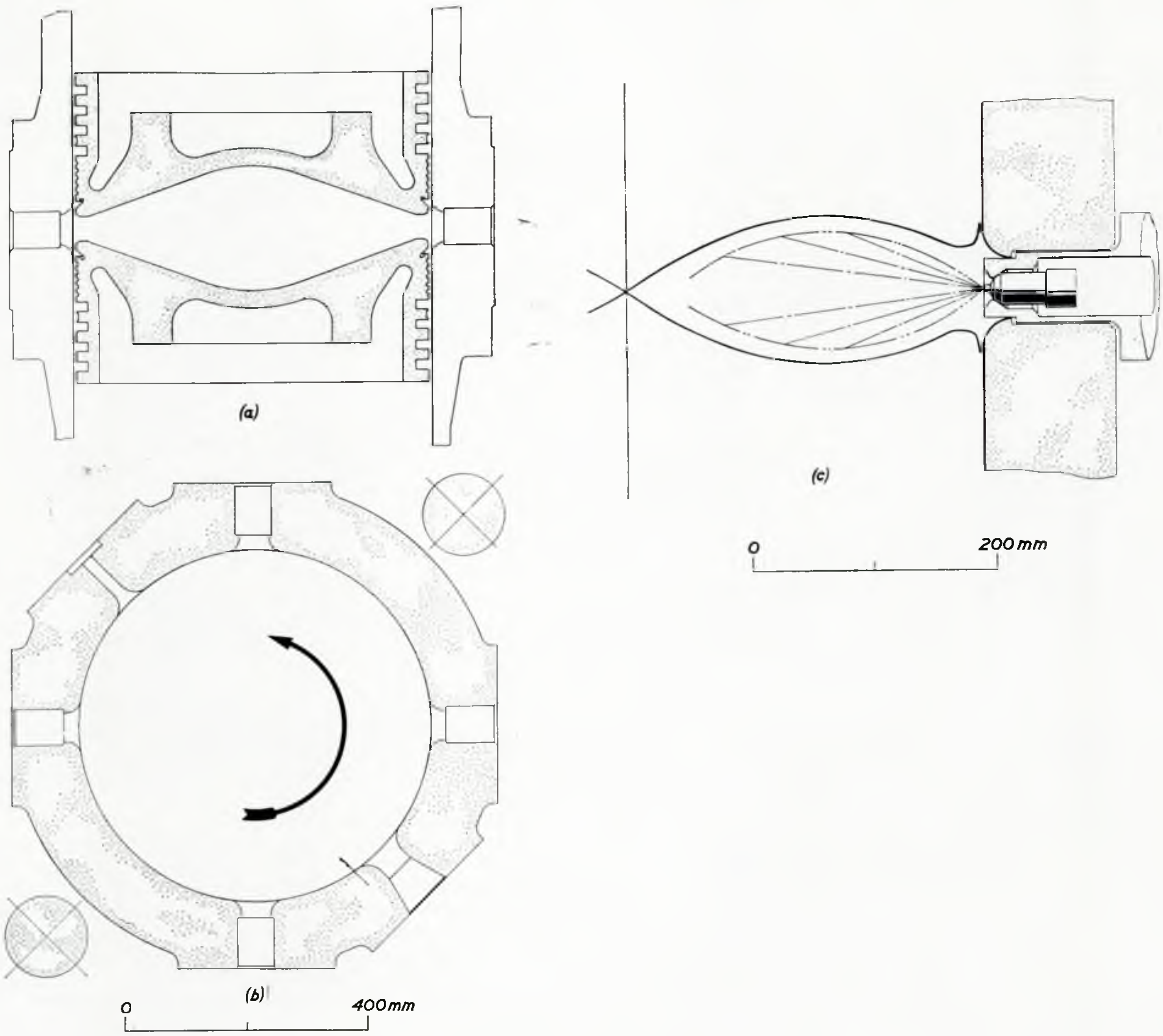


FIG. 8—*Combustion space*

engine, however, its main advantage which arises from the very much lighter auxiliary drive required by the constant pressure pump compared with a jerk pump, is virtually nullified by the fact that the balancing system and mechanical blower require the provision of a heavier transmission in any case. Accordingly, it was considered appropriate to fit an alternative jerk pump system on the prototype engine to enable a complete technical and cost comparison to be carried out.

2500 hp per cylinder at 300 rev/min demands a higher capacity pump than has previously been offered by the fuel injection equipment specialists. It is, of course, comparable in output per stroke to a four-stroke engine of 1250 hp per cylinder running at 300 rev/min but, in such an instance, the pump would operate at 150 rev/min.

The jerk pump adopted uses proprietary equipment consisting of individual scroll type pumps with integral roller cam followers. These pumps incorporate a gallery in the lower part of the barrel which is fed with lubricating oil at a pressure higher than the fuel surcharge pressure, thus obviating any possibility of fuel dilution of the crankcase oil.

On the prototype engine the pumps are grouped together and driven by a hollow camshaft surrounding the centre section of the balance shaft. The connexion between the concentric shafts is by an hydraulic servo motor which shifts the phase relationship of the shafts as appropriate for the required direction of rotation. The fuel cams which are symmetrical in form are capable of individual adjustment. The profile is designed to give the pump plunger the characteristics shown in Fig. 10. The injection of fuel takes place on a predominantly falling part of the velocity characteristic calculated to prevent an undue rise in injection and combustion pressures. The acceleration characteristic has been rounded off in such a manner that mechanical noise is reduced to a minimum without involving excessive inertia loadings. The leading parameters of the jerk pumps to be used in the initial tests on the prototype are shown in Table II.

The jerk pump system uses the same injectors as the accumulator system, though minor modifications are necessary to the high pressure piping to change from one system to the other.

A spragging mechanism allows individual cylinders to be cut in and out in safety whilst the engine is running. In the event of a governor malfunction all spragging mechanisms are released simultaneously by the spring return on a pneumatic hold-off cylinder, so stopping the engine.

With the jerk pump system the amount of fuel admitted to the cylinders at each stroke is constant at all speeds for any particular control position. On the other hand, the alternative timing valve system gives a constant fuel flow per unit of time so that the amount of fuel injected per stroke varies in inverse ratio to engine speed. The jerk pump system thus simplifies control of cylinder pressures during starting, but demands greater governor movements to maintain constant engine speed with changing load.

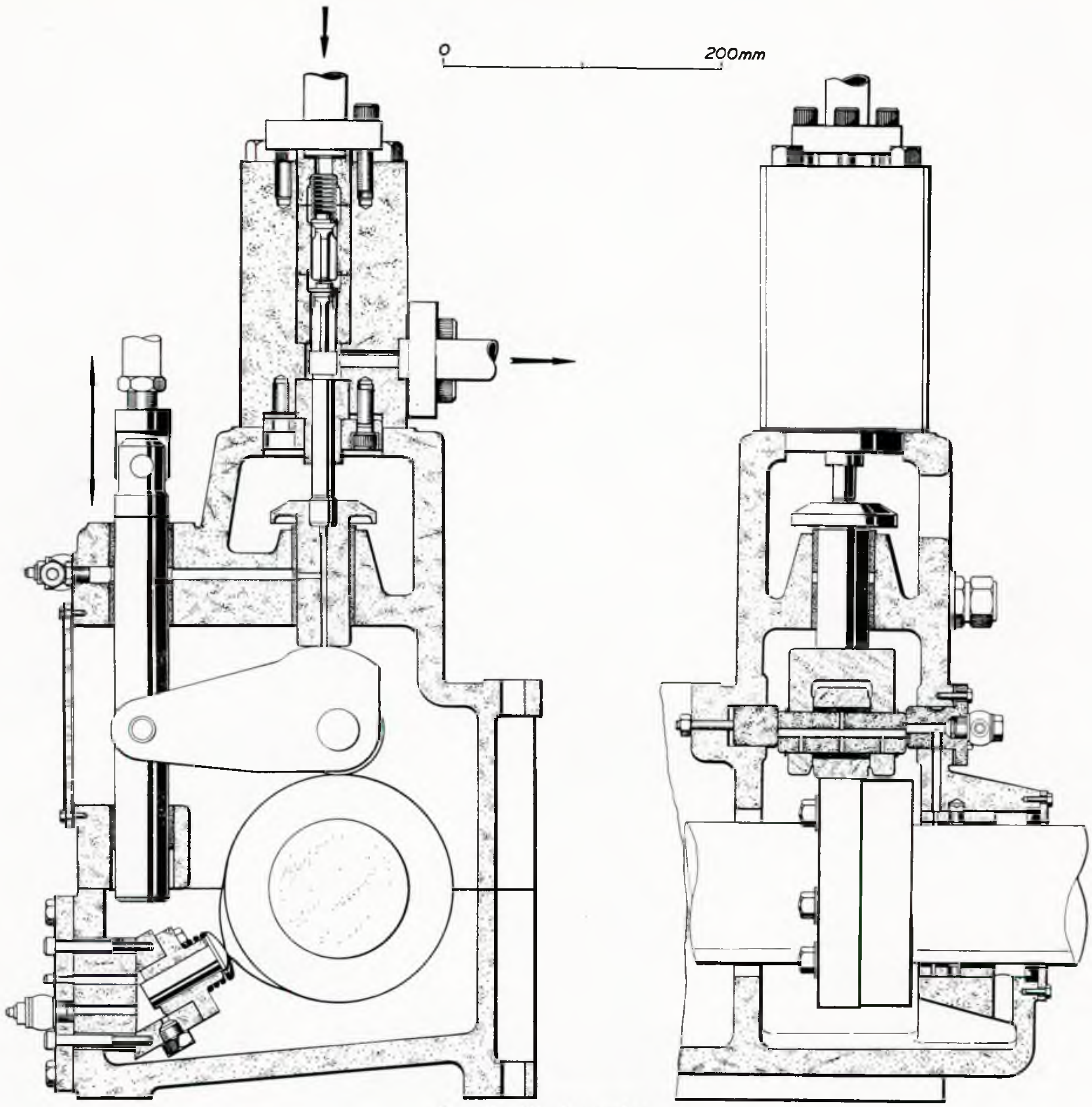


FIG. 9—*Timing valve*

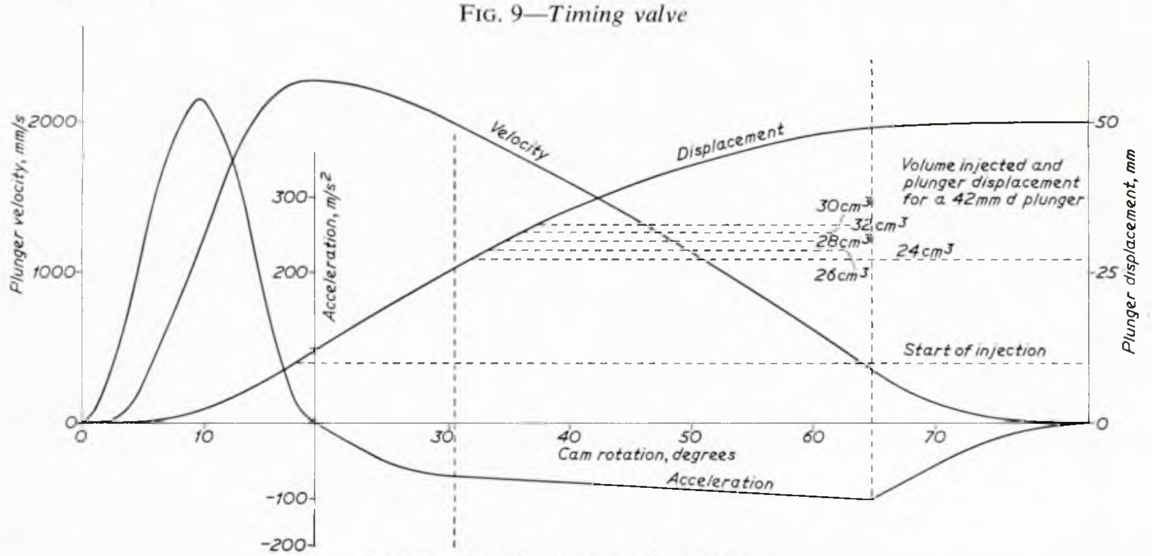


FIG. 10—*Jerk fuel pump characteristics*

TABLE II — LEADING PARAMETERS OF JERK PUMPS USED IN INITIAL TESTS ON PROTOTYPE ENGINE

Plunger diameter	42 mm
Total stroke of plunger	50 mm
Volume injected/stroke at m.c.r.	24 cc
Effective stroke of plunger at m.c.r.	17·3 mm
Static injection period at m.c.r.	14½°
Dynamic injection period at m.c.r. assuming 50 per cent spreadover	22°
Maximum plunger acceleration	43·8 g
Maximum plunger velocity	2280 mm/sec
Variation of plunger velocity during static injection period at m.c.r.	2270–2280–1930 mm/sec
Maximum instantaneous hp (metric) assuming a fuel pressure variation of 840–950 kg/cm² during injection	370
Mean hp (metric)	19
Maximum combined inertia and hydraulic load	13·5 tonnes
Maximum hertzian stress in cam	11 800 kg/cm²

Thermal Loading—Liner and Piston Cooling

Calculation based on Ref. 2 suggests that the density of heat flow rate (specific heat flux) in the new engine will be the same as on the 76J6 engine at maximum continuous rating because the smaller cylinder size and lower fuel/air ratio compensate for the higher speed and higher cylinder pressure. However, there is some disagreement about the part that radiation plays in heat flux in Diesel engines and there is some doubt about the degree of swirl necessary to procure complete combustion. Since the amount of swirl may also have a fairly important effect on heat flux it has been thought prudent to allow for some increase compared with existing engines.

Calculations of component temperatures and stresses have therefore been made on the basis of a mean flux to the pistons of 294 kW/m² (7 cal/cm² sec) and a maximum flux to the cylinder liner in the combustion area of 462 kW/m² (11 cal/cm² sec), both of which are about 12 per cent higher than those measured on existing J-engines.

Estimated temperature patterns in scavenge piston and liner are shown in Figs 11(a) and (b), based on this heat flux and temperatures measured on existing engines. It will be seen that the maximum crown temperature in the lower piston is expected to be about 500°C and the temperature at the firing ring about 160°C. These temperatures are quite satisfactory for a mild steel piston, but the maximum temperature of the surface in contact with cooling oil is rather above the carbonizing temperature of the oil, so it is necessary to have a high flow in this area to permit a vigorous scouring action and avoid any of the oil reaching the metal temperature.

This is achieved by having 24 cooling holes through the

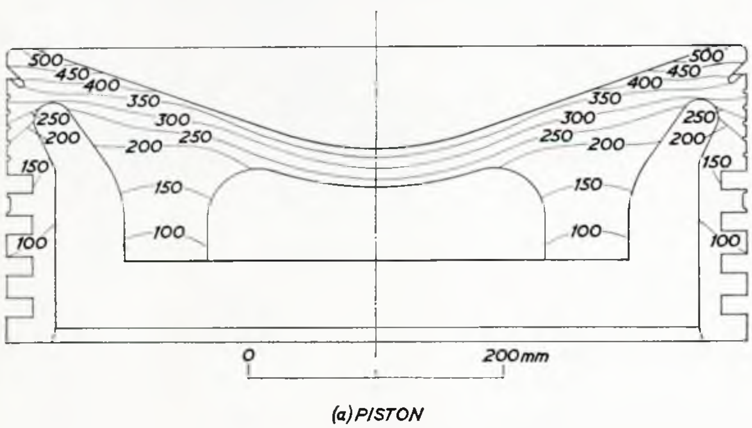


FIG. 11—*Combustion chamber isotherms*

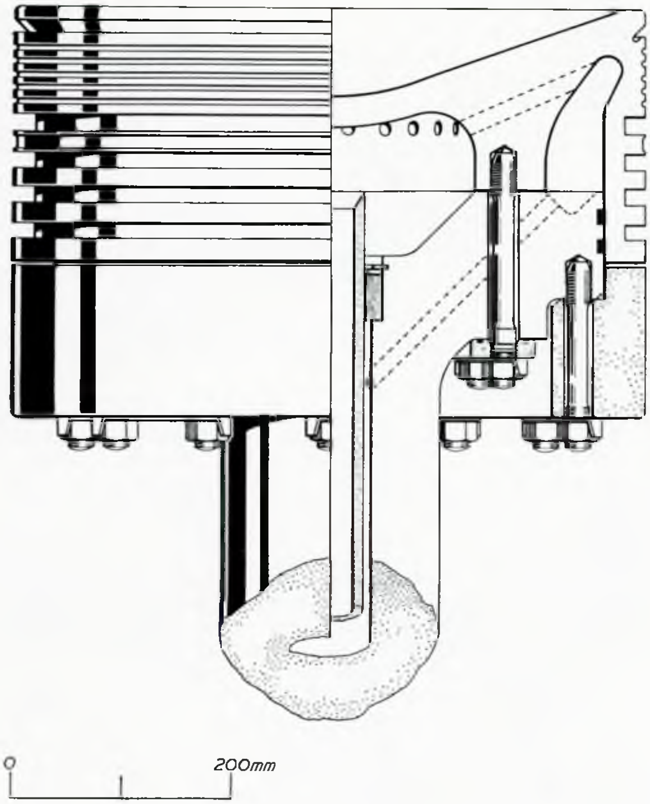


FIG. 12—*Lower piston assembly*

support ring of the piston crown carrying the oil from the central cooling space to the outer annulus. It is probable that the temperature figures given are pessimistic since the heat transmitted to the oil in these passages should provide more cooling than has been estimated. Extensive temperature measurements of a 67P engine showed that such cooling holes with a surface area only 9·5 per cent of the total piston area, wetted by coolant, carried away 38 per cent of the total heat input. In the new engine the surface area of the cooling passages is 22 per cent of the total wetted area of the cooling surface. Extra heat transfer from increased passage surface area plus a higher rate of heat transfer on the other surfaces due to increased "cocktail shaker" effect at higher speed may be expected to reduce the temperature difference between metal and coolant.

The upper and lower piston heads, shown in Fig. 12 as a part assembly of the lower piston, are interchangeable steel forgings generally similar to those used in previous engines. The computer calculations (Ref. 3) showed that under running conditions the thermal stresses would be very much greater than the pressure stress and that, consequently, an overall gain would be made by thinning the piston crown. This modification made it possible to dispense with the usual expansion grooves in the crown top, but the angled groove in the crown land has been retained as a heat dam, and the outer wall of the piston has been reduced in thickness above the ring belt to provide increased flexibility and to reduce still further the heat flow to the ring area. The one-piece cast iron liner, Fig. 13, is clamped between the exhaust belt and a lower cast iron jacket bolted to the entablature top face. The design is such that the complete cylinder assembly with all valves, lubricator quills and pipes, and exhaust belt and pipes, can be withdrawn from the engine as one piece. The short exhaust pipes at front and back of each exhaust belt are connected to adjacent cylinders by flexible pipes.

The maximum estimated temperature in the gas side liner wall in the combustion area is only 250°C which is entirely

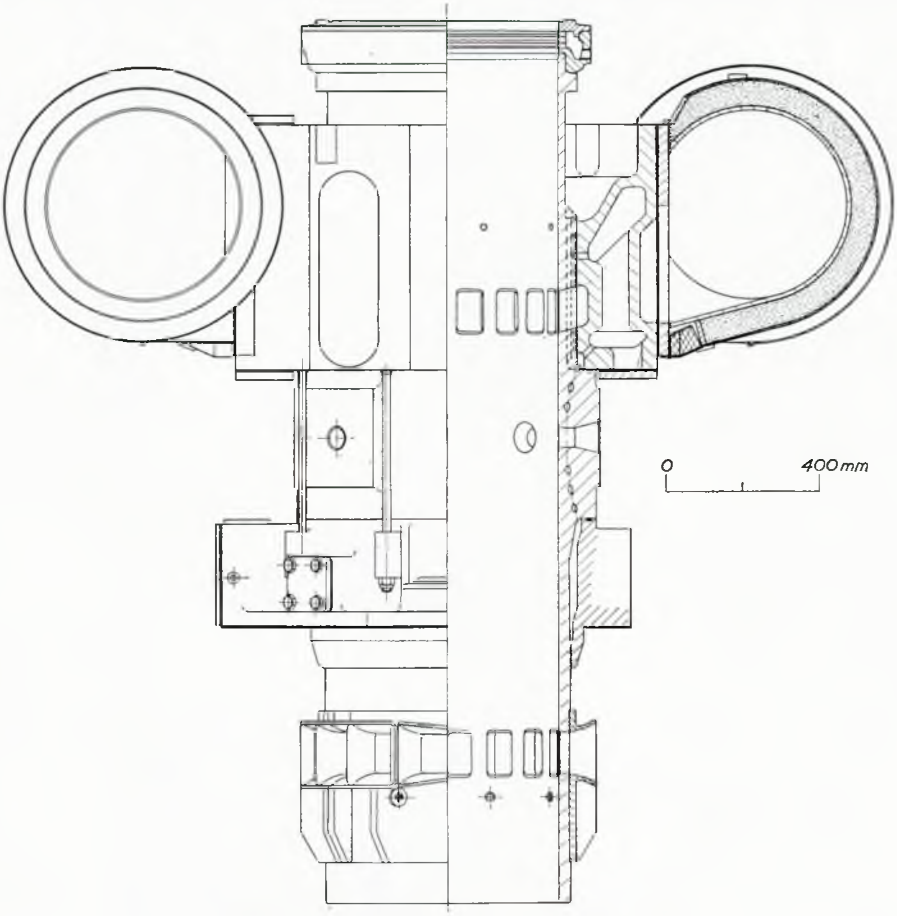


FIG. 13—*Cylinder assembly*

satisfactory for the vanadium titanium cast iron used. This low temperature is due to the method of cooling by 56 oblique tangential cooling holes which reduce the effective heat path length between gas side and water side to only 1·8 cm, which is much less than can be attained by any other means. The cooling effect is deliberately reduced away from the central combustion area to make the liner temperature as even as possible. In way of the piston ring travel the surface temperature of the liner is expected to range from 180°C at the upper limit down to 40°C at the bottom of the liner below the scavenge ports.

From these estimated temperatures the thermal and pressure stresses have been calculated by a computer programme known as FEX developed by the Berkeley Nuclear Laboratories of CEGB, England from a method derived at the University of California (Refs 3 and 4). These calculations give a maximum Von Mises equivalent stress in the centre of the piston crown of 524 MN/m² (76 000 lb/in²), and in the cylinder liner 166 MN/m² (24 000 lb/in²). Comparison of these stresses with the fatigue strength of the materials suggests that the piston should have a fatigue life of around thirty years, allowing for ten full thermal cycles (cold to MCR) per day. On the same basis a life of over one hundred years can be expected from the liner.

Experience in previous engines shows that the exhaust piston will have slightly lower temperatures than the scavenge piston because the better cooling effect of water overbalances the difference in heating effect of the hot exhaust gas flowing over the piston crown compared with the cool scavenge air on the lower piston.

Due to the relatively high speed of the lower piston, there is a problem in getting the cooling oil in and out of the piston but because of the short stroke it has been possible to make the main guide shoe longer than the stroke. This makes it possible to pass oil through the shoe from an opening in the side of one guide bar into a groove in the side face of the shoe, and from there through a drilling in the crosshead bracket to the lower end of the piston rod. The piston rod has a central bore with an internal tube and the oil passes up the tube to the inner space in the piston crown and through the 24 holes in the piston supporting annulus to the outer annular cooling space adjacent to the ring path, then back down the piston rod through the space around the central tube. At the lower end of the piston rod the oil passes through another drilling to a groove in the other side face of the crosshead shoe and then through the other main guide bar by a fixed drain connexion with a sight glass, to the sump. This method avoids the need for either telescopic pipes or swinging links. Leakage of oil at the guide face serves to lubricate these parts and has been calculated to be small enough to be unimportant in the total oil flow.

Water supply and discharge to and from the upper piston is through swinging links, Fig. 14. As compared with telescopic pipes this arrangement has several advantages. The rubbing speed of the glands is only one-fifteenth of the piston speed, the circumferential oscillatory movement at the gland does not carry in any dirt or dried leakage products to cause scoring, and any wear that occurs at the gland is in a circumferential direction and, therefore, at right angles to the direction of leakage. In addition, the reciprocating mass of the cooling gear is much less than can be attained with telescopic pipes.

The swinging links oscillate on water lubricated plastic bearings running on stainless steel spindles. At the end of the swinging links one hollow spindle for each link is rigidly attached to the transverse beam which is bored to carry the water to and from the piston in a similar way to the arrangement for the lower piston. The hollow spindle at the other end of the link connects to a stand pipe consisting of a flexible hose inside a system of links and trunions which carry the inertia loads but permit the necessary horizontal movement due to angularity of the swinging link, as well as small movements needed to accommodate misalignment of the transverse beam, resulting from normal piston movement within the bore clearance.

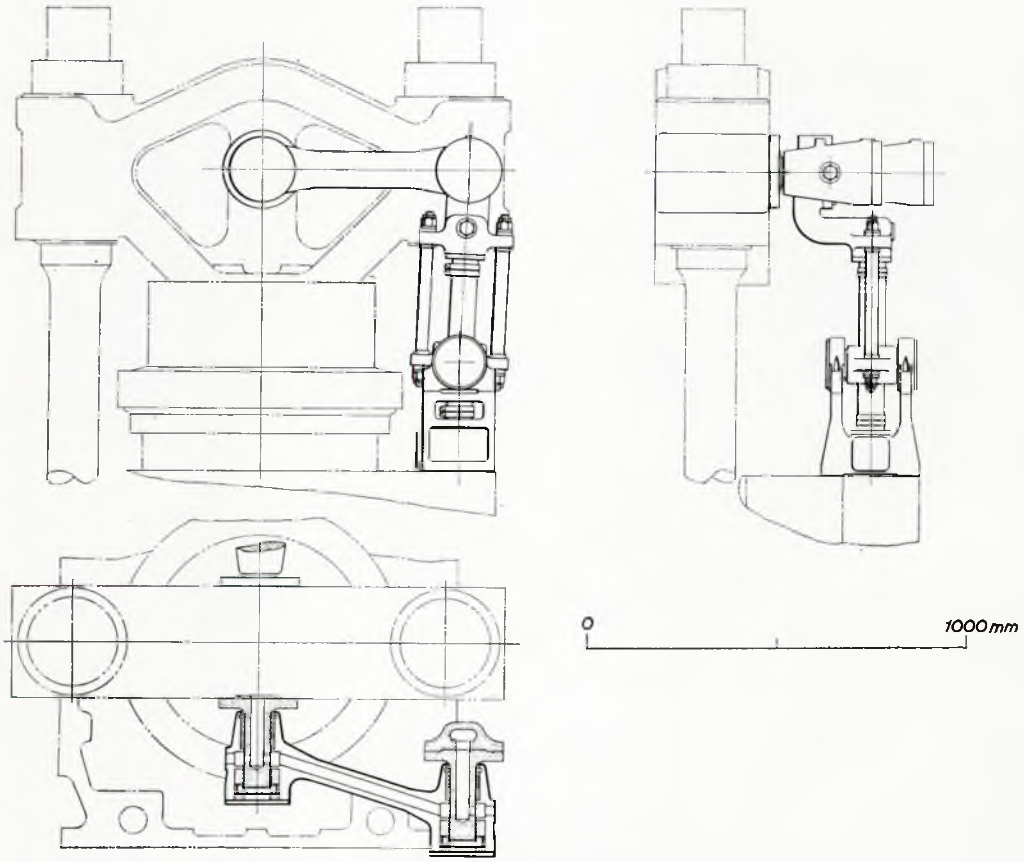


FIG. 14—*Upper piston cooling water linkage*

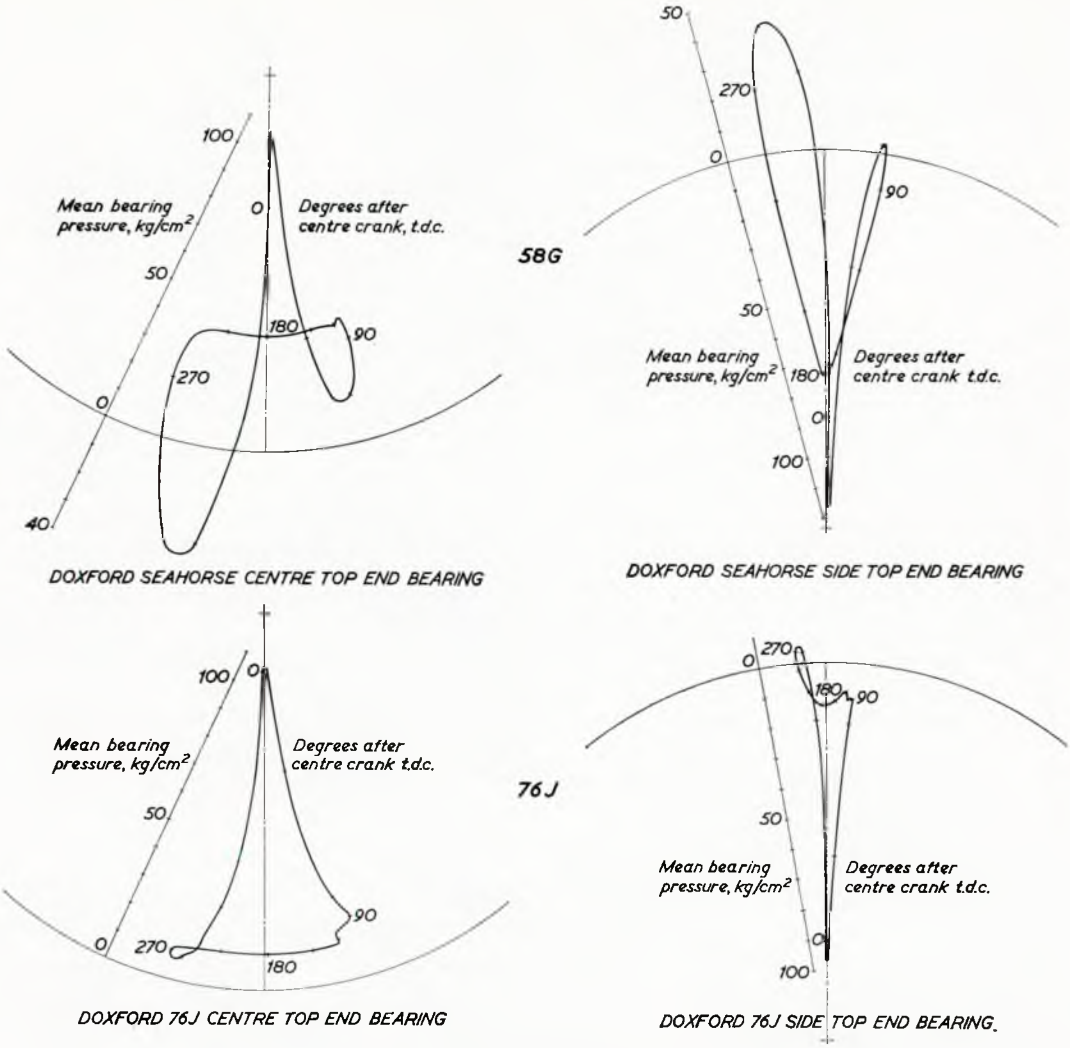


Fig. 15—Side and centre top end bearing polar loading diagrams

The links and trunions are so arranged that the flexible hose bends into a smooth arc with an angular movement of only $\pm 2{\cdot}5$ degrees.

MECHANICAL LOADING

All the important bearings in the engine have thin steel shells lined with white metal. This type of bearing has the advantage that because of close control of materials and manufacturing methods, the bond between white metal and steel is consistently good, which helps to give long life. In addition, inspection and maintenance are much easier than with bearings having white metal cast directly into the housing. There are, however, some problems; the bore of the housing for the bearing shell must be machined extremely accurately and neither it nor the back of the bearing may be scraped or fitted by filing. This care is essential to preserve good thermal contact between the bearing shell and the housing, which is achieved by having a closely toleranced "nip" or interference fit. The nip results in a compressive hoop stress in the bearing shell of upwards of 69 MN/m² (10 000 lb/in²) which calls for increased load in the clamping bolts.

The most sensitive bearing in a normal two-cycle engine is the centre top end. In this engine, however, the higher rotational speed, and the crosshead construction, result in sufficient inertia of the piston, piston rod, and crosshead, to provide a reversal during each cycle. Fig. 15 shows polar loading diagrams for centre and side top end bearings with corresponding diagrams for a 76J engine, shown below. It will be seen that the load pattern for the centre bearing is very similar to that for the side top end bearing of a normal engine, and experience shows that these side bearings are not subject to deterioration in service and have a very long life. It is also common experience that top end bearings on four-stroke engines, where there is a similar load reversal, can tolerate much higher specific bearing loads than those of normal two-cycle engines. The calculation indicates that the load reversal takes place from full speed down to 90 rev/min, with propeller law engine load, and from 160 rev/min upwards with the maximum load which is likely to occur during acceleration.

The specific service load which can be carried by any bearing tends to increase with rotational and rubbing speed. For this reason it has become the practice with new designs to calculate not only the polar load diagram for each bearing but also the variation of oil film thickness during the cycle. Fig. 16 shows the results of this calculation for the main and bottom end bearings based on hydrodynamic film theory and in the case of the main bearing using a simplified analysis of bearing loading. The actual minimum film thicknesses are probably somewhat greater than those shown because of the effect of displacement between the journal and bearing producing additional film thickness due to "squeeze" effect. Unfortunately, in the case of the top end bearings, due to the preponderance of the squeeze effect, it is not yet possible to produce a satisfactory calculation of film thickness.

The crankshaft is of conventional semi-built design with forged main webs and pins. The side crankpins, which are of greater diameter than is needed for strength, are forged integral with main journals which have a radius rather less than the sum

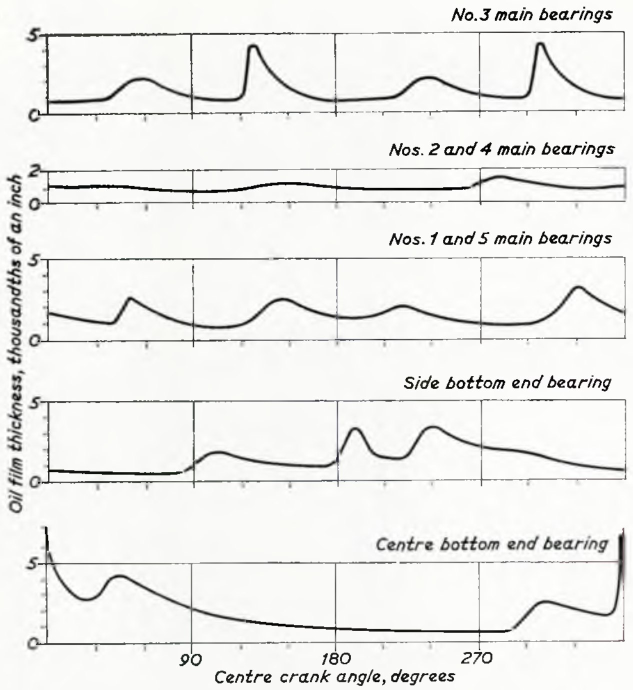


FIG. 16—*Main and bottom end bearing oil film thicknesses*

of the side crankpin throw and radius. This construction keeps down the rotating inertia and allows the main bearings and keeps to be smaller, so facilitating handling.

No premium is paid for adopting a uniform crankshaft pin diameter throughout the range since the maximum stress in the seven cylinder engine is only very slightly in excess of that for the four.

The centre connecting rod is also conventional and has the top end bearing shown in Fig. 17 with a one-piece lower bearing shell providing bearing surface over the full length of the crosshead pin, broken only by the oil supply grooves. Oil is fed from the piston cooling supply in the crosshead bracket through two holes in the crosshead pin near its outer ends. The grooves in the shell are arranged so that the oil flows towards the centre where it passes down a drilling in the connecting rod to the bottom end bearing. During the upper part of the piston stroke, when the connecting rod is subject to acceleration of up to 560 m/s^2 (57g), the column of oil in the rod bore acts as a reservoir to enhance the oil pressure in the top end bearing during the period of maximum load. The forced flow through the oil grooves along the bearing assists cooling and ensures a plentiful supply of lubricant to the bearing surface throughout the cycle.

The conventional connecting rod construction, with a central strut and bearing clamping bolts at each end, is logical for a rod subject mainly to compressive load but not for the side connecting rods of an opposed piston engine, which have to carry a heavy tensile load but a relatively light compressive load.

The essential parts of such a connecting rod are a strong keep above the top end pin, a strong keep below the crankpin, and tensile members between the two to carry the main load. Inner half bearings are needed to retain the oil and have to be separated by a relatively light member to carry the small compressive load.

The design shown in Figs 18(a) and (b) is developed from this principle, but the compression member, incorporating the two inner half bearings, is a rigid steel casting designed to maintain bearing shape. The design also provides a stiff abutment to reduce the stress range in the bolts and thus give a greater margin against fatigue failure. Since the tensile load is carried by the through bolts the heavy bending stresses are avoided which occur in a normal connecting rod end, where the load is carried from the central strut to the bolt fixing. The rod can be readily removed through the front of the engine when required but for normal survey purposes it is only necessary to lift the keeps and turn out the thin shells.

An example of the effect of bearing shell compression on clamping bolt load is shown by this connecting rod. The load per rod during acceleration is 1370 kN (137 tons) reduced to 1000 kN (100 tons) by inertia at full speed. Two bolts have to carry this load plus the 240 kN (24 tons) necessary to compress the shells and ensure that the abutments remain in contact at all times. To produce equilibrium between the loads acting on the bearing keep, requires a compressive stress at the abutment increasing from the inside position next to the bearing shell to a maximum at the extreme outer position, Fig. 18b. The amount that the abutment load changes when firing load is applied to the keep depends on the relative stiffnesses of the abutments and of the bearing keep as a beam. It is clear that the closer the bolts are to the bearing shell and the further away is the outer heel of the abutment, then the lower is the load needed in the bolts.

It will be noted that the side rod has a top end bearing of the same diameter and length as that of the bottom end.

A rather extensive calculation showed that the through bolt construction used with a crosshead pin 410 mm diameter but with the bolt centres 630 mm apart, to suit the 520 mm diameter crankpin, required an initial load in each bolt of 2000 kN (200 tons) increasing to 2280 kN (228 tons) with the firing load. Increasing the crosshead pin diameter to 520 mm, as for the bottom end, reduced the necessary initial bolt load to 1250 kN (125 tons), increasing to 1530 kN (153 tons) with firing load, whilst still keeping the abutments in contact both with and without firing load, and maintaining the necessary compressive stress in the bearing shell. It is interesting to note that even with optimized design the total load in the two clamping bolts is more than twice the firing load.

BALANCE

To appreciate the thinking which determined the balancing arrangements, it is necessary to refer back to the characteristics of the alternative single piston crosshead engines. The out-of-

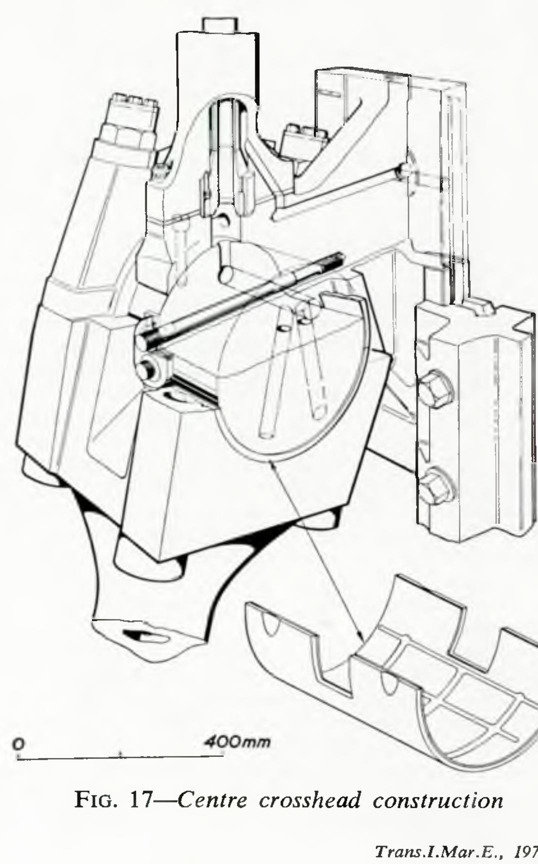


FIG. 17—*Centre crosshead construction*

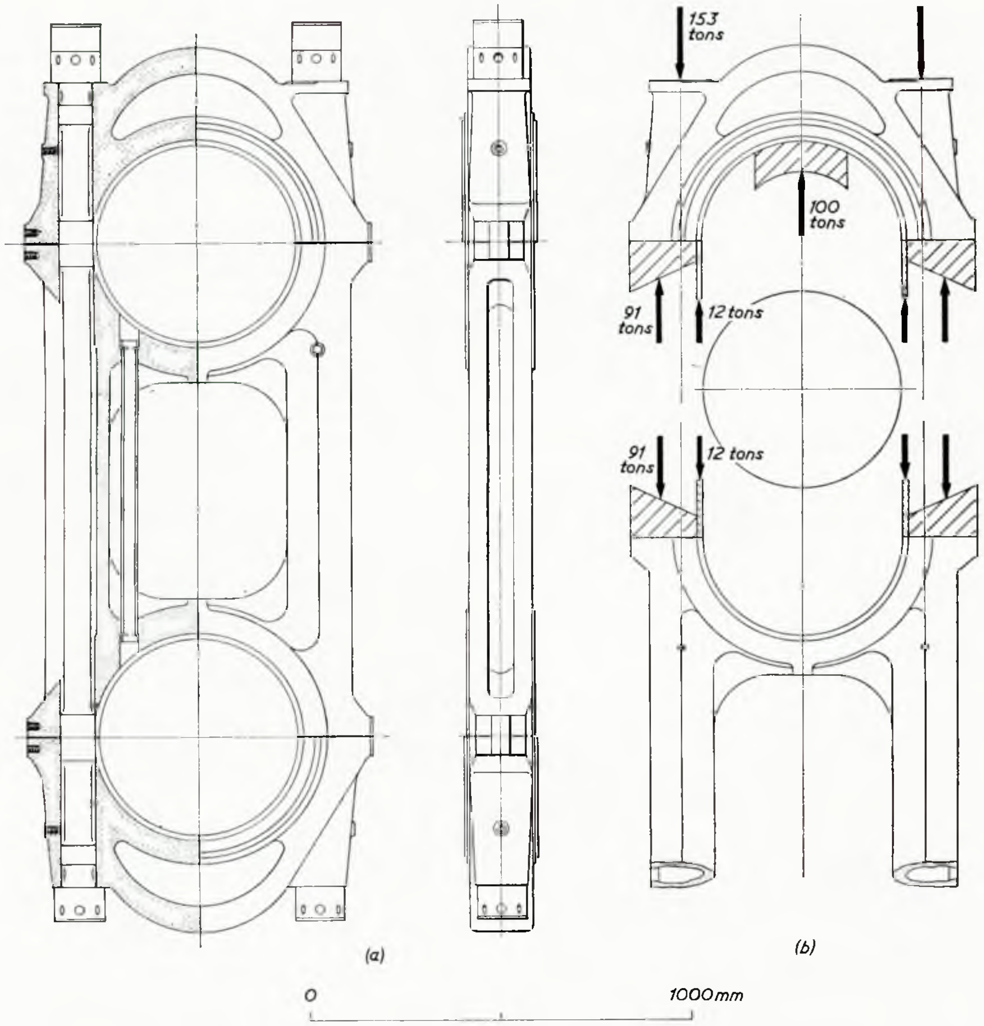


FIG. 18—*Side connecting rod construction (a) general arrangement (b) abutment forces*

balance forces of the hypothetical engine of only 2000 hp/cylinder whose principal particulars are set out in Table I would be approximately as follows:—

Primary Reciprocating	102 tonnef (1·00 MN)
Secondary Reciprocating	28 tonnef (0·28 MN)

Table III illustrates the external unbalanced couples which would result from both a two- and four-stroke version of a design of these proportions, in comparison with the figures for the Seahorse. In its 6-, 7- and 8-cylinder builds, the single piston engine operating on the four-stroke cycle is acceptable but the number of major orders in its torsional vibration spectrum might well rule out this working principle. Quite apart from this, both the two- and four-stroke versions are quite unacceptable in the five cylinder form, with an external couple of 183 tonnef m (1·79 MN m) associated with an 180 tonne engine.

In a single-crank opposed piston engine, the main bearings are not subjected to firing loads and the inertia loads are approximately half those of the single piston alternative. These two factors enable the crankshaft alignment to be maintained within the limits defined in the section of the paper dealing with the structure.

An opposed piston engine in which the upper and lower pistons are exactly 180° out of phase, can be arranged to be in complete primary balance per cylinder by equating the product of each stroke and its associated reciprocating mass. In the Seahorse engine, however, the lead given to the exhaust piston, to extend the blow down period, results in the force vectors for the two pistons being displaced 8° from true opposition. In consequence, there is a small out-of-balance primary reciprocating force per cylinder, acting approximately at right angles to the centre crank vector and amounting to about 0·2 MN (20 tonnef).

It was considered worthwhile to introduce compensation to achieve complete freedom from any external out-of-balance couples due to this small resultant primary force. This is done by a combination of forward and counter rotating balance masses. The mass rotating in the same sense as the crankshaft compensating one half of the primary reciprocating force per cylinder is amalgamated with that required to ensure rotational balance of the crank section with its appropriate element of running gear, and the resultant mass distribution accounts for the asymmetrical appearance of the centre crankweb. The counter rotating mass is driven by the shaft which exists to transmit power to the auxiliary blower, as well as to the valve gear and fuel injection equipment. Rather than developing the necessary individual rotational forces of 0·1 MN (10 tonnef) needed at each cylinder, suitably sized rotating masses are fitted at each end of the engine to neutralize the resultant couple.

The secondary forces arising at each cylinder amount to 0·54 MN (55 tonnef) and cannot readily be compensated individually. However, a crank sequence can be chosen which minimizes any resultant external couples since the use of constant pressure turbo-charging and the partial at-source compensation of the primary forces, afford freedom of choice in the selection of firing orders. As will be seen from Tables III(a) and III(b), only in the case of the five-cylinder engine, is the secondary

TABLE IIIA—UNBALANCED EXTERNAL COUPLES (SI UNITS) *comparison of the Seahorse with a single piston engine having a bore and stroke of 715 and 800 mm respectively, developing 2000 metric hp per cylinder at 256 rev/min.*

	Reciprocating Couple at Rated Speed (MN m)				
	Primary				
	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	—
	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	—

TABLE IIIB UNBALANCED EXTERNAL COUPLES (IMPERIAL UNITS)

	Reciprocating Couple at Rated Speed (tonf ft).				
	Primary				
	4 Cyl.	*5 Cyl.*	*6 Cyl.*	*7 Cyl.*	*8 Cyl.*
Single piston 4-stroke	—	198	0	119	0
Single piston 2-stroke	—	198	0	119	198
Seahorse	0	0	0	0	—
	Secondary				
Single piston 4-stroke	—	589	0	119	0
Single piston 2-stroke	—	589	408	119	0
Seahorse	0	112	0	20	—

couple of any significance and even this can be mitigated by the adoption of uneven firing intervals.

ENGINE STRUCTURE

An engine structure requires a high degree of rigidity for the following reasons:

1) to maintain the alignment of the moving parts within acceptable limits; such limits being determined by considerations of (a) bearing loadings (b) avoidance of fatigue failures in rotating components;
2) to ensure that the natural frequencies of the structure do not resonate with the exciting forces within the working range.

In the past, manufacturers of marine engines have tended to rely on an inherently rigid foundation being provided for their products. To a considerable extent, this relieved the engine designer of problems of bearing failure due to changes in support reaction arising from deformation of the machinery structure. If the design loadings were reasonable, trouble free service could be expected, provided highly skilled personnel could be relied upon to install the machinery in a manner that accurately reproduced the alignment that obtained during shop erection and testing.

Constant commerical pressures have resulted in the more general use of high tensile steels, revised classification society requirements and increases in the size of ships and their associated engines. These factors have led to generally more flexible construction which has, in turn, led to alignment problems with large conventional Diesel engines.

Furthermore, there is a considerable history of bearing and crankshaft failure which should rightly be attributed to thermal distortion. In the case of a seagoing vessel, the differential temperature between the hot lubricating oil in the engine and the cold sea water in contact with the ship's shell plates can amount to 50°C—with an engine of 30 m length this gives rise to a differential expansion of 18 mm.

For these reasons it was decided that the Seahorse engine should be isolated from the foundation and have sufficient inherent rigidity to meet the requirements stated above.

The rigidity of a crankshaft, considered as the load required to produce unit deflexion at the free end of a cantilever of length equal to one main bearing span, is in a series of geometrically similar engines, proportional to the crank diameter. Though this relationship makes alignment no more onerous in the larger engine, the alignment requirement becomes rapidly more critical for highly rated multi-cylinder engines since the crank diameter for a specific engine is determined by the firing loads and total torque transmitted.

In the Seahorse engine, the rigidity of the crankshaft is such that a 1/1000 in deflexion of a main bearing results in a change in the specific bearing load of approximately 75 lb/in^2 (1 μm:20 kN/m^2). This relationship created a design pre-requisite for the engine structure that under working loads the deflexion of the main bearings would not exceed 1/100 in (254 μm); equivalent to a change in specific bearing loading of 750 lb/in^2 (5·17 MN/m^2).

To achieve a rigidity of this order it was soon apparent that the bedplate and entablature must be joined by a structure that would enable the complete assembly to be treated as a box girder. In this way it is possible to achieve an improvement of the order of twentyfold in the rigidity of the structure, in both the torsional and longitudinal bending modes, as compared to the sum effect of the individual components.

The adoption of a box structure naturally complicates access to the internal components. Accordingly, the engine frame is essentially in the form of a letter C but with the gap closed by a cast steel cross braced structure designed to give the maximum freedom of access. To allow the use of diagonal members, movable sections were adopted, as shown in Figs 19 and 20, to facilitate the removal of certain main bearing keeps and side connecting rods. These movable sections were so designed that they automatically restore the original alignment when the clamping plates are retightened, even should the framework relax on their removal.

Model Testing

Because of the difficulty in analysing the structure mathematically, a model of the engine frame was made from Vybak rigid P.V.C. material to a scale of 1 to 5·7 and tested at Lloyds Research Department (Fig. 21). Deflexion curves were plotted for specified points to establish the effect of a series of loads simulating the reactions of the running gear on the engine structure.

These tests confirmed that the maximum main bearing deflexions likely to be encountered in the full-scale engine would be within the design criteria.

Additionally a one-tenth scale solid steel model of the crankshaft was tested at Newcastle University (Fig. 22). The analysis of these results has shown that the deflexion at any point can be considered as arising from three distinct modes of flexure, namely pure bending, pure torsion and "corkscrew" bending, all of which can be expressed mathematically in terms of coefficients and crank angles. In the course of the testing, proximity gauges positioned between the webs were used to measure the crank deflexions and a multiple regression analysis was carried out, which led to a relationship between web deflexions and journal displacements being established.

Comparison of both model tests showed that a frame rigidity approximately a hundred times greater than the crankshaft rigidity had been achieved. Accordingly, it was considered reasonable to ignore the polar variations in crankshaft stiffness when computing the main bearing loads. These loads have been calculated on two premises: first in the conventional manner which assumes that the crankshaft is composed of individual sections (this treatment is in effect equivalent to assuming that the engine structure has an infinite rigidity) and secondly, on the assumption that the crankshaft is a continuous beam of uniform modulus in an elastic structure.

The results of both calculations are shown in Fig. 23 for each of the unique loading patterns arising in a 4-cylinder engine, which incidentally encompass the most onerous conditions in the entire range.

The more refined treatment shows that the actual loads likely to be experienced in service are substantially different from those predicted by the simplified approach.

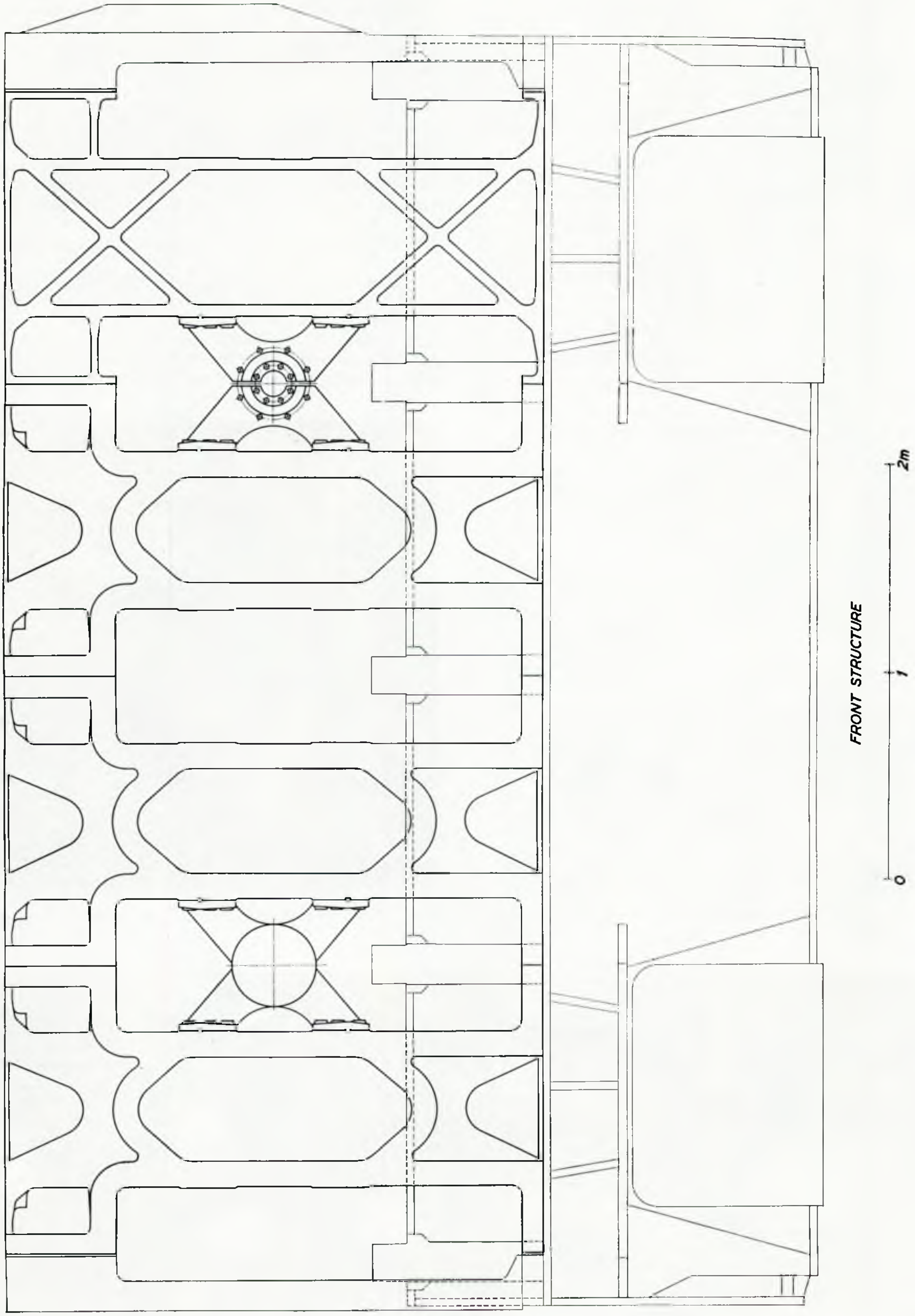


FIG. 19

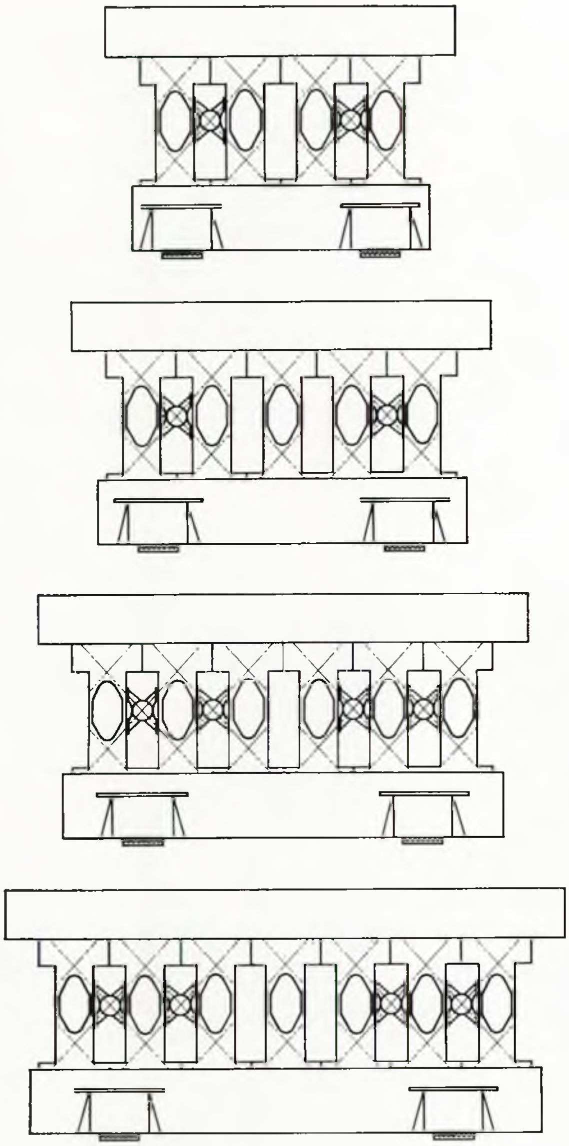


FIG. 20—*Front structure key diagram*

Mounting Arrangements

The engine will be supported on four mounts each incorporating a special design of resilient expansion chock. This method of mounting will accommodate differential thermal expansion between the bedplate and the underbed and will insulate the engine from any bending distortion of the foundation whilst maintaining external alignment with the drive.

The chocks themselves consist of semi-circular sandwiches of cast iron with rubber filling of oil resistant grade. The mating surfaces comprise a series of interlocking parallel corrugations separated by the thickness of rubber which is bonded to both halves. This construction results in a high degree of rigidity normal to the axes of the corrugations in both the vertical and horizontal planes, where the filling is virtually in direct compression, and a very low stiffness along the corrugations where the rubber is in shear, the relative stiffness figures being:

Vertical loads	100
Horizontal loads along corrugations	1
Horizontal loads normal to corrugations	1000

Fig. 24(a) shows in detail a complete mount, which includes long holding down bolts pre-tensioned to resist vertical loads and capable of accommodating horizontal displacement without adding significantly to the minimum stiffness of the chock.

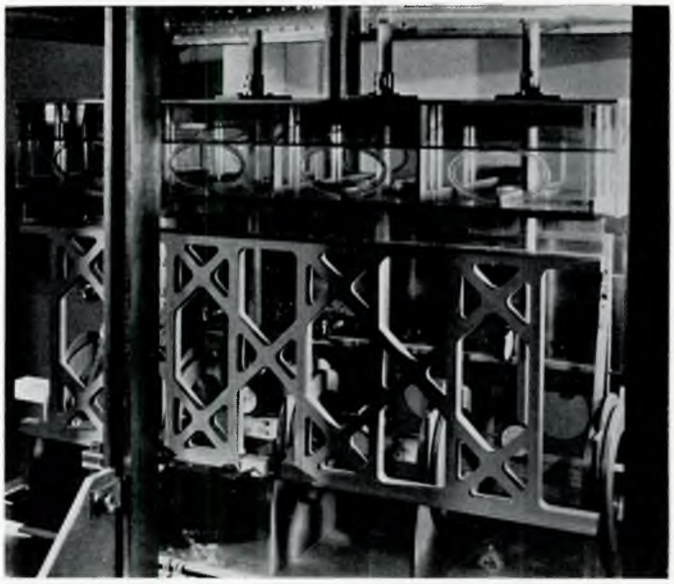


FIG. 21.—*Vybac model engine structure test rig*

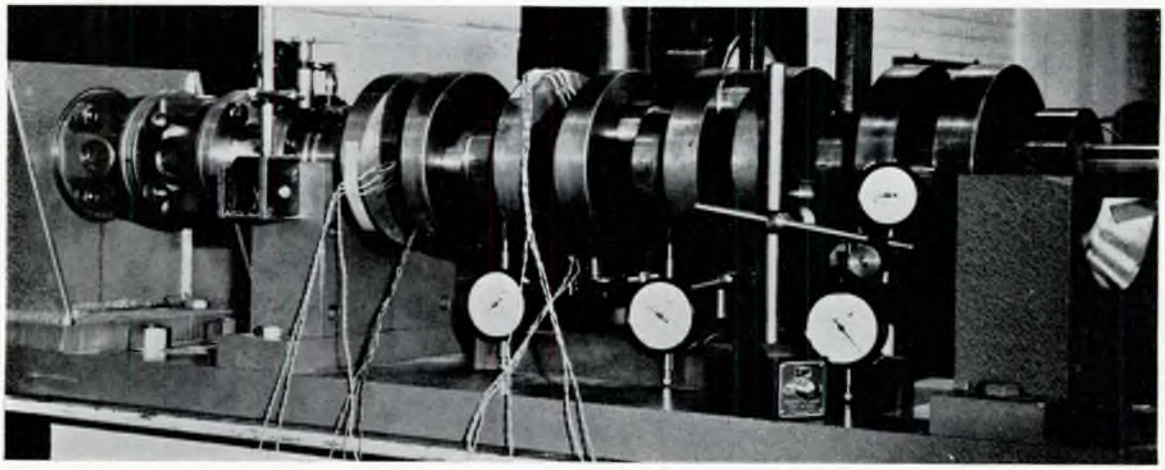
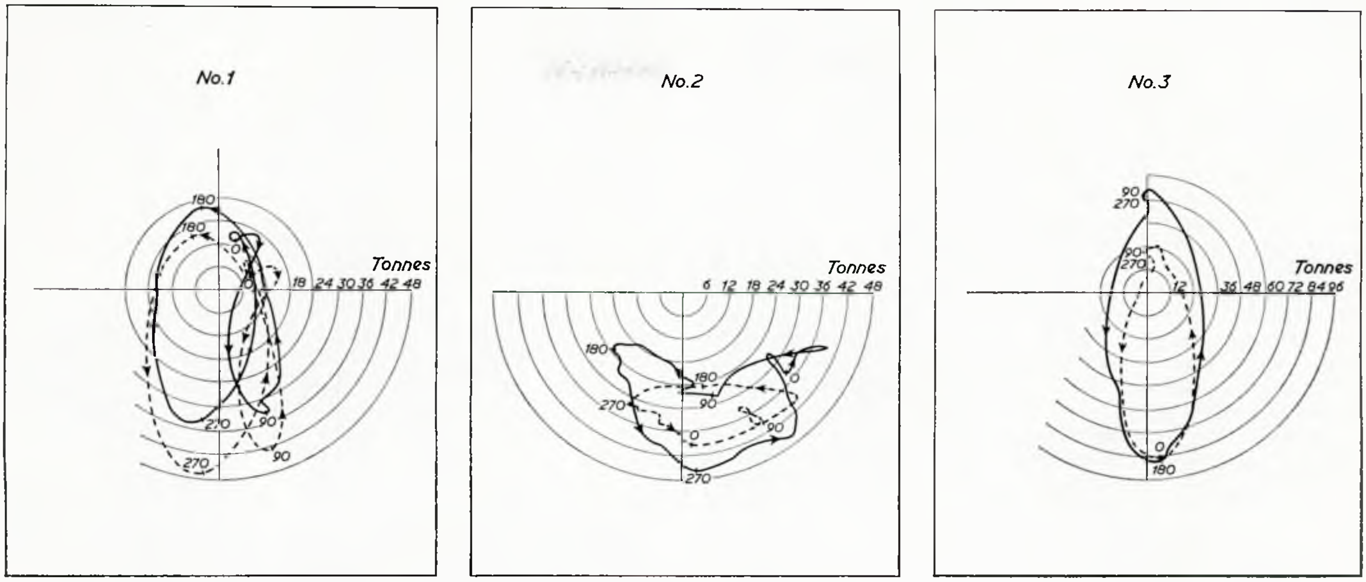


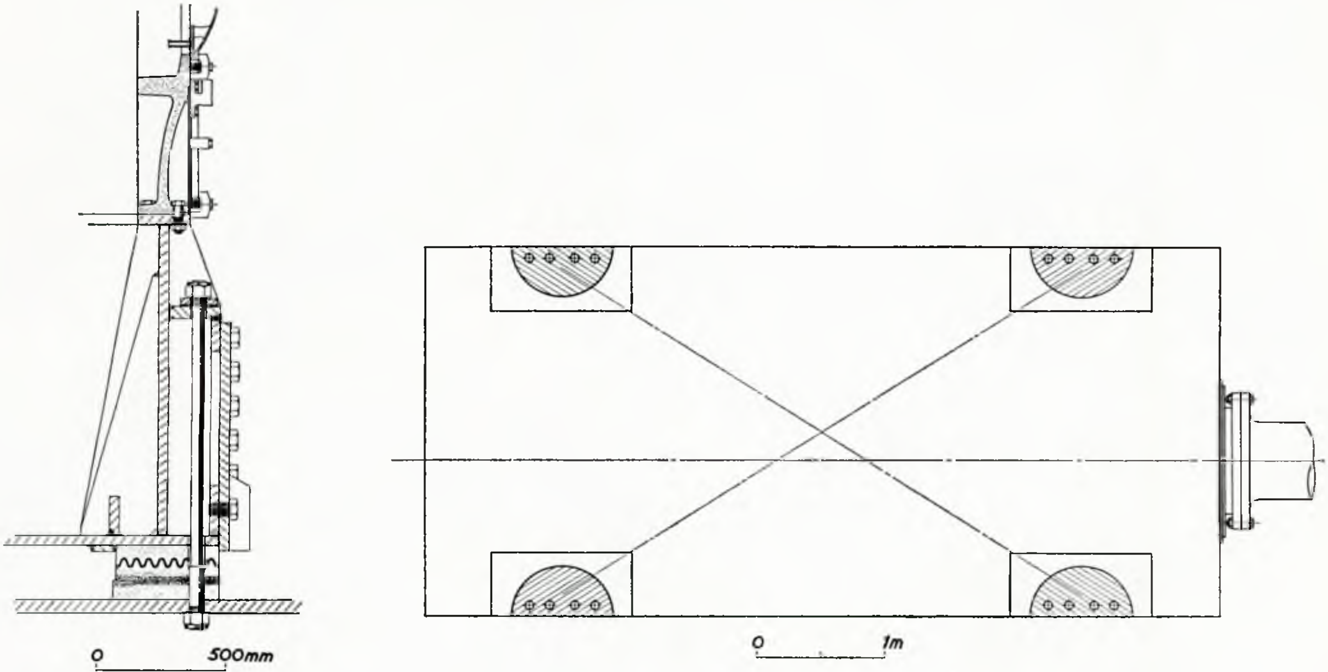
FIG. 22—*Model crankshaft test rig*



——— *Analysis based on an infinitely stiff structure with crankshaft pin jointed at each main bearing*

- - - - - *Analysis based on structure and crankshaft stiffnesses obtained from model tests*

FIG. 23—*Nos 1, 2, and 3 main bearing polar loading diagrams*



(a) detail of foot and expansion chock *(b) key plan*

FIG. 24—*Support arrangements*

The disposition of the chocks can be seen in Fig. 24(b).

The corrugations align with the diagonals through the mounts, allowing virtually free radial thermal expansion from the point of intersection but with no displacement of the engine centre lines. Since, in most applications, there will be a flexible coupling in the drive, the slight axial movement of the engine output flange with the arrangement shown, is of no consequence. Where this is important, the chock corrugations could be orientated to radiate from a fixed point under the engine coupling with no disadvantage other than the loss of reverse symmetry.

The longitudinal position of the feet has been chosen to give the minimum bending moment in the structure due to the weight of the engine.

AUXILIARY BLOWER MECHANICAL DRIVE

The auxiliary blower is driven via a speed increasing gearbox from the free end of the balance shaft, which rotates at 300 rev/min. The auxiliary blower itself is a centrifugal compressor as used in an existing turbo-charger with an appropriate pressure ratio and having a service speed of 22 500 rev/min. The speed-increasing gearing between the balance shaft and blower, therefore, has an overall ratio of 75 to 1 and, furthermore, is arranged so that the direction of rotation of the compressor is the same whether the engine is running ahead or astern.

Two designs of gearbox have been developed to achieve this end, employing slightly different principles, so that should any difficulty develop with one of the designs the general programme of engine testing will not be delayed. The gearbox fitted to the prototype engine for the initial tests employs the flexible planetary pin principle. This gearbox has four stages, all of which are employed whichever the direction of rotation of the main engine. Reversal is effected by two S.S.S. clutches, the appropriate one engaging on reversal of torque. An hydraulic coupling is included between the third and fourth stages to isolate the inertia of the high speed train and impeller. In the alternative design, only the first stage gearing is of the planetary type. This achieves load sharing by means of a floating annulus driven by a toothed flexible coupling: the remaining gear trains are of the multilayshaft type employing quillshafts to ensure even torque distribution.

This gearbox includes five separate stages but only four of these are used in either direction. Two hydraulic couplings are incorporated, one for ahead operation and one for astern running—reversal being effected by filling the appropriate coupling. This latter design avoids the use of S.S.S. clutches or ratchets and employs throughout conventional engineering that is tried and proven in all respects. It is, however, substantially more expensive than the alternative design. The engine control system incorporates equipment to ensure that the appropriate coupling is filled as soon as the required direction of rotation is selected. Special quick emptying valves are incorporated in the couplings to facilitate rapid changeover from ahead to astern and vice versa which obviate the possibility of both the ahead and astern couplings being active during the changeover. The gearboxes are designed to transmit a maximum of 550 hp and a careful analysis has been made of their torsional characteristics with special attention to the transient condition under rapid engine acceleration.

AIR STARTING

The engine is reversible and uses starting air at a pressure of 31 bar (450 lb/in^2). Tests of the pilot air-operated cylinder starting valves on current engines have shown that there can be a delay of as much as three-quarters of a crankshaft degree per engine rev/min between the opening point of the distributor and the opening of the valve in the cylinder, i.e. at thirty rev/min on starting air, there can be a lag of twenty-two degrees. To give more precise control, the cylinder starting air valve on this engine is fitted on the camshaft side of the cylinder and is operated mechanically, by levers actuated by ahead and astern cams. The two levers for each cylinder are mounted on eccentric fulcra and are brought into operating positions by means of low pressure pilot air cylinders (Fig. 25). The precise timing economises starting air consumption and enables astern air to be used for braking when manoeuvring from ahead to astern.

VIBRATION AND NOISE

Any piston engine is subject to torsional and axial crankshaft vibrations and transverse vibration of the engine structure due to varying guide forces and couples. In addition, high frequency components of the gas pressure variation and of the force pattern of mechanical parts such as fuel pumps, can result in airborne and structure-borne noise.

The torsional vibration characteristics of this engine are shown in Fig. 26 for four- to seven-cylinder configurations. The gear drive to the auxiliary blower at the forward end of the crankshaft is taken through the flexible coupling shown in Fig. 27 which is designed with stiffness to give a system frequency, with the node in the coupling, of 210 vibrations per minute. The coupling thus serves to isolate the auxiliary drive from crankshaft vibrations of any frequency above 400 vibrations per minute. In marine applications the drive between engine and main gears is taken through a quill shaft which has sufficient flexibility to provide isolation for the main crankshaft frequencies.

For marine applications there are primarily two types of torsional vibration to be considered. In the first type there will be one or more nodes in the shafting and gearing system between the engines and propeller and the resonant frequencies will depend mainly on the inertias of engine, gearing, and propeller, and shafting flexibility. With a quill shaft drive, adjustment of the quill shaft stiffness can be used to control these frequencies.

The second type of vibration has resonant frequencies almost entirely determined by engine inertias and crankshaft stiffness. With quill shaft drive and the flexible coupling driving

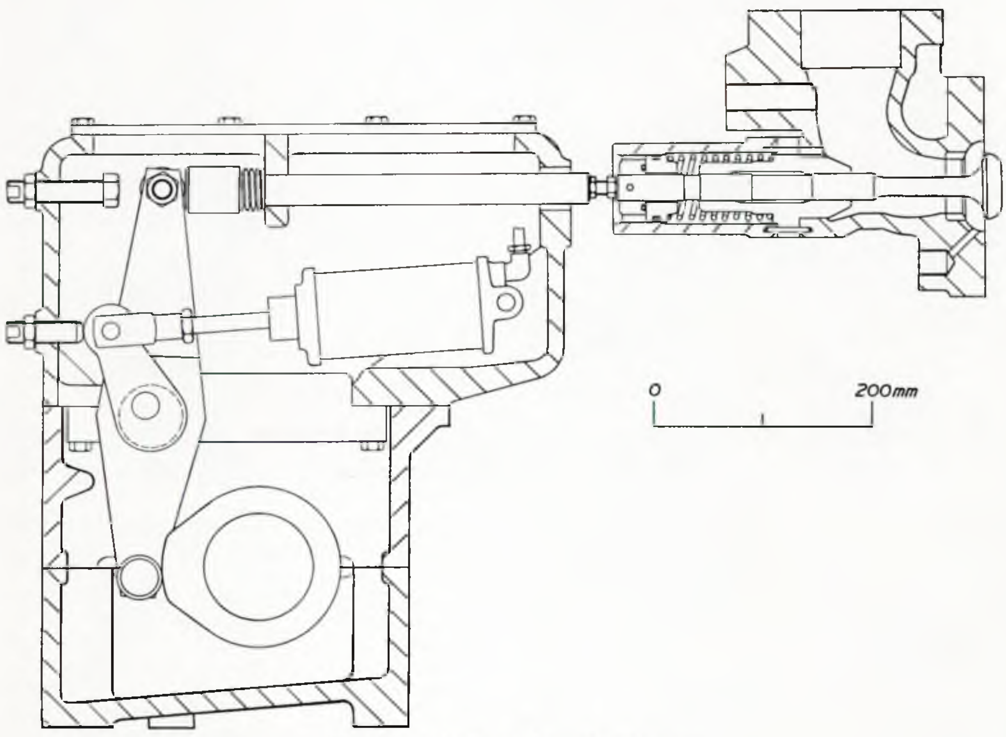


Fig. 25—Starting air mechanism

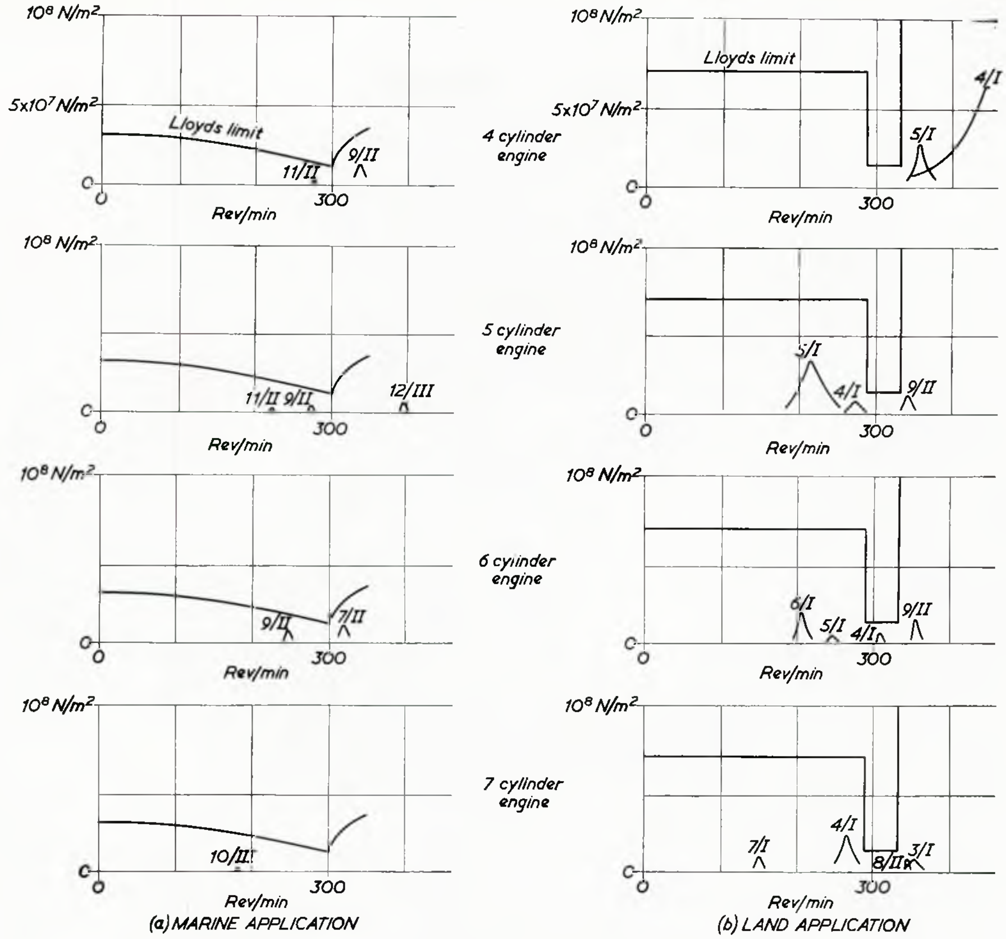


FIG. 26—*Torsional vibration stress*

the auxiliaries this type of vibration is characterised by having a node close to the middle of the crankshaft. The stresses due to vibration of this type are shown on Fig. 26(a) for engines with four to seven cylinders.

The normal crankshaft shown on the engine cross section has relatively heavy main webs, designed for ease of manufacture. In the 6-cylinder engines, crankwebs with a more complex profile will be used to place the seventh order resonance peak above the running speed.

For industrial applications, with a heavy alternator driven direct from the aft end of the crankshaft, the major order resonance for a 4-cylinder engine is well above the running speed. The corresponding major order resonances are well below the running speed for the 6- and 7-cylinder engines, but the 5-cylinder engine requires some flexibility in the drive to the alternator. This will normally be provided by a short length of shaft with a diameter rather smaller than that of the crankshaft, although other forms of flexible coupling may be used. Fig. 26(b) shows the crankshaft torsional vibration stresses for engines driving alternators.

In both marine and industrial arrangements the flexible coupling for the auxiliary drive also acts as a torsional vibration damper which is capable of a high degree of energy absorption because of the large inertia of the turning wheel and other driven parts.

A complete investigation has been carried out of the possible modes of vibration of the driven system up to and including the auxiliary compressor, showing possible resonant frequencies ranging from 2000 to 5000 vibrations per minute, but due to the decoupling effect of the drive from the crankshaft, the excitation couples encountered at these frequencies are low and comparatively moderate damping in the system is sufficient to avoid undesirable stresses.

Axial crankshaft vibrations are mainly of consequence in long stroke slow speed engines because of the flexibility of the main crankwebs. Although in this engine it is not possible to keep all axial resonant speeds above the running speed range, the very stiff short throw crankshaft ensures that both amplitude and stress are unimportant. This is so, even in industrial installations where the mass of the alternator brings the axial vibration node to the aft end of the crankshaft instead of in the centre.

In a previous section, the methods adopted to produce virtually perfect balance have been described and it is clear, therefore, that no problems are likely to arise from vibrations due to out-of-balance forces and couples. Crosshead guide thrust variation produces a twisting moment tending to oscillate the engine about both its longitudinal and vertical axes, as on any other reciprocating engine. In this case, however, the stiff structure and the broad base of the mounting feet, combined with the small engine height make the natural frequency of the engine structure very much higher than that of the excitation.

Medium speed geared engines tend to be noisier than slow speed engines because of the higher rate of pressure rise in the cylinders, the accelerated action of the fuel pumps, the rapid movement of the valve gear and the use of turbochargers running with high pressure ratio. The opposed piston design avoids valve gear noise but the maximum cylinder pressure and rate of pressure rise are higher than in direct drive engines. Noise from this source, however, is minimized by the very solid one-piece cylinder construction, by the rigid structure of the engine and by the enclosure of the top of the engine covering the cylinders, camshaft, exhaust pistons, and transverse beams. Although

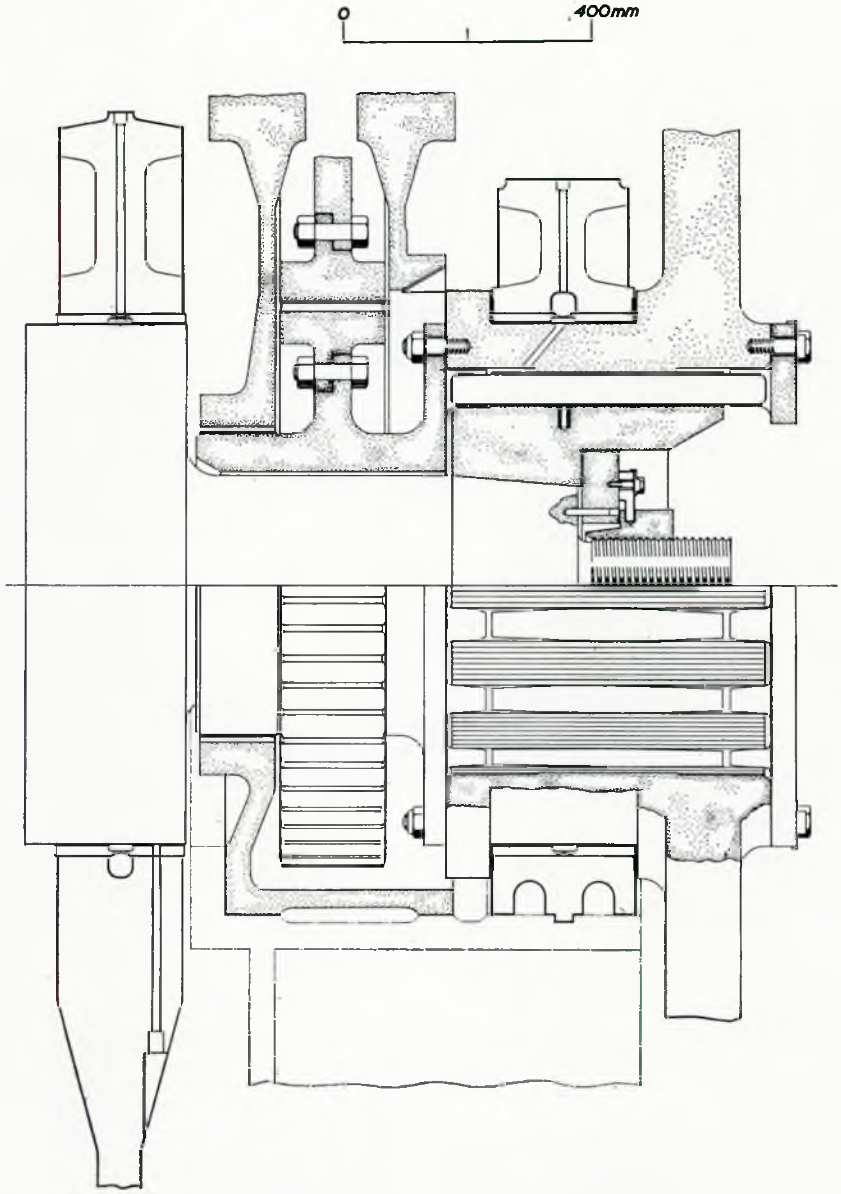


FIG. 27—*Auxiliary drive coupling and detuner*

essentially rigid except in the direction of permitted expansion, the mounting feet with resilient insert help to reduce the transmission of structure-borne noise.

MARINE INSTALLATIONS

With its four frame sizes, the Seahorse engine is able to cover powers from 10 000–70 000 hp per shaft with maximum power increments of 25 per cent when employing single, twin and quadruple input gearboxes.

For powers above 20 000 hp the splitting of the machinery into two or more units adds greatly to the overall reliability of the plant. Statistically, if the chance of a single engined ship being devoid of any propulsive power occurs ten times per year, the chance of such a happening with a twin engined ship, with engines of equal reliability, is reduced to one occurrence per year. Additionally, the opportunity for overhauling the main engine of a single engined ship can be severely limited by navigational and statutory requirements. There are many occasions when a master would accept a 50 per cent reduction in power availability, but could not contemplate a total loss of manoeuvreability. The ability to take advantage of enforced delays, of unpredictable duration, whilst waiting to go alongside or at berth, greatly favour the split engine arrangement and can be shown to be equivalent to somewhere between £2 and £5/hp in first cost.

By employing engines with unequal numbers of cylinders and triple input gearboxes it is possible to supply any power between 10 000 and 70 000 horse power per shaft in steps of 2500 hp, giving a total of 25 different powered installations for single screw ships.

The combination of engines of unequal size can be attractive when the vessel has a dual service rôle or when harbour duties, such as cargo pumping, demand powers comparable with that required for propulsion.

When a variable service power is advantageous, the use of c.p. propellers allows the full potential of the running engines to be utilized. The use of reduction gearing facilitates the installation of c.p. propellers, it being possible to arrange the pitch control mechanisms forward of the gearbox, thus avoiding the introduction of transverse penetrations of the intermediate shaft.

The engines are designed to be built to either hand and are equally suitable for continuous operations in either sense of

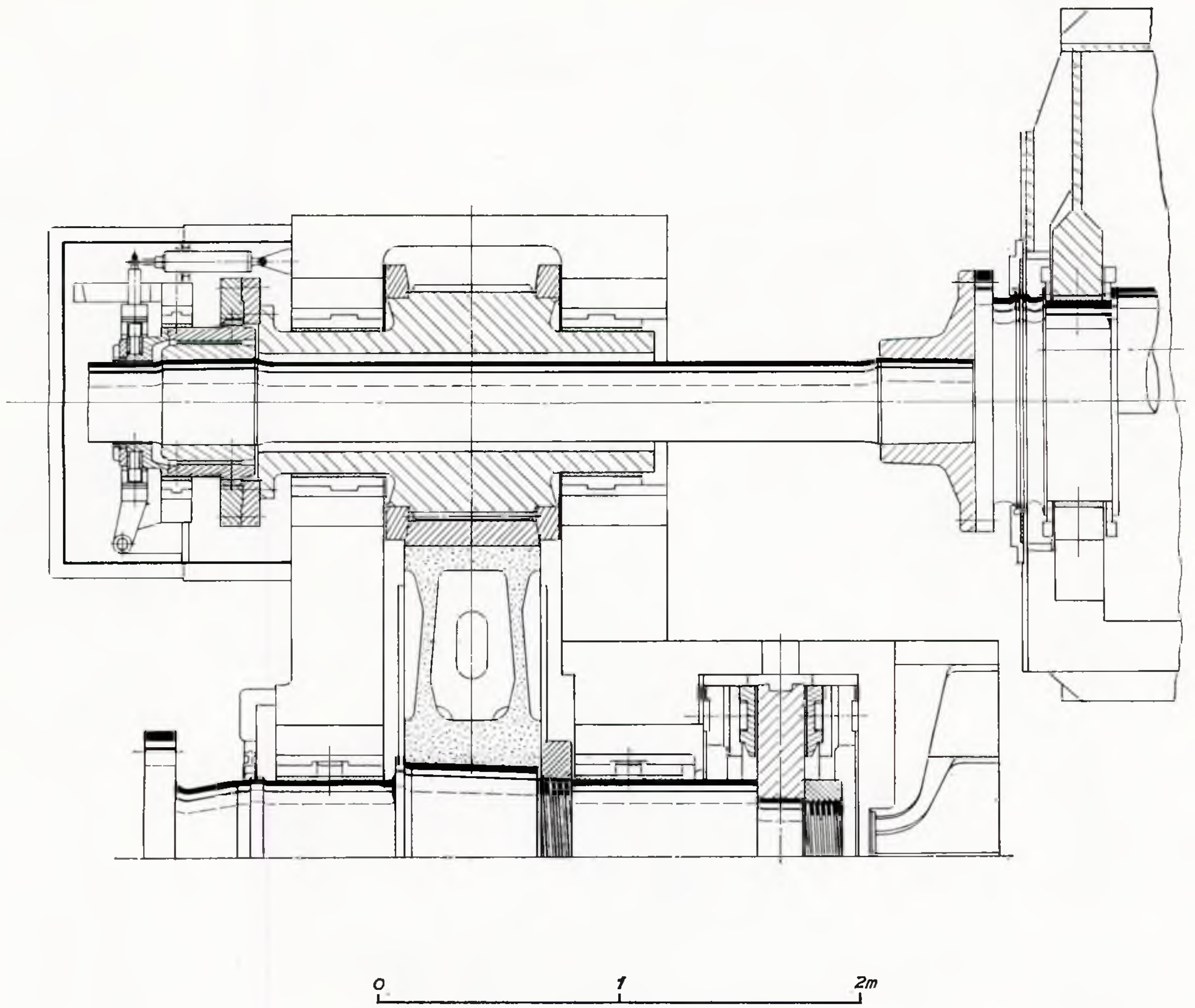


FIG. 28—*Twin engine marine gearing and clutch*

rotation. They are also suitable for delivering 100 per cent power from either end of the crankshaft, so generator or cargo pump drives can be taken from the free end of the engines. The handing of the engines and the "in line" construction allow them to be arranged in pairs, with a common overhauling space between, a centre line distance between the crankshafts of 3·6 m being fully adequate. This centre distance accords well with the optimum dimensions of an appropriate gearbox which is accordingly relatively simple and cheap.

With the likely reduction ratios being between two and five, it is possible to use a rationalized system of gearboxes for the entire range, without undue compromise.

The gearbox shown in Fig. 28 has double wall construction and is arranged with all journals in a single plane to facilitate construction and overhaul. The pinions are of such a size that they can be hollow bored to allow the use of a quillshaft between the engine and the remote side of the pinions. Such a quillshaft will, in almost all circumstances provide sufficient torsional flexibility to avoid gear tooth separation and unacceptable torsional stresses, but if necessary a soft coupling can be incorporated next to the engine. As well as providing torsional flexibility, these quillshafts allow considerable tolerance in the alignment of the engine and gearbox, making the provision of a very stiff double bottom construction unnecessary and setting less exacting requirements on the precise positioning of the engine. Such is the alignment margin in hand that it is possible to use a solid coupling at the engine end of the quillshaft. At the gearbox end, a fully barrelled fine tooth coupling is incorporated as part of a dog clutch mechanism. This dog clutch can be run in either the engaged or disengaged position and is, therefore suitable for harbour use with power take-offs on the remote side of the gearbox. This clutch is operated by a power cylinder and allows a pair of engines to be phased with a particular relationship, should this be found desirable from torsional considerations. Because of the crosshead and diaphragm construction of the engine, a straight run mineral oil is used for crankcase lubrication, since it is not liable to contamination. This type of oil is eminently suitable for use in the main reduction gear and so there is no necessity to use separate lubrication systems for main engines and gearboxes, thus saving the complication of additional storage facilities, drainage tanks, pumps, coolers, filters and control arrangements.

The space saving possible with the Seahorse engine as compared with a slow speed direct drive engine is shown in Fig. 29 which relates to a 20 000 hp installation. For an installation of this size, the saving in weight in respect of the engines and gearbox would be some 290 tons, and there would be an additional 60 tons saving in the steel used in the double bottom construction. Though it is appreciated that it is not always possible to take advantage of a saving in engine weight, there are, nevertheless, many types of vessel in which extra revenue can be obtained from the extra available deadweight. Calculated on a freight rate of 0·05p per ton mile, this saving in weight is equivalent to over £7 per horse power in initial cost.

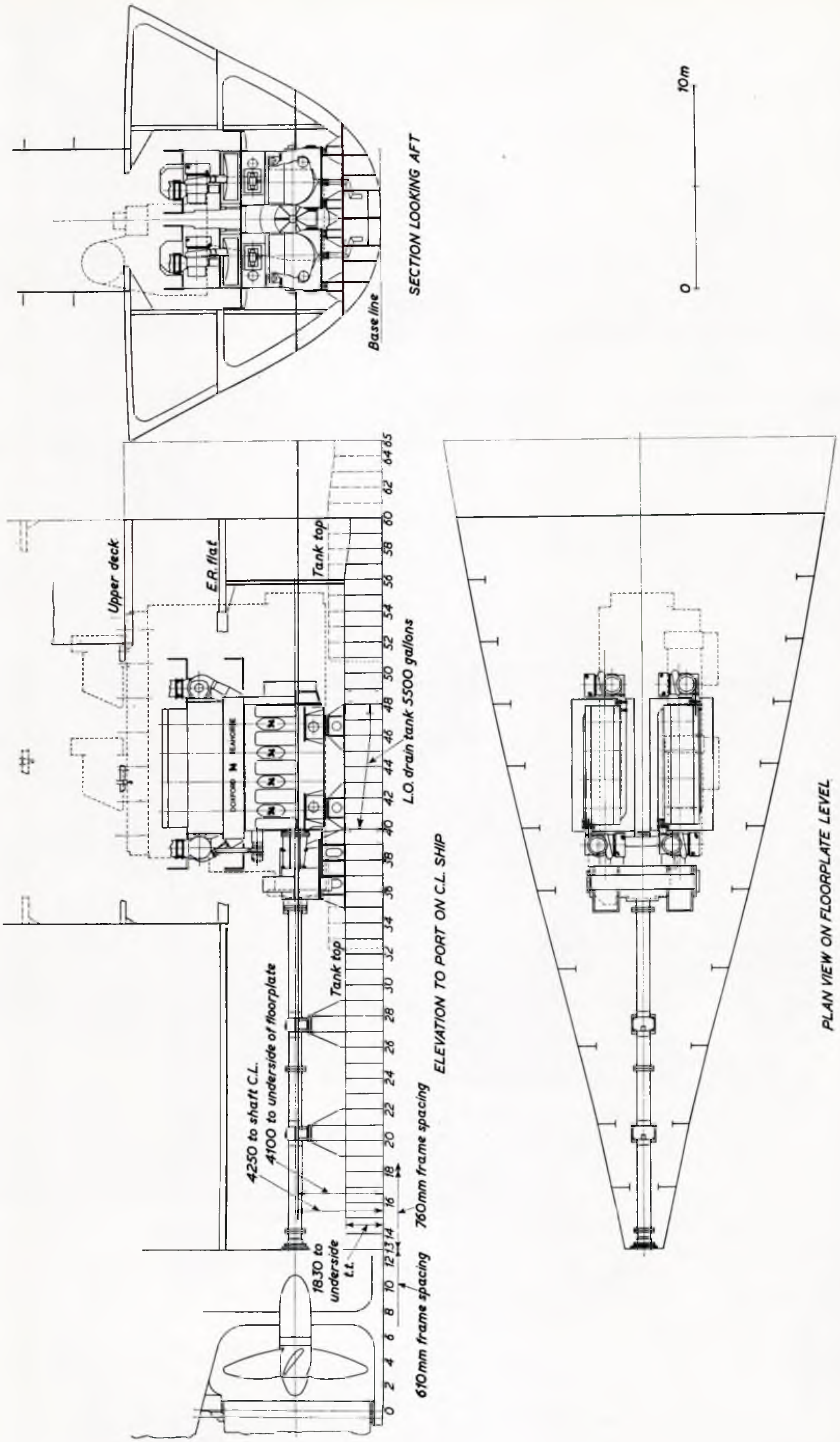


FIG. 29—*Twin engined marine installation*

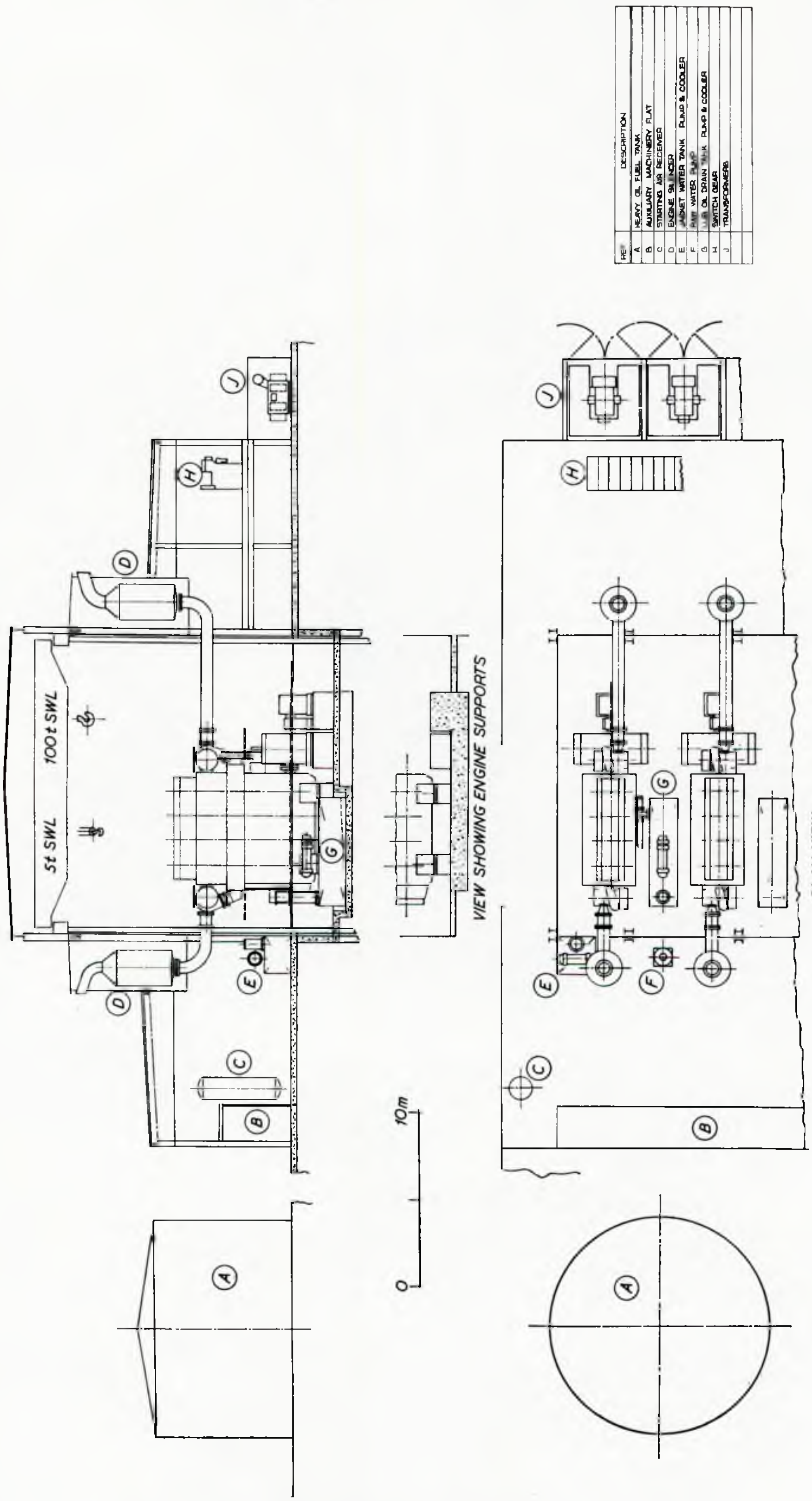


FIG. 30—*7 MW set generating station*

The saving in steelwork arising from the reduced double bottom stiffness permissible with point mounting of the engine makes for a very considerable economy. In addition, this support arrangement results in further savings compared with a conventional engine foundation, attributable directly to reductions in actual installation costs. The stools, which will be required to bring the engine crankshaft centre line up to the pinion centre lines, have limited surfaces requiring careful preparation. These surfaces can either be pre-machined before the stools themselves are welded to the tank top, in which case the chocks will require to be machined "to place". Alternatively, they may be machined *in situ* to a datum plane taken from the main reduction gearing, in which case the engine chocks can be pre-machined and the engine simply landed in its final position.

The use of stools allows adequate space for fitting lubricating oil drain pipes and greatly eases the underfloor arrangement by allowing the passage of transverse pipes other than at the extreme ends of the engine. These installation advantages are equivalent to at least another £1 per horse power over the slow speed direct drive installation with its critical alignment procedures and costly re-erection.

STATIONARY APPLICATIONS

Where there is a demand for power with a high utilization factor these is no better alternative to Diesel drive until the requirements reach the hundreds of megawatts when steam plant starts to come into its own. A Diesel engine generating set has an overall thermal efficiency of about 40 per cent and this high level of conversion is maintained down to quite low loads. The steam plant cannot obtain this efficiency, except when built in extremely large sizes and is always subject to considerable parasitic losses, when running at low load or on standby.

Due to the economics of conventional generating stations, very high distribution costs arise out of the necessity to concentrate the generating capacity. Furthermore, the whole system is vulnerable to breakdown of the cascade type, following a single station outage. Any developing area not yet covered by a grid system should give serious consideration to installing medium sized Diesel power stations which can generate electricity at a cost competitive with the largest conventional stations.

As well as district and industrial electric generation, high powered pumps and compressors can be gainfully of the direct Diesel driven type.

On a 100 per cent load factor the cost per unit of electric power generated in a station incorporating Seahorse engines is calculated to be 0·40 p. If the average load factor falls to 50 per cent this cost would increase to 0·52 p. These figures should be compared with the 25 per cent higher price of electricity bought from the national grid at 0·50p per unit at 100 per cent load factor and 0·62p per unit at 50 per cent load factor.

Fig. 30 shows a possible Diesel power station incorporating 7 MW Seahorse units. The close coupled single bearing alternators and light foundations should be noted. Light foundations are, of course, possible due to the very low levels of imbalance and the ease of meeting alignment requirements with this type of engine. It should also be noted that the station cranage is designed so that the engines can be installed in two lifts and the generators in a single lift, thus requiring a minimum of dismantling, avoiding the ever present possibility of damage to the parts in transit and during re-erection with relatively unskilled personnel. A five ton auxiliary hoist is provided for all normal maintenance.

TESTING

Development and endurance testing has been planned to extend over two years, to ensure that any possible weaknesses are found on the test bed rather than in service. To get the maximum value from this testing, a high degree of automation, including a data logging computer, has been installed on the test bed to enable the engine to be run under unmanned engine room conditions.

The cooling water and oil circuits for the engine, and the brake cooling water circuit are thermostatically controlled. As well as the engine being remotely controlled, the water brake dynamometer load is both registered and controlled pneumatically from the test office. The pneumatic pressure feeds an electrical signal into the computer by means of a transducer.

Fuel consumption measurement is accomplished by means of a weighing tank supported by three load cells. The weighing tank has electrical liquid level control which opens an inlet valve automatically at the lowest fuel level in the tank and closes it when it is filled to a safe level. At any position between these two the weight of the fuel in the tank is fed into the computer by an electrical signal.

At all times when the engine is running, connexions into the computer from pressure and temperature sensors on the engine act through a scanning device to provide a continuous monitoring system. During endurance running, all engine pressures and temperatures, as well as brake load, rev/min, and weight of fuel in the service tank, can be printed out on command by one of the two computer output typewriters.

For fuel consumption tests the system is arranged so that when the order has been given, by pressing a button, to carry out such a test, the computer automatically registers the starting time when the fuel level in the service tank has fallen sufficiently below the top level to give a satisfactory settling period. At this moment the computer starts to register brake load, rev/min, and temperatures and pressures, and continues to do so until the fuel has fallen to a preset level, which has a suitable margin above the low level at which refilling commences. The computer records the time taken for the test and the total weight of fuel consumed during that time and averages the brake load, rev/min, and important temperatures and pressures.

When the test is completed the computer calculates the average horse power during the test and divides the fuel consumed by the product of the horse power and test duration to give the specific fuel consumption. This information with the relevant average temperatures and pressures is then recorded automatically on a test sheet by the typewriter.

This rather elaborate arrangement is justified by the resulting saving in manpower during prolonged testing, by the ability to produce a finished test sheet within minutes of the completion of a consumption test and by the experience that will be gained in the use of sophisticated data logging under near-service conditions.

In addition to the normal readings fed into the computer, arrangements have been made for measuring liner and piston temperatures and the stress in major components, during running, by means of a multiplicity of thermocouples and strain gauges.

ACKNOWLEDGEMENTS

Neither a paper of this sort nor the design which it describes can be the work of two men. The authors want to thank all their very many colleagues for their unstinted help in the design and manufacture of the prototype and in the preparation of the paper.

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Appendices

APPENDIX I—DOXFORD SEAHORSE ENGINE PARTICULARS (See Fig. 31 below)

No. of Cylinders	4	5	6	7
Bhp	10 000	12 500	15 000	17 500
Weight tonnes	160	192	224	256
A (mm)	9060	11 460	12 860	14 260
B	2430	2930	2930	2930
C	105	350	700	870
D	7437	8837	10 237	11 637
E	5005	5100	5100	5100

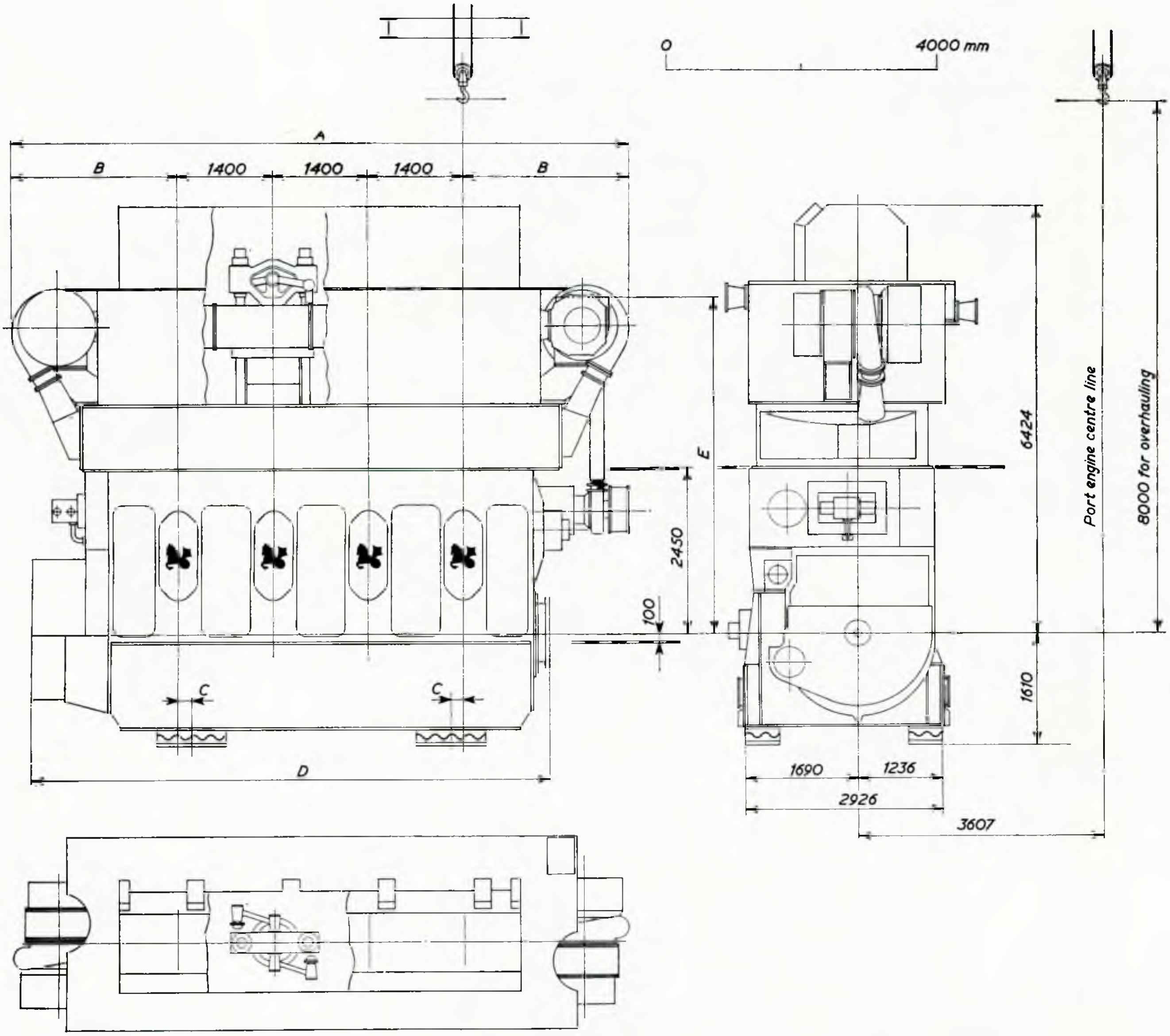


FIG. 31—*Dimensional sketch (see Appendix I above)*

Appendix II—Doxford seahorse engine particulars

	S.I.	*Units* *Metric Technical*	*Imperial*
No. of cylinders	— — — — — — —	4 to 7	— — — — — — —
Cylinder bore	580 mm	580 mm	22·83 in
Piston stroke:—			
lower piston	880 mm	880 mm	34·65 in
upper piston	420 mm	420 mm	16·54 in
combined	1·3 m	1300 mm	51·18 in
Maximum continuous power per cylinder	1·839 MW	2500 metric hp	2466 British hp
Maximum continuous speed	31·42 rad/s	300 rev/min	300 rev/min
Mean lower piston speed	8·8 m/s	8·8 m/s	1732 ft/min
Brake mean effective pressure	1·071 MN/m^2	10·92 kgf/cm^2	155·3 lbf/in^2
Maximum cylinder pressure	10·39 MN/m^2	106 kgf/cm^2	1508 lbf/in^2
Piston area	0·264 2 m^2	2642 cm^2	409·5 in^2
Swept volume per cylinder	0·343 5 m^3	343·5 litres	12·13 ft^3
Maximum continuous power per unit cylinder bore	3·170 MW/m	43·10 metric hp/cm	108·0 British hp/in
Maximum continuous power per unit cylinder cross sectional area	6·959 MW/m^2	0·946 2 metric hp/cm^2	6·021 British hp/in^2
Maximum continuous power per unit swept volume	5·353 MW/m^3	7·279 metric hp/litre	203·3 British hp/ft^3
Brake mean torque per cylinder at maximum continuous power	58·53 kN m	5·968 tf m	19·27 tonf ft
Maximum torque per cylinder at maximum continuous power	417·6 kN m	42·58 tf m	137·5 tonf ft
Cylinder centres	1·4 m	1400 mm	55·12 in
Crankshaft main journal diameter	830 mm	830 mm	32·68 in
Crankpin diameter (centre and side)	520 mm	520 mm	20·47 in
Centre connecting rod length between centres	1·52 m	1520 mm	59·84 in
Side connecting rod length between centres	1·3 m	1300 mm	51·18 in
Exhaust crank lead	139·6 mrad	8°	8°
Centre connecting rod/crank ratio	3·455	3·455	3·455
Side connecting rod/crank ratio	6·190	6·190	6·190
Total mass per unit power	21·8–19·9 g/W	16·0–14·6 kg/metric hp	35·8–32·7 lb/British hp
Weight of heavies assembly lifted during routine maintenance	51·3 kN	5·23 tf	5·15 tonf

Appendix III—s.i. units used in the paper—conversion factors (see also references 5 and 6)

S.I.		METRIC		BRITISH	
Mass					
1 kilogramme	(kg)	1 kilogramme	(kg)	2·205 pounds	(lb)
1 megagramme	(Mg)	1 tonne	(t)	0·9842 ton	(ton)
Force					
1 newton	(N)	0·1020 kilogramme-force	(kgf)	0·2248 pound-force	(lbf)
1 kilonewton	(kN)	0·1020 tonne-force	(tf)	0·1004 ton-force	(tonf)
Length					
1 micrometre	(μm)	1 micron	(μm)	0·03937 thousandth of an inch	
1 millimetre	(mm)	1 millimetre	(mm)	0·03937 inch	
1 metre	(m)	1 metre	(m)	3·281 feet	
Moment of Force					
1 kilonewton metre	(kN m)	0·1020 tonne-force metre	(tf m)	0·3293 ton-force foot	(tonf ft)
Pressure and Stress					
1 bar (= 10^5 N/m^2)	(bar)	1·020 kilogrammes-force per square centimetre	(kgf/cm^2)	14·50 pounds-force per square inch	(lbf/in^2)
1 meganewton per square metre	(MN/m^2)	0·1020 kilogramme-force per square millimetre	(kgf/mm^2)	0·06475 ton-force per square inch	(tonf/in^2)
				145·0 pounds-force per square inch	(lbf/in^2)
Power					
1 kilowatt	(kW)	1·360 cheval vapeur	(cv)	1·341 horse power	(hp)
		1·360 metric horse power	(hp)		
Density of heat flow rate (Specific heat flux)					
1 kilowatt per square metre	(kW/m^2)	0·02388 calories per square centimetre per second	(cal/cm^2 s)	316·9 British thermal units per square foot per hour	(BTU/ft^2 h)

Discussions

DISCUSSION AT THE NORTH EAST COAST INSTITUTION OF ENGINEERS AND SHIPBUILDERS ON MONDAY, 8TH NOVEMBER 1971

MR. G. YELLOWLEY, F.N.E.C.I.E.S. said that during his career in marine engine building, he had had experience of a variety of types of prime mover, both steam and Diesel, but in general, the results had been published after the engines had been tried and proved. He said that they had now had an excellent paper on an engine, or a horse, which had just been born and he did not know who was suffering the parental pains of birth.

The evolution of the engine had been possible due to the great advance in design technology and the authors were to be congratulated on their use of these new techniques and he hoped that the practical results would be successful and justify the predictions.

The market for medium speed Diesel engines had generally been for special purpose vessels, such as coasters, ferries, trawlers etc, but in recent years they had penetrated the slow speed Diesel engine market and were being fitted in deep sea cargo vessels, now at this meeting they had a traditional slow speed Diesel engine builder advocating a change-over to medium speed engines for cargo vessels, with the disadvantage, however that the engine described did not appear to be suitable for the special purpose vessels, due to its height.

One of the chief reasons put forward for this change in policy had been given in the introduction to the paper as the free choice of optimum speed for both prime mover and propeller; this, was something that had been known for many years, but it had not inhibited the development of the slow speed Diesel up to powers of 48 000 bhp per engine.

The calculations regarding the cost saving per horsepower of medium versus slow speed installations, based on optimum speed of the propeller were interesting, but unless the medium speed engines were manufactured in large numbers the prospective owner might have to pay a higher first cost per horsepower to benefit from lower fuel costs over the life of the vessel.

He said that the retention of the crosshead and diaphragm construction to facilitate the burning of residual fuel oil was a strong point in favour of the new engine, but was a penalty against it if the trunk piston medium speed engine could successfully burn this type of fuel and presumably the authors had convinced themselves that this was not possible. Although there were many reports of medium speed trunk piston engines burning residual fuel, the side effects of crankcase pollution etc were, perhaps, not given such wide publicity.

The other important change of policy was the adoption of the constant pressure system of turbocharging. The merits of this system had been given in a paper* read previously at this Institution in 1962, and the predictions made then appeared to be coming true, i.e., that at low degrees of turbocharging, the impulse system was clearly superior, whereas with high values of turbocharging the constant pressure system would be the best.

The only disadvantage of the system adopted for the Seahorse engine was that ten per cent of the total air required had to be supplied by an engine driven compressor which required about three per cent of the engine output. He noted that a combined turbocharger efficiency of 64 per cent was necessary for self-sustaining operation. As efficiencies of that order were at that time attainable on shop test, with constant pressure charged engines, it could well be that, in the foreseeable future, the engine driven compressor could be dispensed with and an auxiliary blower fitted for part-load operation. In view of the high compressor pressure ratios required, had the authors made any provision for condensation occurring after the air coolers?

The authors had stated that to attain a specific brake fuel consumption equal to the slow speed engine, a higher indicated thermal efficiency would have to be achieved, on account of the lower mechanical efficiency consequent upon the use of an engine driven compressor, gearbox and friction losses. The arguments put forward for achieving this improvement, were not convincing: little improvement had been made in specific fuel consumption per horsepower hour over the years and improvements in specific fuel consumption per brake horsepower hour had been due to improvements in mechanical efficiency resulting from introduction of higher degrees of turbocharging and increased b.m.e.p.

Quick combustion and an increased rate of pressure rise, resulting in a maximum combustion pressure of 1500 lb/in^2, was one item mentioned to achieve a better consumption per horsepower hour, this brought Mr. Yellowley to the question of cylinder liner and piston head design.

The bore cooling of the cylinder liner in way of the combustion space followed well-proven design, but it was not clear from Fig. 8 how the problem of accommodating four fuel valves, starting air and relief valve had been overcome.

The upper piston being water cooled should present no problems, but the lower piston being oil cooled would be subject to the highest thermal stresses. The diaphragm cooling of the piston crown, however, was a well-proven feature of the Doxford engine.

Regarding the oil cooling of the lower piston, Mr. Yellowley asked whether he was correct in assuming that there was a common supply to the crosshead shoe and that the oil flow then divided to supply cooling oil to the piston and lubricating oil to the top end bearing and thence to the bottom end bearing. If this was the arrangement, then careful means would have to be provided to apportion the flows correctly to ensure an adequate supply of cooling oil to the piston.

No mention had been made in the paper of piston rings and it was noted from Fig. 11 that only four piston rings were fitted; he wondered whether any special type of ring was to be used, for with maximum pressures of 1500 lb/in^2 in a 580 mm bore cylinder, careful attention to this important item would be essential.

Mr. Yellowley noted that all the important bearings in the engine had thin steel shells lined with white metal, had any cracking of white metal been experienced in the Doxford slow speed engine where this type of bearing had been used? One advantage of the thin shell bearing was, the paper said, that the bond between white metal and steel was consistently good, but with improved processes of white metalling the traditional type of bearing, equal results could be obtained, especially if the keying dovetail grooves were eliminated as in the thin shell bearings.

The load reversal on the top end bearing during each cycle was an excellent feature of the design, and he was glad to have the polar load diagrams in Fig. 15 to illustrate this.

With regard to development and testing, the authors were again using the best of modern techniques and, having been brought up in the hard school of testing engines by laboriously taking readings manually, he really appreciated the system of testing being introduced on the prototype engine which should lead to greater accuracy and better control.

MR. J. MORRISON said that concerning the saving of £1·50 for horsepower based on power savings due to reduced propeller speed, a slow speed engine of 29 400 hp was set against 27 500 hp from a medium speed engine, presumably 11 cylinders of the Seahorse, If the required power from the slow speed engine had been 30 000 or 31 000 hp and the same percentage savings were achievable with the slow speed propeller on the medium speed engine, giving 28 050 and 28 985 hp respectively, would it not be true to say that 12 cylinders had been required with no saving in the first case and only an effective saving of 1000 hp in the second? In other words, because the minimum step in power for the Seahorse was 2500, there was not always a genuine direct saving in first cost, although if the thermal efficiency did not drop with slightly reduced power there would presumably be saving in fuel, additional to that shown as a direct fuel saving for the lower power at rated output.

* Yellowley, G. and Collin, L. Th. 1961–62. "The Götaverken Diesel Engine"—Part II, *Trans. N.E.C.I.E.S.*, Vol. 78, p. 360.

Out of these same figures the price of a Diesel these days was £23 per horsepower and since the Seahorse was one of the engines under comparison, was this the order of price of that engine?

Could the authors explain what factors were included in the figure for the unit costs of electrical power given in the paper? That is, did it include first cost, fuel, oil, maintenance etc? It would be interesting to know the factors included and the method of calculation of all costs and economic comparisons.

He noticed a reference to reliability in the paper, i.e., the potential improvement in reliability when two engines as opposed to one were used. In the future, as human supervision in engine rooms became less, this factor would be most important. Marine engines, in spite of what was said about them at times, had always been essentially reliable pieces of machinery, but with a new engine such as this with a higher power/weight ratio than the slow speed engine, how much attention had been given to reliability of each component and the consequent overall reliability of any one engine comprising these many components? He realized that service experience and statistics therefrom would be necessary to establish figures for M.T.B.F. and to enable this to be increased by improvement of vulnerable components, but felt that in future years all customers would be asking for specific figures and guarantees that they would be met.

He asked whether in addition to designing the control system for the test bed, control systems for service were being considered and whether the engine itself was being designed for remote and automatic control. For example were valves being chosen for their control characteristics, pipes sized for correct pressure drops and facilities being built in to enable the correct control parameters to be sensed and measured accurately etc. He felt, from the tone of the paper, that they were, but probably the authors could elaborate.

He was not overkeen on the dependence on very high cooling oil flows in the lower piston to prevent carbonization—might not the high temperatures lead to carbonization here anyway, with subsequent potential problems in the piston crown and with the lubricating oil?

Nevertheless there were many admirable features about the design, the engine and the proposals for installations in ships, a few being: standardization of sizes and components to facilitate production, maintenance and the spares situation; the attention to design with respect to thermal loading to ensure reliability from this point of view; the full length bottom bearing in the crosshead to reduce the problems which had been experienced with this bearing in the past in some slow speed engines; the diaphragm incorporated in a medium speed engine to alleviate lubrication problems; the novel method of four point mounting to keep bending moments low, and the chocks to accommodate expansion, both of which made the installation less dependent on (but not wholly independent of) the structure on which the engine was to sit.

Professor G. H. Chambers, D.S.C. Vice-President N.E.C.I.E.S., asked, what were the ambient air temperature and cooling water temperatures for which the engine was designed? If these were for the Lloyd's seagoing requirements, this would presumably give the engine some margin on its land application, where British Standards presumably applied.

The paper had referred to the mechanically driven blower taking care of the situation where one or both of the exhaust blowers were out of action. What would the situation be if the reverse obtained, how would the engine start and behave generally if the mechanical blower was out of action?

Mr. A. Rose, B.Sc., M.N.E.C.I.E.S., said that the authors had mentioned the advantages of using a constant pressure fuel oil pump and timing valves, pointing out that one of the major advantages had been lost in the case of this engine. It would seem, therefore, that the jerk pumps would have the advantages of overall simplicity and reliability. Now that tests were being carried out, using both systems had they found any reasons to change their original views?

Fig. 16 gave the film thickness of the main and bottom end bearings, apparently based upon the polar load diagrams for an infinitely stiff structure shown in Fig. 21; comparing the polar load diagrams for the stiff structure and the more realistic structure, it appeared that film thickness for Nos. 2 and 3 main bearings would, if anything, increase slightly, whereas No. 1 main bearing would decrease. An estimate from Figs. 16 and 21 would suggest that the film would decrease from 0·001 in to 0·0008 in. However, in the case of reciprocating bearings, such an estimate, without the backing of a full analysis, could be misleading and he would ask the authors if they had film thickness figures available for the flexible case.

Regarding the centre top end bearing, one would expect the load reversal combined with the higher speed to ensure that this bearing was relatively trouble-free.

Fatigue of white metal often constituted a problem in reciprocating machinery. Had the authors introduced any special materials or techniques to overcome this? In this case he was particularly thinking of the use of a lead overlay to the white metal which, it had been claimed, had considerably reduced fatigue in other marine Diesels. The lead was deposited by electro-chemical means and seemed to protect the bearings particularly in the running-in stages.

Mr. R. B. Conn, A.M.N.E.C.I.E.S. said that a previous speaker had alluded to some consequences of the unusual size of the engine. This could also lead to difficulties for the ship designer from the vibration point of view. Because of the quite large steps in power it would sometimes be necessary to choose a five-cylinder engine, whose secondary unbalance, though much smaller than in other types of engine, was by no means negligible. The difficulties arose because the secondary couple was at a high frequency which corresponded, in a ship for which a 12 500 hp engine was appropriate, to high modes of hull vibration whose frequencies could not be estimated with confidence. In addition, resonant deckhouse vibration which could be of objectionable amplitude with fairly small excitation, occurred typically in this frequency region and was difficult to predict accurately.

Medium-speed engines were notoriously noisy and he would expect the total noise level one metre from a basic engine of this type to exceed 110 dB(A). This particular engine was, however, well suited to noise-reducing measures, since the rigidity of the structure did not depend on the enclosing panels. Did the authors intend making use of heavily damped, resiliently supported panels to reduce the radiation of noise from the external surfaces? The resilient mounting feet should reduce the transmission of high frequency noise to the ship's structure, but would not have a significant effect on noise in the engine room. He pointed out that the noise regulations with which shipbuilders were most often faced, issued by the See-Berufsgenossenschaft of Hamburg, specified that 'in no circumstances must personnel be exposed to noise the total level of which exceeds 110 dB(A) . . .' unless ear-defenders were used.

Mr. P. Manson, said that over the past 10 to 15 years there had been a very rapid increase in the power rating of marine Diesel engines, and that piston crowns, and cylinder liners and covers, which were subjected to both mechanical and thermal stresses due to combustion gas pressure and temperature during operation were the parts most liable to damage, among engine components. With increased outputs and low grades of fuel oil, conditions were becoming more and more severe against the parts around the combustion chamber and, in this respect, he referred to Fig. 11 which showed the maximum surface temperature of the liner as 240°C (464°F)—could the authors indicate the procedure during manufacture of the liners in respect of heat treatment, if any was applied? Some liner makers relied on the so-called seasoning method to remove residual stresses, i.e., the liners were left lying in the shop for a certain period of time. This method was applied in Japan up to a number of years ago until the power ratings had begun to increase every five years or so, since when, running conditions had become more severe.

The practice in Japan was to machine the liners to within 2 mil of the finished dimension, then to place the liners in an annealing furnace heating up to 550°C (1022°F) for four or five hours, and furnace cooling them. In view of the configuration of the liner for the Seahorse engine, this procedure if it had not already been adopted, was worthy of careful consideration.

With regard to piston crowns, particularly with oil cooled pistons, he referred to a statement made in a paper by Dr. Isumi at the Cimac Conference which was held in Sweden in 1971, giving the results of tests carried out on a 520 mm bore UET engine developing 5200 hp with a b.m.e.p. of 10·5 kg/cm², the pistons being oil cooled by the so-called "shaker" type cooling method. The temperatures of the piston crown, as recorded during full power tests, were very similar to those shown in Fig. 11, however, the material of the piston crown in the engine was an alloy steel having 0·6 to 0·9 per cent chromium and 0·3 to 0·5 per cent molybdenum. Further study into the material to be used for the lower piston would be well worth while. Could the authors indicate provisionally the recommended number of running hours before cleaning the oil-cooled surface of the crowns to remove carbon build up. If the material used for the crowns were chrome molybdenum steel, could the period of time before cleaning be extended?

He was pleased to note that the diaphragm had been retained which brought to mind an occasion about ten years ago, when the owner of three ships fitted with Sun Doxford engines built in 1939 had asked advice on whether to convert for running on heavy fuel. The owner had experienced little trouble with liners and pistons, and from personal observation the crankcase and lubricating oil had been considered clean. The idea of conversion was dropped and two of the ships were running to this day. Reference had been made to lighter engine foundation etc; he felt the authors were tempting the devil, and consideration for the unit as a whole should be maintained, especially where one was involved with couplings between engine and gearboxes.

MR. J. W. ECKHARD, Member of Council, N.E.C.I.E.S., said that although the present Doxford design of engine was placed in a low position in the published league of horsepower on order, it was worthy of note that, of the ships built by the Tyne Shipbuilding Consortium, the Doxford design was equal top in numbers of engines installed or to be installed since the formation of the consortium and that one-third of the ships on order which were to be delivered after the end of 1971, were powered by Doxford engines.

Tyne Shipbuilding had, for various reasons, supported the companies developing the Seahorse engine. From the shipbuilder's point of view, it was necessary that several engine builders competed for main engines, and it was also necessary that shipbuilders should be able to obtain the makes of engine which their customers preferred as first choice. The Seahorse engine would, in his opinion, undoubtedly be of interest once it had proved itself.

Apart from approximately two years of test bed trials, how did the companies intend to prove the engine? Would this be done by persuading an owner and builder to buy and fit a prototype engine, or did they propose to make the first engine a free issue until they were proved to be successful? Or was it intended to extend the normal test bed trials to trials on a floating test bed owned by the engine builders?

Fig. 27 showed normal engine casings above the engine—had any installations been drawn out where casings were eliminated? It had been foreseen that these engines would be attractive in multi-deck ships where deck area was of the utmost importance as cargo space, particularly in the large Ro-Ro cargo ships with stern loading ramps.

It would be interesting to have some comparative information on what "shock" standard this engine would withstand compared with the slow running competitors.

DISCUSSION AT THE INSTITUTE OF MARINE ENGINEERS ON TUESDAY, 9TH NOVEMBER 1971

MR. C. C. J. FRENCH, M.I.Mar.E., said that for some yeare there had been predictions about the great inroads which were going to be made into the marine propulsion engine field by ths medium speed Diesel engine. Low speed engines, however, still retained the lion's share of the market. As had been often pointed out, one of the advantages of the medium speed engine had been, rather paradoxically, that as it ran faster, it could be more easily geared to give a low propeller speed, with a resultant gain in efficiency. Up to the present time the author's company had had to look to their low speed engine line, but now there was promise of a medium speed Doxford engine. Mr. French believed it to be not just another medium speed engine which would have serious trouble in breaking into the shipping market. It offered, as had been seen, something like two to two and a half times as much power per cylinder as existing medium speed engines and had the same advantages of small numbers of components as did the slow speed engine. For that reason this was an important paper describing an important engine which, if it was as successful technically as it promised to be, had a considerable future. These were early days since the prototype was only just running-in, although clearly much theoretical work had already gone into design.

His own company had been involved in assisting in the initial design and in the thermal loading analyses which guided the final form of the piston crown and cylinder liner. As a part of the experimental programme, the isotherms upon which the stressing was based were to be checked. The installation of the instrumentation of one complete line had been completed; that is, the two

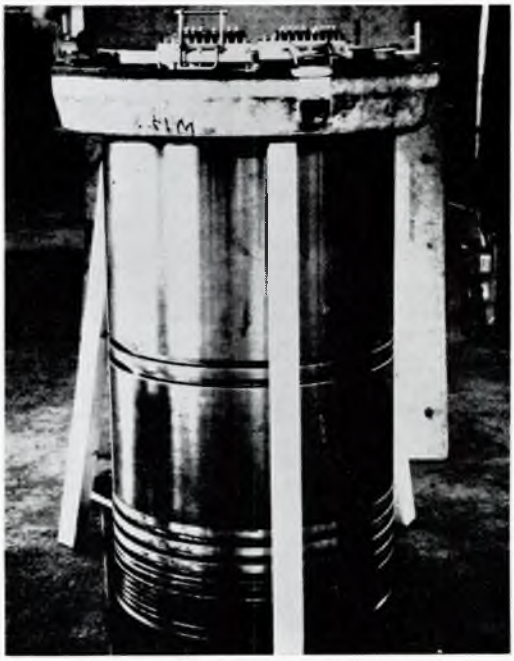


FIG. 32—*Exhaust piston with seal ring carrying the contacts*

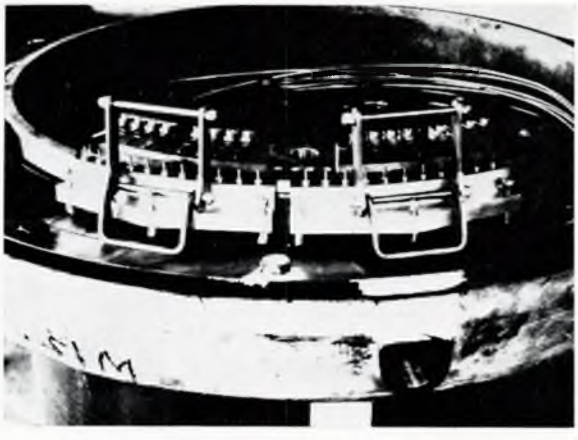


FIG. 33—*Springs mounted on piston skirt*

pistons, together with the cylinder liner. Measurements were to be taken as a part of the testing of the prototype engine. Four fixed and six traversing thermocouples had been fitted.

Figs. 32 and 33 showed the piston with the contact gear ready for installation in the engine.

Mr. French asked the authors what their expectation of liner life was. The single-piston, medium speed four cycle engines exhibited much more uniform and lower wear figures than did the lower speed two cycle engines, due to the better lubrication conditions and to the use of a higher oil consumption. Presumably the Seahorse would exhibit figures closer to the conventional low speed engine than the medium speed engine, but he would be interested to hear the authors' comments.

Mr. Butler's remarks on engine noise were interesting. Mr. French's company had always felt that the side rod opposed piston engine should be very good from the structural noise point of view, because there was so little stressed structure to act as a loud speaker to transmit noise.

Mr. C. Anderson said that from the figures quoted in the paper it could be concluded that the selling price of the Seashore would be £23·10/bhp. It would be interesting to know if this was the case and whether it included gears, couplings and thrust bearing?

Later in the paper the following phrase occurred: "crankshaft revolutions restricted to artificially low levels" for direct drive engines. The authors would surely be aware that uniflow engines at least had stroke/bore ratios of more than 2:1 and thus had been able to give an economically advantageous high piston velocity with relatively low revolutions.

Mr. Anderson understood that the figures under the heading "Combustion Efficiency and Fuel Consumption" regarding air amounts, fuel consumption and thermal loading were based on theoretical considerations and not on actual tests.

Regarding fuel consumption, he wondered whether this referred to the gear output shaft or to the engine output shaft and what kind of fuel was referred to and what was its calorific value?

Bearing in mind that an increase in maximum pressure generally reduced any possible gain in thermal efficiency by increasing friction, he asked for details of the expected mechanical efficiency upon which the consumption figure was based.

He understood that the calculations of wall temperatures were based on the measurements from the 76J6 engine. By what means had the authors assured themselves that there would be no material difference in heat transmission to the walls due to higher revolutions and piston speed which might have an appreciable influence on burning conditions and radiation from the burning charge?

Furthermore, he supposed that the temperatures in the upper piston ring groove, which were rather high, had given satisfactory results in the previous engines.

He congratulated the authors on their designs of the cooling arrangements for the pistons, both lower and upper, which seemed to be simple and reliable, and almost envied them for their design of the side connecting rods. Regarding the fixing of the crosshead journal to the shoe he wondered how they had covered themselves against pitting corrosion.

The design of the engine support was unique and would probably, correctly dimensioned, give some reduction in noise transmission. He hoped that it would give a satisfactory lifetime for the rubber bonding without any side effects due to vibration.

One of his company's British patents that expired a long time ago described a crankshaft design exactly like that of the Seahorse. But they had never dared to use it due to the eccentricity that would result from a slip in the shrinking of the side rod journals. Had Doxfords taken special precautions to make such slipping an unlikely occurrence?

It was well known that an engine turbocharged by the constant pressure system needed supplementary air, especially at part loads.

At full load for the Seahorse, the engine driven auxiliary blower was stated to give ten per cent of the total air. The turbochargers could thus be matched to give 90 per cent of the total air at full scavenging air pressure, with reasonable margin for surge.

At part loads, according to the propeller law, the drop in speed of the engine driven compressor would be relatively smaller than the drop in speed of the turboblowers. This meant that the engine driven compressor would deliver relatively more than ten per cent of the total air amount at part loads and thus "push" the turboblowers towards the surge line. To avoid surging, the turboblowers had to be matched less efficient also at full load.

Better matching of the turboblowers could be possible by closing the control valve in the discharge pipe from the auxiliary blower, but as long as the auxiliary blower contributed more than ten per cent of the air at low load, the matching of the turbochargers at full load, would be adversely affected.

As the requirement for a constant pressure turbocharging system would be to allow for increased supplementary scavenging air to the engine at low load, a compromise had to be found. What was the estimated rate of supplementary air delivery as a percentage of total air delivery at different loads?

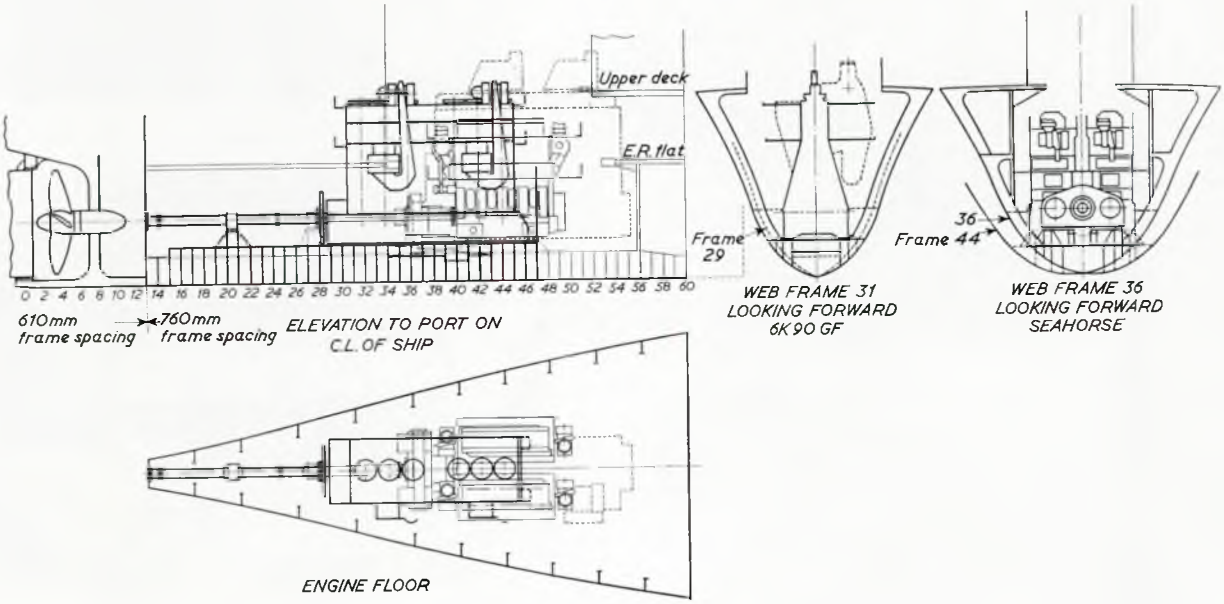


Fig. 34

Fig. 34 showed an arrangement with a 6K90GF engine direct coupled to the propeller, with 20 500 (21 200) bhp (m.c.r.) at 114 (118) rev/min, corresponding to the Seahorse solution, considering the difference in propulsion revolutions and gear revolutions. The 6K90GF was shown with "thick" lines and required about three frame spacings less than the geared Seahorse for the very slender ship Doxfords had chosen.

For ships with more common aft hull form the 6K90GF could be moved about three metres further aft.

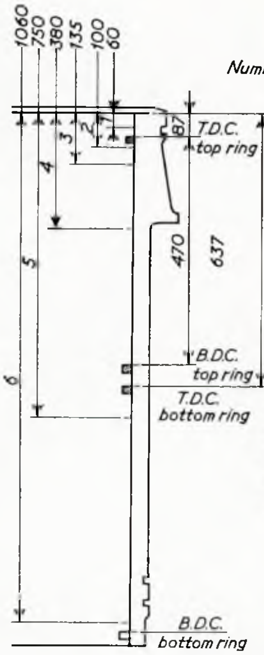
MR. J. H. WESSELO said that he was particularly interested in opposed piston engines and subscribed to many of their advantages. Because of this interest his company had carried out an opposed piston engine study in the same power range as the Seahorse; this study would form the background for some later remarks.

First, however, he wished to comment on the opinions expressed by the authors on the ability of four-stroke engines to burn heavy fuel. He hoped that the success of the Seahorse engine did not depend on these statements since they were contradictory to certain facts in his company's experience. In support of this he gave some results obtained with the TM 410 medium speed engine, designed for service with heavy fuel.

The first two engines delivered were inspected after 1000 hours running, mainly on heavy fuel and the cylinder wear averaged less than 0·005 mm/1000h (0·0002 in/1000h)—see Fig. 35. At the same inspection, the routine lubricating oil analyses report showed that the oil, which had never been changed, was still fit for use. This confirmed earlier experience that a diaphragm was not necessary for maintaining good crankcase oil condition.

The argument about the effect of size was fundamentally true but had no effect in this case because:

a) fuel spray impingement probably occurred and, if so, probably on the piston; however, due to the low piston surface temperatures (see Fig. 36) no harm came to the piston;
b) the effect of fuel viscosity on combustion delay was so small on this engine type that, at 550 rev/min, the cylinder pressure diagram (see Fig. 37) was similar with all grades of fuel, from Marine Diesel to 3500s Redwood I at 100°F.



Measurements in 0·01mm above 410mm

Cyl.		Measuring point 1	2	3	4	5	6
1	L	+5	+13	+20	+1	0	+1
	D	+5	+ 9	+20	+8	+2	+3
2	L	+3	+ 8	+10	+2	+1	+2
	D	+5	+ 7	+ 6	+4	+2	+2
3	L	+1	+ 2	+ 2	0	−2	0
	D	+1	+15	+ 7	+1	−1	−1
4	L	+1	+ 5	+ 6	0	−1	+1
	D	+2	+ 4	+ 4	+1	0	+2
5	L	−1	+ 4	+ 2	+2	0	0
	D	+2	+ 4	+ 4	0	−2	−1
6	L	+2	+10	+ 7	+2	+2	+3
	D	+5	+12	+ 9	+4	+3	+3

New $410^{+0.063}_{+0}$

Wear, mm/1000hour, of point 2

	Cylinder 1	2	3	4	5	6
Point 2	0·0035	0·0021	0·0044	0·00[illegible]8	0·0021	0·0044

FIG. 35—*Cylinder liner wear—6TM410 engine*

The fact that burnt exhaust valves were unknown on the four-stroke engines referred to here was something he had discussed at an earlier meeting of the Institute.* However, the present authors had not specially stressed this point.

Users of the engines had observed no difference, in maintenance costs and work-load, between the first 3000 hours on marine Diesel fuel and the later 14 000 hours on heavy fuel.

* Wesselo, J. H. 1972. Contribution to discussion on "Operational Experience with Medium Speed Diesel Engines". *Trans.I.Mar.E.*, Vol. 84, p. 59.

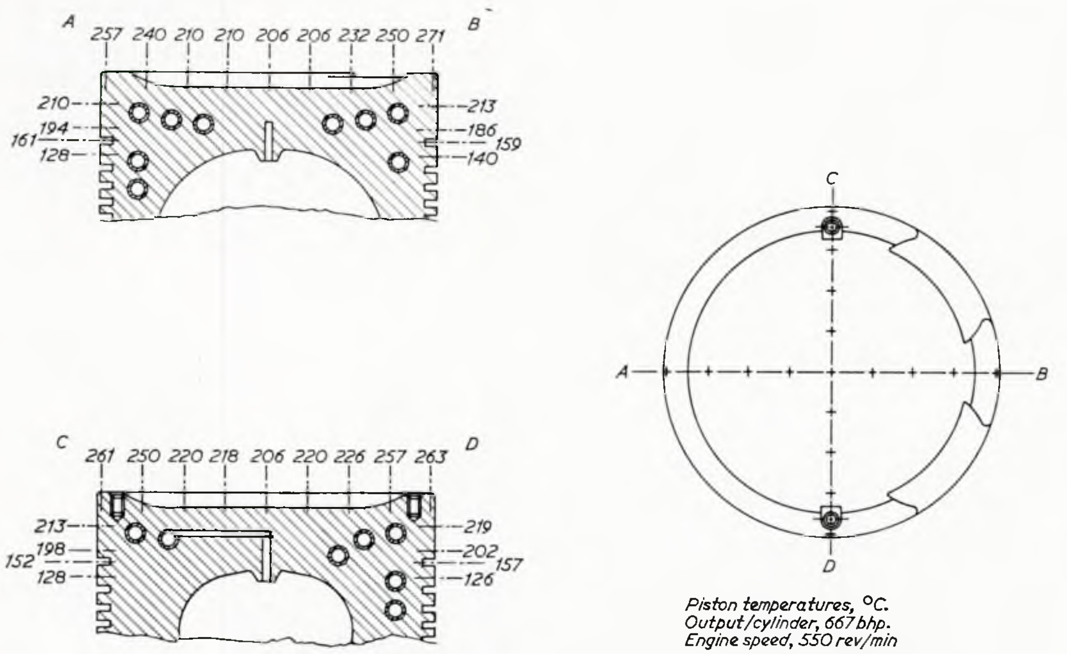


FIG. 36

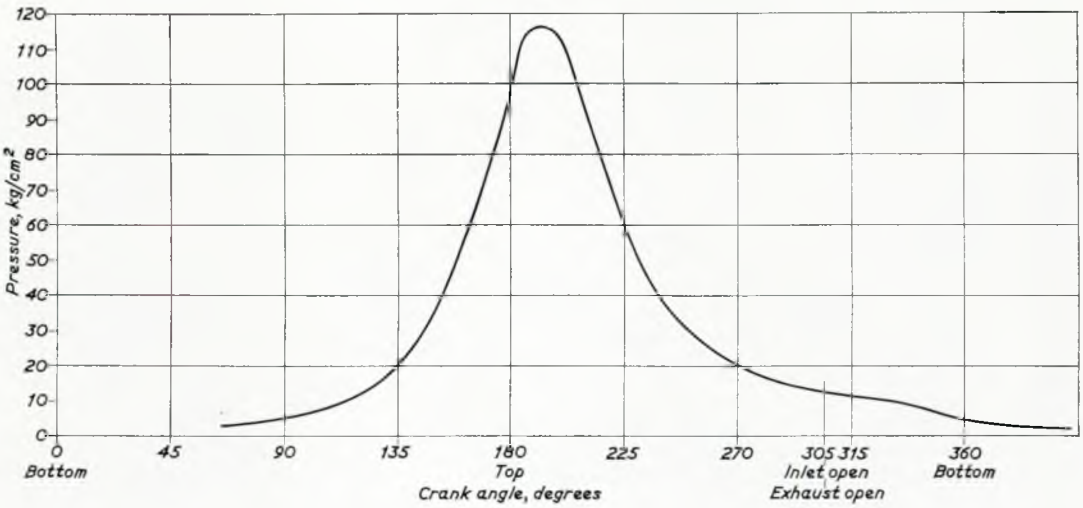


FIG. 37

As a result of this kind of information, no limits were specified for fuel viscosity.

In view of the foregoing, it was not surprising that the design study, made by his company, was for an opposed-piston engine without diaphragms. This necessitated two crankshafts, but saved one diaphragm, three crossheads and a number of rods and jigs for each cylinder. It also allowed integrating the camshaft drive and reduction gear.

One interesting result of the study, on an engine of practically the same cylinder bore as the Seahorse, but with higher speed and b.m.e.p. ratings, was that its cost would be roughly ten per cent less than that of modern four-stroke engines. In view of the larger, more complicated Seahorse design, he thought, therefore, that it would be difficult to achieve a cost level competitive with medium speed engines on the market.

Other features in his company's design showed a large degree of agreement with the author's ideas, e.g.

1) large port area—his company had designed even larger ports, considering the similar "gas-piston speed" (higher speed, shorter stroke); perhaps this was why their computer calculations showed that a turbocharger efficiency of 60 per cent would result on a specific air consumption of 7 kg/bhph without support from the auxiliary blower;
2) they would, like the authors, use the constant pressure system with an auxiliary blower, at least for part load;
3) they also thought that a longer exhaust stroke would have a salutary effect, but obtained this automatically by the symmetric design;
4) the thermal load might, by his company's calculations, be similar or somewhat higher—perhaps due to a stronger anticipated swirl—and they thought, like the authors, that the piston should be cooler than calculated, particularly on the oil side; as the piston was the most critical component and so probably determined the output achieved with the engine, it was felt that a different cooling system, for instance with liquid metal as an intermediate medium, might be necessary;
5) Mr. Wesselo would be content with a well cooled, one-piece, cast iron liner as applied to a free piston design some 20 years previously, thus avoiding difficult construction with clamps etc., and the troubles involved.

Was the 40°C lowest liner temperature mentioned in the paper achieved by an even lower cooling water temperature and did the authors not fear corrosion at such a low temperature?

MR. L. SINCLAIR, M.I.Mar.E., complimented the authors on Fig. 2. A diagram which gave ship and propeller dimensions, optimum rev/min for direct and geared drives, as well as fuel first cost assessment and q.p.c. must surely be unique. However, he had carefully considered the diagram and could not quarrel with any of the information so deviously supplied, except perhaps on one detail, and that was in relation to propeller size. Facilities now existed to make propellers up to 80 tons in weight; this would mean that there was no need to restrict the propeller size, and with advantage the revolutions could be reduced and propeller diameters two-thirds of the draught used throughout the diagram, thus providing a further propulsive gain in the case of the geared installation at higher powers.

The trend towards lower revolutions and larger propellers, where this was permitted was continuing and as an indication of this. Fig. 38 showed the number of propellers over 50 tons in weight made in a U.K. propeller works during the period 1967 to date.

There seemed no immediate prospect of this trend being halted, and at this moment serious projects were in hand utilizing propellers up to 78 tons in weight. The use of lower revolutions on large tankers in recent years would appear to have given a certain stimulus to the steam turbine and it was heartening to see that Seahorse engines could match this requirement with the multi-engined installation through a suitable gearbox.

The use of controllable pitch propellers for the multi-engined installation presented the classic case for this type of propeller, giving as it did the facility to choose the optimum pitch position for each power requirement should service conditions demand. It had to be remembered however when making an economic case that a controllable pitch propeller with a boss/diameter ratio around 30 per cent was 2 to 3 per cent less efficient than a fixed pitch installation having a boss/diameter ratio of about 18 per cent. Furthermore, when choosing pitch

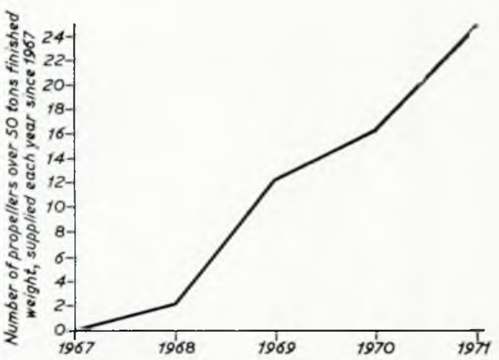


FIG. 38

settings remote from the design condition, such as the case when using one engine of a twin screw installation, the blade sections were working at unfortunate incidence angles and the geometry of the blade sections was unfavourably affected. The efficiency compared with a purpose designed fixed pitch propeller was therefore further decreased. The facility for manoeuvring and remote control, as well as the availability of maximum power for any condition might, of course, more than offset this diminution in efficiency.

MR. G. B. FRASER, M.I.Mar.E., said that regarding the fuel injector position, the spray from this injector shown under ideal circumstances seemed to be very close to the high temperature area of the piston (Fig. 11) and it was very likely that after deterioration of the nozzle, fuel would empinge on this part of the piston, which could be a source of considerable trouble when the engine was in service.

The joints of the exhaust gas manifolds on most engines had proved to be a constant source of trouble and involved quite considerable maintenance. Had any special care been taken to ensure that the joints on this engine would survive reliably during service?

The interchange of piston heads could be a source of trouble because the lower piston head was oil cooled and the upper piston head was water cooled, and in all probability, oil would remain in the oil cooled piston heads in some form or other. Should this enter the cooling water system, serious trouble could be experienced.

In comparing the installation powers quoted in the paper, it had been found that in installed medium speed engines during ships' trials, the flexible coupling, clutch and gearbox losses ranged from 7 per cent to over 11 per cent.

Had the authors had similar experience? If so, had they taken this into consideration? If not, what losses were taken into consideration when this theoretical exercise was conducted?

Mr. Fraser asked the authors if, in the trial programmes for this engine, any provision had been made for actually measuring the power of the engines in service. This facet had proved to be a problem especially with multi-engine installations in service. It was thought that at present there was no system available to measure the output of a medium speed engine accurately.

It was probably early days to consider this point, but obviously maintenance procedures and special equipment would be required to overhaul this engine. Had any provisions been made for this because once the engine was in service very little could be done about the development of such aspects.

It would be very interesting if the authors could give some idea of the de-rating which would be required on this engine when operating under tropical conditions. This was of particular relevance considering the very high pressure drop across the inlet ports of this engine.

MR. J. H. MILTON, M.I.Mar.E., asked to what degree the efficiency of the engine was affected when running astern, in view of the 10° advance angle given to the top piston?

From the authors' discourse on combustion efficiency, it would appear that the reputation of the original Doxford oil engine for smooth operation—due largely to the comparatively low cylinder pressures—was being abandoned in this new design, where cylinder pressures of up to 1500 lb/in² were envisaged. Did not such combustion pressures approach the characteristics of an explosion, and would this not have adverse effects on running gear, and, in particular, crossheads?

The design of the one-piece cylinder liner embodying tangential drilled waterways in way of the combustion space, thereby obviating a water jacket at this point, was an interesting innovation.

Whether the expectancy of long life of these liners from the fatigue aspect, predicted by the computer, would be confirmed in service, only time would tell.

The history of top piston cooling arrangements of Doxford engines was one of constant change, swinging links, flexible hoses, telescopic pipes and, now, back to swinging links. Presumably swinging links were considered to be more suitable for the high revolutions.

The authors had stated that bearing and crankshaft failures should be attributed to thermal distortion and, for that reason, decided that this engine should be self-rigid and isolated from its foundation.

This philosophy was sound from the alignment within the engine aspect, but was not conducive to the maintenance of satisfactory alignment to the gearing.

It would be interesting to know what variations in centre lines of crankshaft and gear pinions were envisaged, bearing in mind two factors:

a) fine tooth couplings were prone to fretting failures even when transmitting the smooth torque of a turbine:
b) the vessel was not a stable platform.

It was concluded that the engine was not to be lifted into the vessel in one piece; would the authors state what arrangements they were proposing to make for ensuring that the shop alignment was reproduced on board ship, as crank deflexion readings alone for this purpose would not be adequate?

IR. A. HOOTSEN said that the authors had provided a wealth of information on the design of and the philosophy behind their very interesting Seahorse engine. They were, however, very brief

TABLE IV—MARKET SHARES OF MEDIUM SPEED DIESEL ENGINES
in percentages of numbers of ships on order 1 January 1971 in the Western Hemisphere

Ship type		Per propulsion power range 3000–6000 bhp	6001–10 000	10 001–18 000	18 001–27 000	Per ship type % MS	% ship's × market share	= % M.S. market share
Tankers		83	0	18	0	19	21	4
Bulk carriers		60	36	14	0	15	18	3
Cargo ships		85	38	28	26	45	52	23
Ferries		100	100	93	83	93	9	8
	% M.S.	84	38	30	13	Totals:		
	×							
per power	% total market share	18	21	44	17		100	
	=							
range	% M.S. market share	15	8	13	2			38

in their review of the development of the geared Diesel propulsion market, with their simple statement that:

"Geared marine Diesel installations have been available for many years, but they have hitherto never succeeded in obtaining more than a small portion of the market, confined in the main to specialized tonnage."

This statement struck Mr. Hootsen as unrealistic and disparaging of the position of the trunk piston, medium speed Diesel engine.

Table IV illustrated the factual position of this engine type, with an overall market share of 38 per cent by numbers of ships and (by multiplication per power range with the average output per range) 29 per cent of the total output.

The upper limit of 27 000 bhp had been set by the maximum output up to now of medium speed propulsion in the four applications in the table. It would rise with the advent of the second generation of trunk piston, medium speed Diesel engines and of course by the introduction of the Seahorse engine. This new impulse would further improve the progress of medium speed propulsion which had already shown an average annual growth rate in power of 32 per cent over the last seven years, against an average of 18 per cent for crosshead Diesels and turbines over the same period.

"Medium speed propulsion has thus amply passed the stage of a small portion of the market and can be expected to grow into majority within the next 10 years."

The table also clarified the position of medium speed propulsion in relation to the type of ship, as it illustrated that 68 per cent was applied in normal cargo ships and bulk carriers and that 45 per cent of the cargo ships with 3000–27 000 bhp preferred this engine type.

This point was also illustrated in Fig. 39, with 62 per cent for cargo ships and bulk carriers and the rest for specialized ships; including also icebreakers, dredgers etc, in this case.

The diagram moreover illustrated that the factual impulse for the progressive growth of medium speed propulsion had to come from its acceptance for cargo ships and bulk carriers and that a recession in orders for these ships in 1969 practically stopped it.

"Medium speed propulsion has thus also completely outgrown its original confinement to specialized tonnage to find its major application now in normal cargo transport."

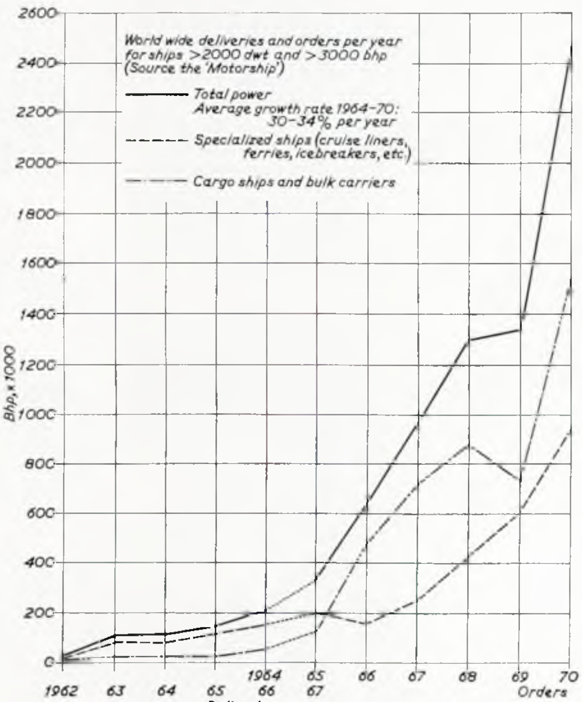


FIG. 39—*Progress of medium speed Diesel engines*

MR. D. CASTLE was interested to note that under the heading "Production Costs" it was, in effect, accepted that the smaller the cylinder size, the lower the production cost per horsepower.

Certainly this argument broke down if production capacity exceeded demand and it was refreshing to find that the authors had taken due account of such factors, even if their predisposition to a crosshead design had barred them from the logical result, which would have been a trunk piston engine.

But Fig. 2 was a quite original attempt to present the case for geared medium speed engines and the authors were to be congratulated on their approach.

Mr. Castle was nevertheless very pleased when Mr. Crowdy introduced a note of ridicule into his presentation. He had underlined what Mr. Castle always said to owners, which was "Please do your own sums for your own operations and don't take the engine or shipbuilders' word for cost analysis."

Early in the paper the authors had stated their case for justifying the bulk which resulted from their crosshead and diaphragm. It had been claimed to permit the burning of heavy fuel and cylinder lubrication with highly alkaline lubricant and yet enable the engine crankcase to use a straight mineral oil, which no highly rated trunk piston could tolerate with safety. They also claimed that this same oil could be used in the gearbox, thus presumably simplifying storage.

Mr. Castle pointed out that heavy fuels had been used in trunk piston engines since the mid 1950s; for example, there were now 252 ships in service with Pielstick PC engines burning heavy oil. Moreover, Mr. Castle's company had had many years of experience of trunk piston two-stroke engines in which straight run oil was used in the crankcase along with highly alkaline oil in cylinder feeds.

Regarding gearbox oil, he could see no reason why, if a given straight run oil was suitable for gearboxes, the corresponding fortified engine oil should not be equally suitable, and indeed he had strong reservations about using straight mineral oil in an engine crankcase where it was called upon to cool pistons with a metal temperature of 500°C (932°F). In one section of the paper the authors appeared to express similar reservations. He wondered if actual engine tests had given any reassurance on this point.

Mr. Butler had been clearly relieved to find that he had load reversal at his top ends. Inertia on the oil column in the connecting rod was at the outlet side of the oiling system and when the pin lifted, the oil would return to flow in under the pin. This could mitigate against the cooling effect hoped for, but certainly improved the overall situation.

It was not altogether surprising that the authors were concerned about fuel impingement; with radially mounted injectors this must be a worrying aspect.

Mr. Castle agreed with the authors that a turbocharger of the efficiency necessary to "close the loop" so to speak, for all but a single matched engine load, was not a realistic expectation.

The consequent need for a balancing parallel flow blower was a complication which must spoil the thermal efficiency and which detracted from the simplicity of the opposed piston engine, of which the Seahorse was otherwise a fine example. He hoped that the engine test results would reward the authors and their teams for the extremely thorough design and rig work reported in the paper.

MR. A. C. WORDSWORTH said that it was quite accepted that, on a modern engine, shell bearings would be used, but with the majority of bearing failures being due to fatigue, would it not have been worth using some bearing material with better fatigue resistance than white metal, e.g. aluminium tin?

Fig. 17 showed top end bearing shells as having oilways with blind ends. Was that in fact the case or was any leak off provided to increase the oil flow at this point and thus improve the cooling of this bore?

It was said that the acceleration of the hollow centre connecting rod increased the oil pressure at the top end bearing at the critical period, but would not this improvement be at the expense of the oil pressure at the bottom end?

In the design of the engine it was originally intended to mount it on three points. What was the reason for the additional fourth foot?

What was the reason for the adoption of the somewhat heavier structure, Fig. 19, rather than a lighter structure which, was decided on for the engine model, as well as by Lloyd's Register?

MR. J. NEUMANN, B.Sc., A.M.I.Mar.E., said that the authors' approach to the overall difficult problem of designing for future conditions—those to be met in ten or fifteen years' time—had been tackled in a scientific way, rather than by subjective argument, and the diagrams and tables were of great interest. Included was a fascinating discussion on the choice of revolutions and cylinder bores.

Mr. Neumann was glad to hear Mr. Sinclair's comments on Fig. 2 and the improvement in Q.P.C. which the authors claimed. The only remark he would make was perhaps to sound a little note of caution. Those figures would certainly apply to some ships but not to all ships. Referring to Fig. 2, had the authors taken into account the increase in the cost of the shafting, gearing and propeller as shaft speed was reduced?

The paper was authoritative on Diesel design matters; among the several ingenious features included was the proposed mounting system. The paper was just a little less convincing on the installation problems. For example: the easy suggestion of fitting a dog clutch in a multi-engine installation, with however, the attendant problems of cutting out and cutting in an engine with a ship under way. A further example was perhaps the short discussion on the generating station, where there was no mention of any maintenance aspects.

The slide shown at the meeting on oil film thicknesses under various conditions was interesting. Mr. Neumann's company had recently measured the oil film thicknesses in high speed shafts under operational conditions, and the results had produced one or two surprises. Mr. Neumann asked if the authors had any measurements to support the calculated values shown.

Correspondence

DR. P. A. MILNE, B.Sc., A.M.I.Mar.E., said that emphasis had quite rightly been placed on economics, but little background information was provided to justify the claims made in various parts of the paper. It would be interesting to know what fuel consumption and cost information, together with number of days at sea per year were used in the calculations in the opening section where the authors had discussed the advantage of being able to choose the optimum propeller revolutions without any restriction being imposed by the machinery.

Again, in Appendix 1 there was no supporting analysis, only a series of figures showing the advantages to be obtained by saving installation weight and simplifying the work involved. This was not meant as a criticism of the figures but an enquiry as to whether the authors had, in fact, given enough stress to the advantages their design might offer. One recent paper* described a study which compared the economics of medium and slow speed Diesel machinery. Many of the features mentioned by the authors such as a saving in weight and engine room volume, were evident as the engine speed was increased. However, the features peculiar to the engine described in the present paper, were that the fuel consumption was of the same order as existing slow speed Diesels, it had the ability to use a wider range of fuels, it gave a lower lubricating oil consumption and used a cheaper mineral oil. This latter feature was one of the main advantages of the slow speed Diesel in the 1967 study and was, in fact, eliminated by the design described in this paper.

The capital cost advantage shown in this earlier study could be further increased by taking advantage of the batch production potential resulting from the use of standard features such as a common cylinder bore for a wide range of powers. Previous difficulties in the engine building industry had undoubtedly resulted from an inability to rationalize production on a series basis, thus reducing machining times and minimizing stock levels. Machine ships also dictated the volume of output and therefore had a critical effect, not only on labour costs but on overhead recovery.

Maintenance costs in the 1967 study were shown to be in favour of the slow speed Diesel mainly as the result of the larger number of components in the medium speed installation, although in fact, even after allowing for this factor, the differences were not significant at lower powers. At higher powers the main competitor was the steam turbine and the maintenance feature became more important. Information collected from Norwegian shipowners showed that the slow speed Diesel maintenance costs increased rapidly with power and this was probably due to the size of the components which were now beyond manageable proportions. The higher ratings now used also made the condition of the main engine more critical—otherwise balance and other vibration features were upset. "Off hire" time was a further advantage of the steam turbine and this cap might be closed with the improved availability of a multi-engine installation. When choosing to develop the new engine, had the authors concluded that the economics and engineering feasibility of the slow speed Diesel were now at their limit. Could the authors also say what shore testing programme they envisaged before an actual installation in a ship, because this could have a critical effect on the initial reliability of their design and ultimately affect its market potential.

When discussing marine installations, a claim had been made to the effect that 60 tons would be saved from the structure in the double bottom. In high powered slow speed Diesel installations additional structure was required in way of the engines, but this was partly to ensure that the machinery and its associated shafting had an adequately stiff propulsion platform and also to avoid aft and vibration difficulties. Large savings in steel weight in this area would, therefore, have to be treated with some caution.

Dr. Milne's own experience in medium speed Diesel installations would in general, support the alignment arrangements described by the author, particularly the installation of a quill shaft and a coupling able to accommodate misalignment between the engine and the pinion in the gearbox. Some difficulty could, however, be anticipated with the installation arrangements shown in Fig. 27, where the drain tanks for the machinery were adjacent to each other on the centre line of the vessel. Measurements taken in service and a mathematical analysis carried out by Newcastle Polytechnic showed that this not only caused vertical movement of the machinery, but also appreciable athwartship displacement by tipping of the engines in an outward direction. To minimize this affect the drain tanks should be separated from each other, or better still, a wet sump arrangement adopted for the machinery. Was it possible to run the authors' engine in the wet sump condition?

Towards the end of the paper the authors had described arrangements made on the test bed for the engine for both control and the collection of data. Had any consideration been given to providing a standard controls and instrumentation package with the Seahorse engine? There would appear to be some advantage in adopting a limited number of controls and instrumentation arrangements from which a customer could select. It had also been mentioned that arrangements had been made for measuring liner and piston temperatures. Would tests be carried out to establish the temperature limits within which the cooling water should be controlled to ensure that the temperature

* Neumann, J. and Carr, J. 1967. "The Use of Medium Speed Geared Diesel Engines for Ocean-going Merchant Ship Propulsion", *Trans.I.Mar.E.*, Vol. 79, p. 89.

variations of components within the engine were kept at acceptable levels? There was now a considerable amount of information on how to achieve various levels of control but no definition of what levels were necessary.

A number of medium speed designs offered engine driven auxiliaries. This had the advantage of offering a package in which a number of the engine services were self-contained, reducing the installation costs. In addition to this, the generating capacity required on board was substantially reduced and the drive for the auxiliaries provided at main engine efficiency using a lower grade of fuel than that burned in Diesel generators. As the Seahorse engine already had a number of auxiliary drives, did the authors intend offering engine driven services as an optional feature?

MR. M. HARPER, M.I.Mar.E., wrote that the statement that a maximum piston crown temperature of 500°C "is quite satisfactory for a mild steel piston" could prove to be optimistic when using oil cooling. However, any weaknesses in this area ought to be discovered during the planned two year endurance test.

Mr. Harper did not expect the fitting of a mechanically driven blower to find favour with many operators, and this could be an important factor in the final decision regarding the type of engine to be installed in a particular vessel.

To assume the potential of the Seahorse engine more accurately it should be compared with other medium speed engines rather than a slow speed engine.

Table V showed that Seahorse did not compare favourably with its competitors on the basis of either height or weight and this could prejudice its adoption for car ferries, cruise vessels etc.

DR. D. J. COLLINS, M.I.Mar.E., said that his particular interest in this excellent paper was the water cooling system. Would distilled water be specified for the jacket and upper piston cooling which was normal for Doxford engines and if so, by what means would the distilled water be produced?

Concerning the oblique tangential cooling holes in the liner, what diameter were these and would there not be a tendency for choking and possibly corrosion and/or erosion?

Would the cooling system be the closed or pressurized type using a header tank, or the traditional Doxford system using a base circulating tank?

Mr. Collins' company had had many years of experience in the chemical treatment of cooling systems for inhibition of corrosion and scale prevention. Experience had shown that successful results in this field depended largely on the design of the cooling water system.

MR. A. HILL, B.Sc., M.I.Mar.E., said in a written contribution that the Seahorse engine was a most interesting concept which appeared to combine the advantages of both the geared medium speed four-stroke engine and the direct coupled low speed two-stroke engine, whilst at the same time avoiding the shortcomings of these two engine types.

It offered the ability to burn residual fuels with a clean crankcase, a high power output with a reasonable number of cylinders and, through the gearing, an output speed to match any propeller desiderata. The saving in weight and length should be very attractive in that it allowed additional cargo space. The low vibration and noise levels which one would expect, would always be welcome and should make the engine of particular relevance for passenger carrying vessels.

The provision for using the main engine as a high power source when in port, either for electric generation or for direct drive to cargo pumps, should be very useful. In the case of tankers and V.L.C.C. it would seem to offer the prospect of both reduced capital cost and a higher cargo discharge rate.

Mr. Hill was impressed with the fundamental approach the authors had made to the many and various design problems, and the boldness and ingenuity of their solutions.

The adoption of point supports involving the provision of adequate "girder stiffness" in the engine itself should overcome many of the present problems. However, the use of four support points could still result in the engine being subjected to torsional forces. A three-point support would be preferable. Would the authors say why they chose four points?

The savings offered in installed weight and space were clearly substantial. However, he had some comments to make with reference Fig. 29.

The authors had stated that the double bottom stiffness was reduced. It was necessary to exercise great care here and, whilst one would agree that the usual long "rigid" fore and aft girder was no longer needed for engine support, it would still be necessary to provide supports of adequate strength to carry the high point loads.

In addition, it would be necessary to provide adequate stiffness to prevent torsional loads being applied to the engine by four support points. Investigations in large vessels had shown that the double bottom deflected appreciably between loaded and ballast conditions and in an installation, as Fig. 24, would produce torsional loads on the engine supports.

A possible solution was to design the forward engine supports so that they carried the appropriate loads, but could not transmit flexure of the double bottom in such a way as to change the support loads to cause torsional forces on the engine.

The stiffness had to be adequate to avoid risk of resonant vibrations which could be excited by engines or propeller.

The main thrust block was combined with the forward end of the gearbox. Whilst the fitting of the main thrust in the gearbox was a convenient solution, acceptable for low to medium powers, much higher powers, 20 000 shp and over were being considered and this was therefore not a desirable feature.

The transmission through the gearbox structure of the large axial forces from the thrust was not consistent with the maintenance of the tooth and bearing alignment necessary for high duty gears.

The design of gearbox seating in Fig. 29 was relatively short and relied on the double bottom stiffness, which was rarely adequate. The seating was subjected to large bending moments from the main thrust and would almost certainly deflect, tending to cause gearcase distortion and misalingment with engines and shafting. It could result in fretting of the coupling splines or a transfer of undesirable forces or vibration to the engine.

It would be beneficial to extend the gearcase seating to include the aft engine seatings and thereby provide greatly increased stiffness against bending in this area.

However, Mr. Hill suggested a better solution would be to design the gearing only to transmit torque, using a similar philosophy regarding the supports as used for the engines, and adopt an independent thrust block.

Such a block would be preferably of centre-line mounted type and could be fitted in place of the forward plummer bearing. In this position, a very stiff seating could easily be provided and

TABLE V

	BHP	Dimensions, mm: Length Top	Length Bottom	Width	Height	Weight tonnes
Four-cyl Seahorse	10 000	9060	7437	2926	8034	160
Werkspoor 20 TM410	10 000	9686	8344	3500	4078	155
Six-cyl Seahorse	15 000	12860	10237	2926	8034	224
Pielstick 18 PC3V	15 000	10870	9935	3660	4380	160

there would be minimal transfer of deflexions caused by the thrust to gearbox and engine structure or supports.

MR. G. C. VOLCY, M.Sc., M.I.Mar.E., said that he was interested to note that the Seahorse engine had been designed on the concept of a stiff "I" girder beam instead of the conventional stiff bedplate principle. This new principle (see Fig. 40) had been presented in a paper in 1966,* but the theory had been refuted at the time by various specialists in the field of marine engineering, who, as the authors had rightly said, supported the conventional stiff bedplate. He agreed with the authors that an engine structure should be considered as:

i) a bedplate, representing the lower flange of the beam;
ii) a triangular structure, representing the web of the beam;
iii) a cylinder block, representing the upper flange of the beam.

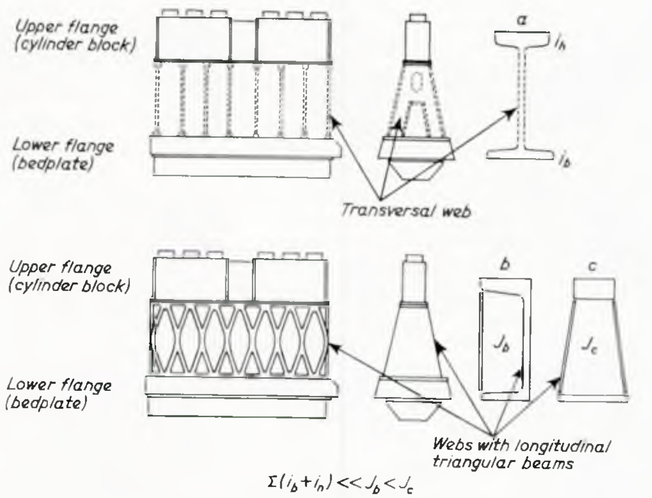


FIG. 40—*Development of the concept of the engine as a beam*

Many problems involving broken crankshafts and damage to main bearings could be attributed to the distortion of the ships' structure and double bottom tanks in way of the main engines, and the problem was not solved simply by increasing the stiffness of the bedplate. Mr. Bourceau and he had commented on this point in 1964† and he thought that the weakness in the hull structure had held back the installation of Diesel engines in large tankers.

He was also glad to note the arrangement for chocking the main engines at four points which would prevent movement of the ship's structure from affecting the engine. He would, however, like the authors to comment on the calculations of natural frequencies in the vertical and transverse modes of the assembly containing the engine and the double bottom tanks.

Bureau Veritas was carrying out a very sophisticated study on the deformation of the ship's structure and crankshaft alignment problems, and he hoped to report on their findings in the near future.

Some of the results of the research were however, given in a recent paper.‡

MR. J. F. WARRINER wrote that it was stated in the paper that the oil film thicknesses were computed for the crankshaft bearings using hydrodynamic theory, and that the minimum film thicknesses would probably be greater than predicted due to the "squeeze" effect produced by the displacement between the journal and bearing. The computations were carried out on his company's computer programme which took into account both wedge (commonly termed hydrodynamic) and squeeze effects.

One of the assumptions made in the computations was that the shaft and bearing bore were rigid and truly cylindrical. In practice, the bearing housings and bores deflected throughout the load cycle and this was one of the effects which could contribute to the actual oil film thickness being larger than computed. Even so, the computations for the crankshaft bearings could be used as a basis for comparison. On this basis, the figures obtained for the Seahorse four-cylinder engine indicated that satisfactory performance should be obtained, and in fact, showed a good safety margin, based on general past experience obtained by his company in interpreting these computations.

The work by the author on crankcase and crankshaft stiffnesses, leading eventually to modified polar load diagrams was revealing. It was to be hoped that the findings could be confirmed on full scale tests. The simplifying assumption of an infinitely stiff structure relative to the crankshaft, with the shaft pin jointed at each main bearing, usually had to be made for oil film computations, because the true relative stiffnesses were unknown. Even so, the dynamic distortion of the bearing housing could have just as significant an effect in relation to the oil film conditions.

Comments were made on the advantages of pre-finished insert liners, together with the extra care required in the preparation of the housings without resorting to scraping or filing. Mr Warriner's company subscribed to these comments, but would add that it should also be unnecessary to scrape the bores of the shells, and no scraping allowance was provided in the finished bore.

During the discussion, one contributor had questioned why tin aluminium or overlay plated copper lead bearings were not utilized to carry higher loadings. In fact, some tin aluminium bearings, of high tin content, were being tested in the side connecting rods of the four-cylinder prototype engine. This particular tin aluminium had a surface hardness almost identical to that of white metal, but with a fatigue strength some 50 per cent greater. The surface properties were similar to those of white metal with respect to conformability and embeddability, but the structure was less sensitive to high temperature.

Where the load could be carried by white metal bearings, these were used because of their extremely good conformability and embeddability. Making the Seahorse bearings smaller, to take advantage of the higher fatigue strength of stronger bearings, would reduce oil film thicknesses, therefore imposing more severe conditions.

Another contributor had asked whether the oil film thickness figures could be backed up by measurement. To obtain such measurements would be extremely involved and whilst such measurements would be valuable, they would normally be considered above that required in a normal engine development programme. As indicated earlier, the oil film thickness computations did not produce absolute results and could only be used as a comparator, against past experience to determine whether problems were likely to be encountered and hence forestall them at the design stage.

* Bourceau, G. and Volcy, G. 1966. "Some Aspects of the Behaviour in service of Crankshafts and their Bearings". ATMA.

† Bourceau, G. and Volcy, G. 1964. "The Influence on the Hull Flexibility of Main Propulsive Units in Motorships". *Nouveautes Techniques Maritimes.*

‡ Toms, A. E. and Martyn, D. K. 1971. "Whirling of Line Shafting". *I.Mar.E.* Vibrations Section Symposium, December.

Authors' Reply

In reply, the authors referred to the question of cost, raised by Mr. Yellowley. Although the material saving in the geared engine compared with a direct drive engine was sufficient to compensate for the cost of gears and couplings, it was hoped that there would be a further useful saving by rationalization of production methods following the use of one cylinder size for a very wide range of installation powers.

Messrs Morrison, Milne, Anderson, Wesselo, Castle and Neumann had queried the basis of the economic comparison and the likely first cost of the engine. The engine/gearbox unit price used in the paper was £25/hp, this being representative of the slow speed direct drive engine prices prevailing at the time the paper was prepared. The authors were confident that they would be able to offer the engine at a price that would be competitive in the market whilst retaining an adequate margin to recover development costs. Annual charges and capital costs could only be compared by applying appropriate life and discount factors In the paper, capital cost was assumed to be equivalent to an annual charge of 15 per cent of that cost or, in other words, annual charges $\times$ $6^2/3$ = capital cost. This was equivalent to capitalization at 5½ per cent discount rate over a life of only nine years, or, as another example, at 12½ per cent discount rate over 15 years.

The prototype engine was not at present fitted with water separators after the air coolers, but the design allowed space for such fittings if they proved necessary.

The authors agreed that not much improvement had been made in indicated fuel consumption in large slow speed engines over the past few years but would suggest that this was mainly because it had been necessary to increase powers with minimal increase in maximum pressure in order to avoid big increases in running gear scantlings. On the other hand, small vehicle engines which from their nature had rather low mechanical efficiencies attained better indicated thermal efficiency than slow speed engines, largely by using high maximum pressures. Mr. Yellowley would agree that if this could be done on a very small engine it could also be done on a larger engine provided the higher maximum pressure could be accepted.

The oil flow from the lower piston was controlled by an orifice at the base of the piston rod in order to ensure that ample oil was available for bearing lubrication. Although the prototype was starting test with normal piston rings, a number of special type rings had been made available and would be tried during the course of the development programme.

Some cracking had occurred in slow speed engines in the centre top end bearing shells with the design in which there were two shell bearings and a centre pad, due to failure to achieve correct load sharing. So far, none had been reported with the type of bearing in use for some time on slow speed engines, described in the paper. Even though excellent results had been obtained with direct white metalled bearings since dovetails had been eliminated, the thin shell type had the advantage of easier replacement should this be required away from the home base.

In reply to Mr. Conn, it was true that the five-cylinder engine with equally spaced cranks had a small secondary unbalanced couple. However, if there was danger of this small couple initiating hull vibration, it was quite practicable to eliminate it completely by a very small charge in crank angles which would leave only a minute unbalanced secondary force.

It was hoped that the total noise level from the engine would be much below 110 dB(A), but although the present engine covers were largely plain steel plates, space had been allowed for fitting double walled acoustic damping covers should these be required.

In reply to Mr. Rose, the opportunity to test both types of injection on the engine had not yet arisen, but such tests would be carried out in the near future. The full analysis of the film thickness for No. 1 main bearing had not been carried out because there was such a large margin in this particular bearing. The centre top end bearings had a lead overlay to the whitemetal—this had been the makers practice for some time on slow speed engines. The lead overlay appeared to help considerably in obtaining an even bed during the early running of the engine.

Mr. Morrison had pointed out that the smooth curves shown in Fig. 2 were not always directly applicable since inevitably all Diesel engines were marketed with discrete steps in size. Sttis-atically, however, they represented the true position and the authors naturally prefered that comparisons should be made with specific geared crosshead engines as a reference, rather than specific slow speed direct driven engines.

Space had precluded inclusion in this paper of the data on which unit electrical costs were computed, but these would be published under the auspices of The Diesel Engineers and Users Association.*

With regard to the facilitating of automatic control, fairly elaborate provision had been made for measuring flows and pressure drops on the test bed to ensure that on production engines, the builders would be able to specify the correct control characteristics for automatic valves. The authors appreciated the importance of this point since many installations in the past had been unstable as a result of thermostatically controlled valves being too large for their purpose.

Experience of existing engines showed that although it was normally considered that 220°C was the highest metal temperature which a straight mineral oil should be allowed to contact, higher temperatures up to 260°C were satisfactory in the hot corner of the piston crown, provided there was sufficient oil flow to produce a scouring action. The estimated temperatures in the Seahorse lower piston were no higher than those in existing slow speed engines.

It was gratifying to have an independent testimonial regarding the reliability of slow speed marine engines. During design every component part of the new engine had been compared with its slow speed counterpart to ensure that its service reliability would be equal or better. Because of the use of only one cylinder size, greater attention to each detail had been given—not only in design but also to inspection and testing—than was possible when a wide range of cylinder sizes was being manufactured,

In reply to Professor Chambers' question, the engine was designed for an ambient air temperature of 43°C and a sea water temperature of 32°C. This would indeed give the engine some margin in relation to the British Standard for land application. No tests had yet been made with the mechanical blower out of action, but these would be done. Judging from previous engines, it was to be expected that without the mechanical blower rather more starting air would need to be used to get the turbochargers moving, and the power range would be limited at both its upper and lower ends. Nevertheless it should be perfectly possible to get the ship to port under these unusual circumstances. One advantage of the constant pressure turbocharger arrangement, with two turbochargers, was that if one of these should fail the other could continue to provide scavenge air without the liability to surging which could occur on a pulse charged engine under similar circumstances.

Mr. Manson had raised a most interesting point about heat treatment of the cylinder liners. So far this had not been carried out, but careful measurements would be taken to check for distortion so that, if necessary, a stress relieving process could be carried out on future engines. The authors' experience of chrome molybdenum steels for piston crowns had been that the added hot strength barely compensated for the reduced conductivity compared with mild steel, and there was therefore little gain in using the alloyed material at the temperatures expected in this engine. From previous experience of pistons running at similar temperatures, it was not expected that the oil cooled inner surface

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* Butler, J. F. and Crowdy, E. P. 1972. "Industrial Application of the Doxford Seahorse Engine". Paper read at D.E.U.A. on 17 February.

of the crowns would need cleaning before the pistons were due for reconditioning for other reasons.

Messrs Manson, Milne, Eckard, Milton, Wordsworth, Hill and Volcy had all raised doubts about the satisfactory operation of the drive between the engine and gearbox as a consequence of relative movement between these two components, possibly of a vibratory nature, if the scantlings of the double bottom were reduced. Naturally, the engine builders would work in close concert with the shipbuilders to ensure that the resonant frequency of the engine/double bottom composite was avoided. The length and diameter of the quill shaft was such that the fine tooth coupling at the free end would be subject to minimal angular displacement. If the double bottom structure had extreme flexibility it would be possible to reduce the angularity further by converting the quill shaft into a cardan shaft, by fitting fine tooth couplings at both ends.

Dr. Milne had queried whether the authors had given enough stress in their economic comparison to the advantages their design might offer. The authors felt that he was aware of their conservative approach in such matters, but would like to take the opportunity of publishing the précis of economic advantage shown in Table VI. The specific fuel consumption assumed in this study was 162 g/bhph and the number of hours at sea per year were taken as 8000.

He had also asked whether the authors had concluded that the economics and engineering feasibility of the slow speed Diesel were at their limit. This was the case and the view was formed that it would be more profitable to develop an engine which could produce more total power per shaft for future needs, without exceeding known parameters of thermal stress and without requiring castings and forgings more massive than those so far produced. A shore testing programme lasting two years had been planned to ensure adequate endurance testing to enable any possible failings to be corrected before they occurred at sea. On the other hand, it was hoped that a limited number of sea installations would be put in hand before the completion of the two-year programme.

During testing, information obtained from the controls and instrumentation fitted would enable the builders to produce a detailed specification to enable other instrument makers to offer standard packages.

Most potential customers appeared to prefer separately driven auxiliaries, but the gearing arrangement on this engine was such that if there should be sufficient demand it would be fairly simple to make some of these engine driven, in particular perhaps, the lubricating oil pump.

TABLE VI—PRESENT DAY VALUE DIFFERENTIALS (£/hp)

	Origin of differential	Engine type: Trunk piston medium speed	Engine type: Slow speed direct drive	Engine type: Seahorse
1	Propulsive efficiency	+ 7·5	0	+ 7·5
2	Fuel and lubricants	− 40·0	0	0
3	Weight	+ 9·0	0	+ 7·0
4	Installation costs	+ 1·5	0	+ 1·0
5	Split machinery	+ 3·5	0	+ 3·5
	Total 1–5	− 18·5	0	+ 19·0

For example the p.d.v. of a Seahorse engined:

a) 35 000 hp tanker exceeds that of a conventional medium speed engined tanker by £1 310 000

b) 10 000 hp ore carrier exceeds that of a slow speed direct drive engined ore carrier by £127 500

c) 25 000 hp ore carrier exceeds that of a slow speed direct drive engined ore carrier by £475 000

The authors felt that Mr. Eckhard presumably had his tongue in his cheek when he suggested that the company had initially made a free issue of engines until they had been proved successful at sea. It was contended that with the proposed method of engine mounting, shore trials would better represent sea conditions than had been the case hitherto.

Fig. 29 showed normal engine room casings which facilitated the arrangement of the exhaust uptakes and auxiliary boilers, if fitted. The authors considered that the type of engine described lent itself to an engine room closed in by a continuous main deck.

As regards "shock" standards it should be noted that all the structural components of the engine were manufactured from steel rather than cast iron and its inherent shock resistance was of a very high order. Concentration of the chocks into only four points would require adequate attention to the double bottom design, but the main machinery would probably survive any hull distortion short of complete rupture of the shell plating.

They were grateful for Mr. French's description of the means used for measuring piston and liner temperatures, which was a valuable addition to the paper. It was agreed that trunk piston engines tended to have lower liner wear rates than crosshead engines, possibly because of better oil distribution, but despite this, low speed engines could attain liner life of up to 50 000 h with a total diametral wear of 1/200 of the bore. The medium speed crosshead Diesel had less liner surface per horsepower to keep oil wetted, and the higher speed should help to maintain the oil film and so give usefully less wear than on direct drive engines. The crosshead engine had the advantage that the lubricating oil consumption was independent of liner wear so that a greater increase in diameter could be tolerated before the liner needed to be changed.

In reply to Mr. Anderson, it was agreed that long stroke direct drive engines had quite a high piston speed, but this was at the expense of height, weight and cost, and they thought he would agree that any designer given a free choice of speed, would have used higher revolutions and a shorter stroke. The fuel and air amounts and thermal loading given in the paper were based on theoretical considerations, supported by extrapolation from other engines of similar power. The fuel consumption given referred to the engine output shaft with fuel of up to 3500 Redwood viscosity with a nominal calorific value of 10 000 cal/g and was based on mechanical efficiency of 88 per cent at full load.

The liner and piston wall temperatures were calculated by the formulae given in Ref. 2, and checked independently using a different method. The ring groove temperatures were similar to those measured in previous engines giving satisfactory service.

The bolt pre-stressing in the crosshead pin fixing had been set at a level sufficiently high to avoid relative movement between journal and shoe to prevent fretting. The point raised about the crankshaft design was pertinent, but the same risk applied to the shrink between crankpin and web in any fully built shaft, and both Mr. Anderson's company and the authors had many such shafts in successful operation.

The matching of turbochargers and engine driven blower at part loads was very complex, but calculations indicated that although the percentage of total air delivered by the auxiliary blower would increase at low load, the build of the turbochargers could be chosen to avoid harmful surge at all loads without losing more than 1–1½ per cent of their potential full load efficiency.

Mr. Wesselo's most instructive comments showed that with very good design, a four-stroke trunk piston engine could burn heavy fuel with minimal increase in maintenance costs and did much to strengthen the case for geared engines. It was conceded that if the lubricating oil consumption was at a sufficiently high rate, the quality of the oil remaining in the sump would remain sufficiently high to obviate changing due to continuous dilution by make up supplies.

The authors were grateful for Mr. Wesselo's various comments about his design study for a large opposed piston engine which, in general, confirmed their own views, but they had to disagree about the relative stokes of the pistons. It could be shown that, because of the necessary pre-opening of the exhaust ports, the maximum air and gas flow for any given proportion of useful working stroke was obtained with the exhaust piston stroke about half that of the scavenge piston. The single crankshaft design had the major advantage of avoiding firing loads on the main bearings and engine structure.

The engine described was designed with low swirl to minimize heat transfer to the pistons and to ensure satisfactory cooling without complications such as the use of liquid metal. The low temperature at the bottom of the liner was due to the cooling effect of the scavenge air. The gas slide liner temperature in way of the water cooling was about 90°C with cooling water entering the jacket at about 65°C.

It was gratifying to learn from Mr. Sinclair that facilities now existed to make propellers up to 80 tons in weight since this further enhanced the advantages of the geared engine installation. It was agreed that the use of lower rev/min on large tankers in recent years had given a certain stimulus to the steam turbine and it was certainly anticipated that the higher power geared Diesel engine would be able to gain some of this market. Mr. Sinclair's commentary regarding the application of c.p. propellers to multiple engine installations was fully appreciated by the authors, if not by all the protaganists of dual speed vessels.

In reply to Mr. Fraser, any engine making reasonably full use of the air in the combustion space must spray fuel close to the piston surface to avoid leaving pockets of unburnt air. Normally short stroke engines with central injectors had to burn a fairly high proportion of the fuel after it had hit the pistons.

The authors' experience with exhaust manifolds was that it was of paramount importance to provide firm mountings for the rigid parts of the piping and to ensure that the flexible connexions were fitted at the correct length, so that they could accept the expansion of the fixed parts without overstress. Provided these considerations were applied, satisfactory life could be obtained with exhaust temperatures up to 480°C.

The main object of making the upper and lower piston heads interchangeable was to enable spares to be used in either position. If a head which had previously been used on the lower piston was fitted in the upper position, normal practice would be to degrease the cooling surfaces.

Mr. Fraser quoted transmission losses ranging from seven to over 11 per cent in the case of medium speed installations. With reduction gearing ratio in the range of 2 to 4, the power loss in the gearing was expected to be only $1\frac{1}{2}$ to 2 per cent and since, in the majority of installations, slip couplings were not envisaged, there would be no further power loss. The gains from improved propulsive efficiency would therefore outweigh transmission losses in any likely application.

The authors agreed that present methods of measuring power in service were not entirely satisfactory. During test bed trials, routine cylinder pressure measurements were being made electronically and used for measuring indicated power. It was hoped that this method could be simplified to provide means of showing indicated power at sea, but further development was needed.

During the design stage of the engine, special equipment for overhaul was made and tried on a full scale wooden mock-up. This equipment would be used throughout the test bed trials.

The design of the engine was based on sea water temperatures as given in the reply to Professor Chambers, beyond this, the design margins in the coolers were such that full power could be used under any known conditions provided the air and exhaust systems were reasonably clean.

In reply to Mr. Milton, it was planned to carry out tests with the engine running astern and, if possible, further tests with the lead angle reduced from the present eight degrees. In theory, increasing crank angle lead should have reduced thermal efficiency, but increased effective compression stroke and specific power. On an existing engine increasing cylinder pressures were bound to have an adverse effect on running gear but, in this case, the engine had been designed *ab initio* with sufficient strength and rigidity to accept maximum cylinder pressures up to 1500 lb/in^2.

Arrangements had been incorporated for taking telescopic sightings of the crankshaft alignment without removing any running gear. This was achieved by using spring loaded probes contacting the underside of the crankshaft journals. A valuable feature of the engine in this connexion was the capability of checking the relative position of the crankshaft journals and bearing with feelers, which could be applied at any point round the circumference.

The engine frame was of sufficient strength to allow any one of the four chocks to be removed so that the "daylight" between the engine and tank top could be checked against the actual chock thickness, thus providing a ready means of checking for possible torsional distortion of the double bottom, as might be prudent after heavy weather damage, or grounding.

In designing the new engine, all previous arrangements used by Doxford had been considered and it was decided that with careful engineering the swinging link arrangement would be best.

The authors thanked Mr. Hootsen for his most valuable contribution which corrected a false impression, which may inadvertently have been given, regarding the present day position of geared engines. It was the growing market share being enjoyed by the geared engine which had encouraged another slow speed engine manufacturer to embark on the new engine project designed from the outset to operate economically on residual fuels. Mr. Hootsen's table and graph endorsed the opportune timing of this development.

The authors agreed with Mr. Castle that trunk piston engines with cylinder lubricators could use two different oils but, in this case, there must be some mixing and, therefore, loss of cylinder oil into the crankcase with deterioration of the crankcase oil by combustion residues.

The authors felt somewhat relieved about the load reversal in the top end, since the engine had been designed from the start with this in mind. The adverse effect on cooling of the back flow of lubricating oil from the top end would be very small, since the oil concerned would previously only have been in the connecting rod bore.

Compared with engines with central injectors, fuel impingement was much less with radially mounted injectors and experience had shown that this arrangement with a flattened spherical combustion chamber was very tolerant of injector imperfection.

Any two-stroke engine with turbocharging on the constant pressure system heeded an auxiliary blower for part load running so it was a logical simplification to use this blower also to give extra air at full load.

Mr. Wordsworth's question on alternative bearing materials was answered by the written contribution from Mr. Warriner whom the authors also thanked for his helpful contribution on bearing oil film thicknesses.

It was true that the increase of lubricating oil pressure at the top end round i.d.c. was to some extent at the expense of the bottom end, but at this point in the cycle, due to intertia, the bottom end bearing would have been enjoying a period of running with large clearance on the side to be loaded by firing pressure so that ample oil would be available to form a film during the firing period. The blind ended grooves shown in Fig. 17 were the result of artistic license. In the actual engine there were small leak-offs cut at the ends of the axial grooves.

Mr. Wordsworth correctly pointed out that it had been originally intended to mount the engine on three points and, in company with Mr. Hill, wondered why this kinematic correct solution had been abandoned in favour of four-point support. It could be shown that the natural frequency of a series of geometrically similar engines was proportional to the square root of the modulus of elasticity and in inverse proportion to linear size. It was also clear that for a constant piston speed, the frequency of excitation would also be inversely proportional to linear size. As a consequence, the ratio of natural to exciting frequencies was independent of size for engines made of similar materials. Since the cast iron automotive engines could be mounted without external torsional restraint, it seemed that, if the frame of a marine engine could be made proportionately stiff, it could be mounted on three points. However, the stiffness figures derived from the tests carried out by Lloyd's Research Department showed that whilst no trouble should be expected with four-, five-, and six-cylinder engines, the seven-cylinder engine was likely to be too close to a resonant torsional vibration. It had therefore been decided to design for four point support so that external torsional restraints could be applied to change the resonant frequency and provide the possibility of introducing damping if necessary.

The reason for the adoption of the apparently heavier structure shown in Fig. 19 rather than that of Fig. 21 had arisen out of the decision to adopt modular cast steel units.

In reply to Mr. Neumann, the increase in cost of shafting etc with reduced revolutions had been allowed for. It was regretted that there was not space in the paper to discuss maintenance at any length, but the subject would form part of the paper on the same engine to be read to the Diesel Engineers and Users Association. No actual measurements of oil film thickness had been made, but the appearance of the bearings after test bed running indicated that loading conditions were very satisfactory.

In reply to Dr. Collins, distilled water would be used for jacket and piston cooling produced by a normal fresh water generator and treated with a suitable corrosion inhibitor.

The tangential cooling holes in the liner on the prototype engine were 16 mm diameter, but these would be increased to 20 mm on later engines to simplify drilling. Because of the straight flow path there should be little liability to choking and moderate water speed should prevent erosion. The prototype engine had been built with the traditional open cooling system because of the advantage of being able to use sight glasses, but work was proceeding on the development of a reliable flow indicator and, when this was successful, a closed system might be used.

The authors were very grateful to Mr. Hill for his written contribution to the paper and would refer him to the reply to Messrs Manson and Wordsworth regarding double bottom construction and point mounting. It was agreed that the alignment of gear teeth must not be adversely affected by propeller thrust but it was not conceded that this would necessarily be the case with the arrangement shown in Fig. 27.

Mr. Harper, like Mr. Castle, referred to oil cooling of the lower piston with maximum metal surface temperature of 500°C. The important point was that the temperature of the metal in contact with the oil was very much lower and would not exceed 250°C which experience had shown to be satisfactory. The authors agreed that the Seahorse engine was higher and heavier than vee four-cycle engines and was therefore less suitable for such vessels as car ferries. On the other hand, the smaller width and reduced number of cylinders was a big advantage for all other vessels including large tankers and bulk carriers. It was for this resaon that the comparisons in the paper had been made with slow speed direct drive engines such as were commonly used in ocean-going vessels. The mechanical blower had the major advantage of maintaining engine performance if at any time fouling or other fault should reduce the efficiency of the turbochargers.

The authors had been particularly pleased to receive Mr. Volcy's contribution and to learn how closely their own thinking was allied to the work previously published by Messrs Bourceau and Volcy in 1966. The mounting arrangement adopted made the engine virtually independent of any longitudinal or transverse bending of the double bottom and, in conjunction with the special mounting chocks, freed the engine from longitudinal stresses due to differential expansion. It was very gratifying to have the authors' own opinion, that a lot of problems concerning the deterioration of main bearings and broken crankshafts could be attributed to the deformation of the ship's structure, endorsed by one of the major classification societies. They also thanked Mr. Volcy for his warning regarding the possibilities of vibratory problems—exercises were being carried out to investigate the natural frequencies of the combined engine and double bottom structure though no great difficulties were anticipated. The natural frequency of the engine, mounted on its resilient chocks on a theoretical rigid base was substantially above the excitation frequency. The authors hoped that with the advent of the Common Market they would have an early opportunity of working in close co-operation with Bureau Veritas.