Direct Drive Diesel Machinery

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This paper opens with a discussion on the advantages of the slow running, direct drive marine Diesel engine over other forms. It traces the increase in power produced, particularly in recent years. Some of the problems connected with turbocharging and crankshafts are considered in their relation to engines of the larger powers. The paper concludes with an outline description of the main and auxiliary machinery for a 53,000 d.w.t. tanker and some indication of future lines of development.

INTRODUCTION

Diesel propulsion of ships may be arranged either by a simple, direct coupling of a Diesel engine to a propeller shaft or by a more complicated, indirect coupling of, usually, more than one Diesel engine to a propeller shaft as in geared plants, Diesel-electric plants or Diesel gas plants with, for instance, free-piston engines and gas turbines.

The continued preference for low speed Diesel engines, direct-coupled to the propeller shafts, for almost all ordinary merchant ships, if no special circumstances have to be considered, is mainly due to the fact that these plants, in comparison with installations with indirect drive, offer the following essential advantages:

i) Simplicity
ii) Low fuel and lubricating oil consumption
iii) Utilization of the cheapest fuel oil grades
iv) Low noise level.

The first large, ocean-going motor ship, the *Selandia,* which was delivered 50 years ago, had a twin-screw plant, and up to about 1925 all large motor ships were twin-screw ships. The principal reason was, of course, that the necessary engine power could not be obtained in a single engine, but probably also—at least during the first years—the additional security for the vessel which the twin-screw installation afforded.

The number of single-screw motor ships with direct drive increased gradually and since 1940 only a small number of twin-screw ships have been built. In these cases special circumstances have not permitted the use of single-engine plants, for instance in ferry-boats and passenger vessels.

The reason for this development is, in the first place, the improved reliability and better facilities for attendance and maintenance of the engines. Further, that it has been possible to obtain steadily increasing outputs with Diesel engines, which during the past ten years has meant an increase from about 10,000 to about 25,000 b.h.p., whereby engine powers high enough for single-screw installations in even the largest cargo liners, bulk carriers, and tankers is now available.

The development towards the higher engine outputs has been characterized not only by the introduction of greater cylinder dimensions—with bores up to 840-900 mm. and strokes up to 1,800 mm.—but also by the increase of the mean pressure of the two-stroke engines, first in 1951-52 by turbocharging and then in 1959-60 by high pressure turbocharging, with a resultant reduction of the number of cylinders for a given engine output.

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ADVANTAGES OF DIRECT DRIVE DIESEL MACHINERY

The simplicity, which is one of the weightiest advantages of the direct drive Diesel machinery, has thus been substantially improved by engines with increased cylinder outputs, resulting in extended use of single-screw plants and a reduction in the number of cylinders.

The second and the third of the important advantages mentioned earlier, viz. the lower fuel and lubricating oil consumption and the possibility of using the cheapest fuel oil grades, are due to the fact that the engines for direct drive are low speed engines of the crosshead type.

The reduction of the fuel oil consumption from the 162-165 gr./b.h.p.-hr. for the non-turbocharged engines to the 156-160 gr./b.h.p.-hr. for the turbocharged and high pressure turbocharged engines has been achieved due to the following two factors:

First, the increase in the consumption per i.h.p.-hr. which is inevitable at higher mean indicated pressures has been limited in consequence of the fact that the charging air pressures, the compression pressures, and the maximum pressures have been increased in nearly the same proportion as the mean indicated pressures during the development from the non-turbocharged to the turbocharged and high pressure turbocharged engines. Furthermore, an increase of the amount of air per i.h.p.-hr. has been obtained by means of the simple and efficient varying-pressure Burmeister and Wain turbocharging system in the engines with uniflow scavenging.

Secondly, the mechanical efficiency has been considerably increased by the increased engine load and by the elimination of the direct-driven blowers. For the engines with uniflow scavenging and an efficient turbocharging system, the increase in the mechanical efficiency is particularly pronounced, as there are no direct-driven blowers and no under-piston pumping, which would reduce the efficiency.

The mechanical efficiency is about 90 per cent, corresponding to a fuel oil consumption of 141-145 gr./i.h.p.-hr. (0·311-0·320lb./i.h.p.-hr.) and 156-160 gr./b.h.p.-hr. (0·344-0·353lb./b.h.p.-hr.).

These consumption figures apply to light fuel; when using heavy fuel, an additional consumption of up to about 4 per cent—dependent on grade and calorific value—must be foreseen.

As crosshead engines are used for direct drive, the lubricating oil consumption is low. For engines with uniflow scavenging, the consumption of cylinder oil will only be about 0·2 gr./b.h.p.-hr., and assuming the same amount for system oil consumption—which by the way depends on how the machinery is maintained—the total lubricating oil consumption will be about 0·4 gr./b.h.p.-hr.

The afore-mentioned values mean that for direct drive Diesel machinery the fuel and lubricating oil economy is substantially better than for indirect drive, for which medium or high speed trunk-piston engines are used. Besides the loss in the power transmission from the Diesel engines to the propeller shaft, these engines will have a considerably higher lubricating oil consumption and appreciably reduced possibility of using the cheaper heavy fuels.

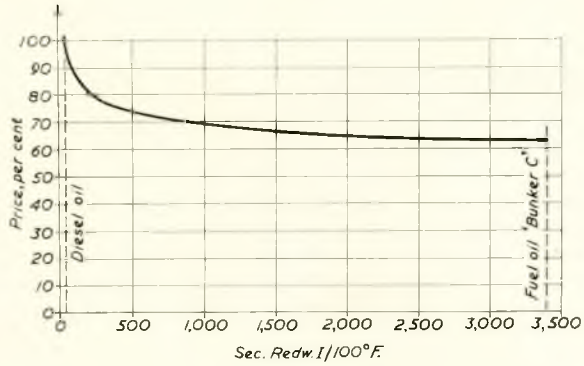


FIG. 1—*Fuel oil price in dependence of the viscosity*

Fig. 1 shows how the price of heavy fuel oil in proportion to that of Diesel oil varies with the viscosity.

Regarding the fourth of the important advantages of the direct drive Diesel machinery, the lower noise level, it may be mentioned that the mechanical noise from a modern two-stroke, low speed engine is of low frequency and normally below the permitted level.

The main sources of noise are the turboblower inlet and outlet, the exhaust from the cylinders, and the pressure variations in the fuel oil system.

Noise from the turboblower air-intake is of a high frequency and must therefore be kept at a low level; this is achieved by using large intake silencers in which the ducts are lined with felt and have adequate large surfaces.

Noise from the turboblower outlet, which is also of a high frequency, is kept at a low level by proper design of the blower diffusers and of the air ducts from the blowers to the air coolers and, by efficient insulation of these ducts.

Noise from the cylinder exhaust has to be reduced by insulating the exhaust pipes and, in the fuel oil system, shock absorbers are fitted to avoid or reduce the noise that may be produced by pressure variations in the system.

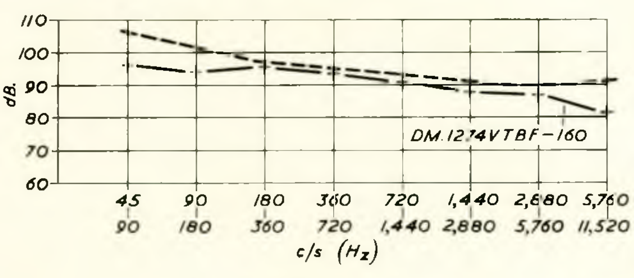


FIG. 2—*Noise level curve*

Fig. 2 shows a curve representing the noise level which is considered permissible in Scandinavia, and also the results of sound measurements on the manœuvring platform for a 15,000 b.h.p. engine fulfilling these requirements.

PROBLEMS CONNECTED WITH TURBOCHARGING AND CRANKSHAFTS

The introduction of high pressure turbocharging and the larger cylinder dimensions have called for special attention mainly to two questions, viz. the turbocharging system and the crankshafts.

In the development from the non-turbocharged to the turbocharged and further to the high-pressure turbocharged engines, it has been possible through the simple and efficient B. and W. turbocharging system with varying pressure, to maintain unaltered the temperatures of the cycle and the wall temperatures in the combustion space, and thereby to maintain the reliability of the engines.

In these engines with uniflow scavenging and with effective cooling of the air charge, the mean indicated pressures have been increased from 6·5 to 8·0 kg./sq. cm. (92 to 114lb./sq. in.) and further to 9·5 kg./sq. cm. (135lb./sq. in.), and the charging air pressures, compression pressures, and maximum pressures have, as already mentioned, been increased in nearly the same proportions, so that the temperatures have been kept unaltered, and the increase in fuel oil consumption per i.h.p.-hr. has been limited. For the high pressure turbocharged engines the following pressures apply:

Mean indicated pressure	9·5 kg./sq. cm.	135lb./sq. in.
Charging pressure ...	0·75 kg./sq. cm.	10·7lb./sq. in.
Compression pressure ...	53 kg./sq. cm.	755lb./sq. in.
Maximum pressure ...	65 kg./sq. cm.	925lb./sq. in.

It has furthermore been possible to increase the amount of air per i.h.p.-hr. and the increased amount of air is mainly used for scavenging, thereby improving the already efficient scavenging and the loadability of the uniflow-scavenged engine. The total air quantity is about 7·2 kg. (15·9lb.)/i.h.p.-hr. at an m.i.p. of 9·5 kg./sq. cm. (135lb./sq. in.) and the air quantity trapped in the cylinder about 4·65 kg. (10·3lb.)/i.h.p.-hr.

Since the wall temperatures have been kept unchanged during the three stages in the development of the engines, it has been possible to continue using oil cooling for the pistons.

During the past year the question of reverting to water cooling of the pistons has been discussed, and water cooling has been mentioned as a necessity for cross-scavenged engines with large cylinder bores and with high pressure turbocharging.

It should therefore be mentioned that, for the uniflow-scavenged engine with efficient turbocharging, the reduction of the wall temperatures of the pistons obtainable by water cooling is so small—about 25-30 deg. C. (77-86 deg. F.)—that a change to the more complicated water cooling of the pistons is not necessary so far as the durability of the pistons in these engines is concerned. The wall temperatures of the pistons, which near the bottom position are not exposed to hot exhaust gases but only to cold scavenging air, can be maintained without difficulty—by oil cooling as well as water-cooling—at the same or a lower level than that obtained by water cooling in cross-scavenged engines.

In this connexion it should further be emphasized that, in the uniflow-scavenged engines with efficient turbocharging, the inside piston temperature, and thus the conditions for the

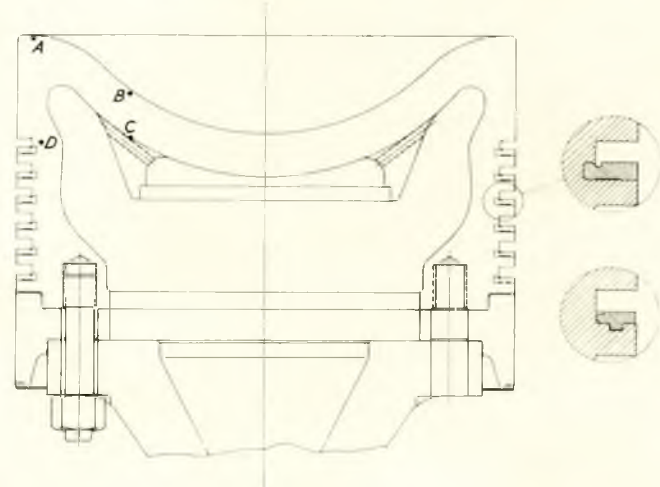


FIG. 3—*Piston for a high pressure turbocharged engine*

cooling oil, will not only be independent of the degree of turbocharging but also of the cylinder bore, the inside wall temperature of the pistons being lower in the larger engines, which have the greater wall thicknesses.

Fig. 3 shows the piston for a high pressure turbocharged engine.

The temperatures at points A, B, C, and D will be:

	A	B	C	D
Oil cooling, deg. C.	315	375	160	100
Water cooling, deg. C.		345	100	

The first line gives the temperatures measured in a piston for an engine with a 740-mm. cylinder liner bore, a mean indicated pressure of 9·5 kg./sq. cm. and oil cooling.

The second line shows the temperatures at points B and C, when using water cooling, and corresponding to the temperatures measured in an engine with an 840-mm. cylinder bore.

Pistons of this type, made of CrMo cast steel, have now been in service for several years with good results in the turbocharged engines with a mean indicated pressure of 8 kg./sq. cm. (114lb./sq. in.) and the same wall temperatures. Crack formation has never been a problem in these engines and, with the piston design shown, the burning away of material from the surface of the piston crown, which was a drawback of the piston design used for some years in an intermediate period, is now of no importance.

The pistons are, as shown, provided with carrier rings of cast iron in the piston ring grooves. These carrier rings can be either caulked-in or fitted loosely. Caulked-in rings have been used with good results in earlier piston types. The loosely fitted rings require great care in the manufacture of the rings as well as ring grooves, but the initial troubles being over now, these rings seem to give satisfactory results, and they have the great advantage that they can be renewed on board, provided the ring grooves are well cleaned.

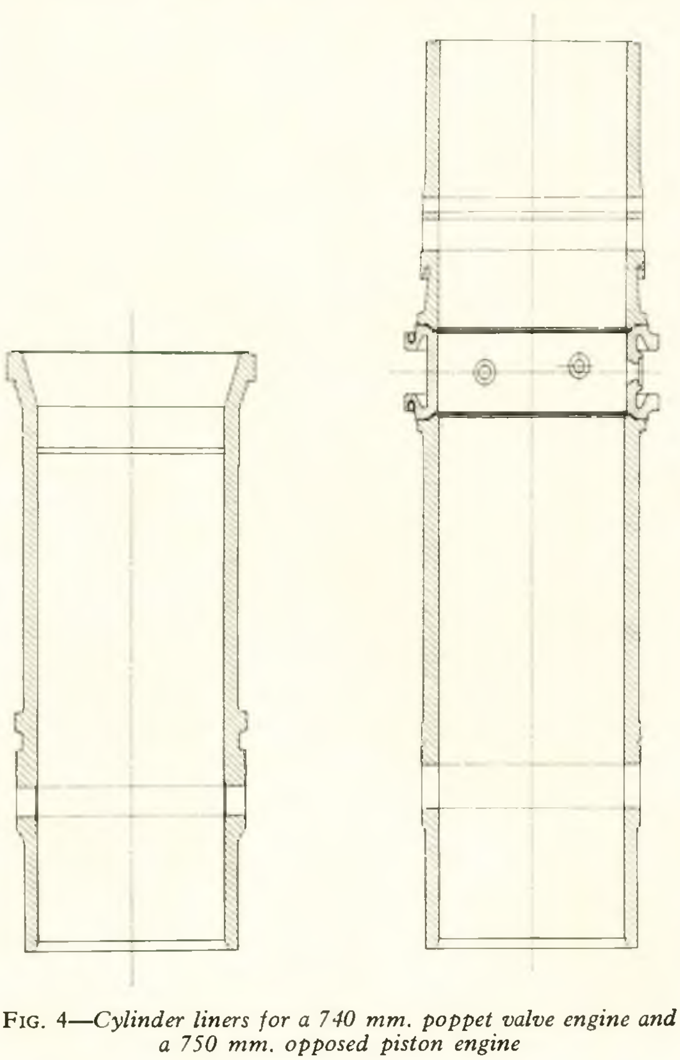


Fig. 4—*Cylinder liners for a 740 mm. poppet valve engine and a 750 mm. opposed piston engine*

Fig. 4 shows cylinder liners for a 740-mm. poppet valve engine and a 750-mm. opposed piston engine.

The cylinder liners for the poppet valve engines are of the simplest possible design and made of vanadium cast iron which, in conjunction with the neutralizing cylinder oils, generally gives satisfactory wear results of about 0·1 mm. or less per 1,000 hours, even when heavy fuel oils are being used. The bigger cylinder liners are provided with six points of lubrication, and the cylinder lubrication is timed. Chromium-plated liners are only used when specially requested by owners and, if so, mainly with a chromium layer of 1·8 mm. thickness in the upper part of the liner, which gives a better possibility of obtaining an economical advantage than with a chromium layer of an even thickness of about 0·4 mm., which was previously used.

For the opposed piston engines, tripartite cylinder liners are being used; the liners for the main piston and the exhaust piston are both of similar design to the liners for the poppet valve engines. The intermediate parts, with the bosses for the valves, are of cast steel.

The spindle seats, for the exhaust valve in the poppet valve engines, are faced, by welding on stellite, to increase the resistance to wear and the chemical action of heavy fuel and the spindles have two separate guide bushings, well apart, which maintain good alignment of the spindles and thereby good durability of the seats.

The distances between cylinder centres and the dimensions of the crankshafts have been determined in consideration of the demands on shafting diameters, widths of webs, shrink fits and surface pressures in bearings. In addition, the firing orders have been chosen so that satisfactory conditions as to turbocharging arrangement, balancing, torsional and axial vibration are obtained.

The crankshafts are dimensioned for 20 per cent overload, corresponding to a mean indicated pressure of 10·6 kg./sq. cm. (151lb./sq. in.).

In consideration of the bearing loads, however, the journal diameters have been increased beyond these requirements, so that, for poppet valve engines and opposed piston engines respectively, distances between cylinder centres of about 1·85 and about 2·50 times the cylinder diameter have been obtained.

The engines being long-stroke engines, with a stroke of about 2·2 times the cylinder diameter, the crankshafts can be fully built for any number of cylinders, if the torsional vibration conditions are suitable, so that for these numbers of cylinders, the design considered preferable (fully-built or semi-built) as regards production facilities, costs, etc., may be chosen.

The minimum thickness of the material between the bores in the web for the fully-built shafts are 27 per cent of the shrinkage diameters and the minimum limit for the lower yield stress is fixed at 24 kg./sq. cm. (15·2 tons/sq. in.), which gives ample security against slipping in the shrinkages.

With reference to the torsional vibration, it should be mentioned that with the increased mean indicated pressure for the high pressure turbocharged engine, the harmonic torques are also increased but, due to the extraordinary increase of the shaft diameter, the extra stresses from torsional vibration, with two nodes, have not been increased.

For axial vibration in the shafting system the natural frequency is chiefly determined by the mass and rigidity of the crankshaft and the rigidity of the thrust bearing, while the shaftline and the propeller normally have a minor influence. The rigidity of the crankshaft depends on the web dimensions, firing order and main bearing clearances.

Torsional vibrations with one and two nodes, having resonance points too close to each other should be avoided,

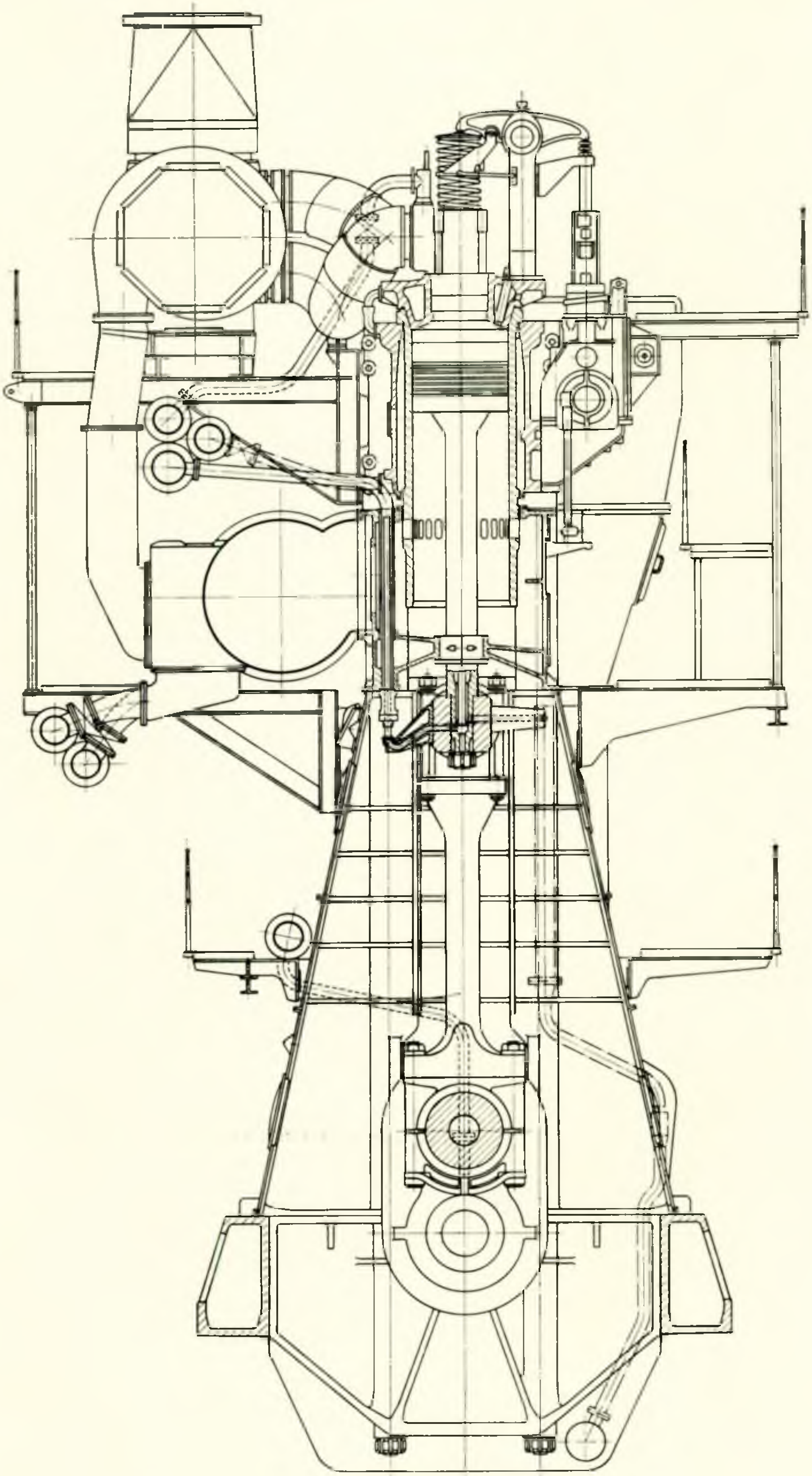


FIG. 5—*High pressure turbocharged poppet valve engine*

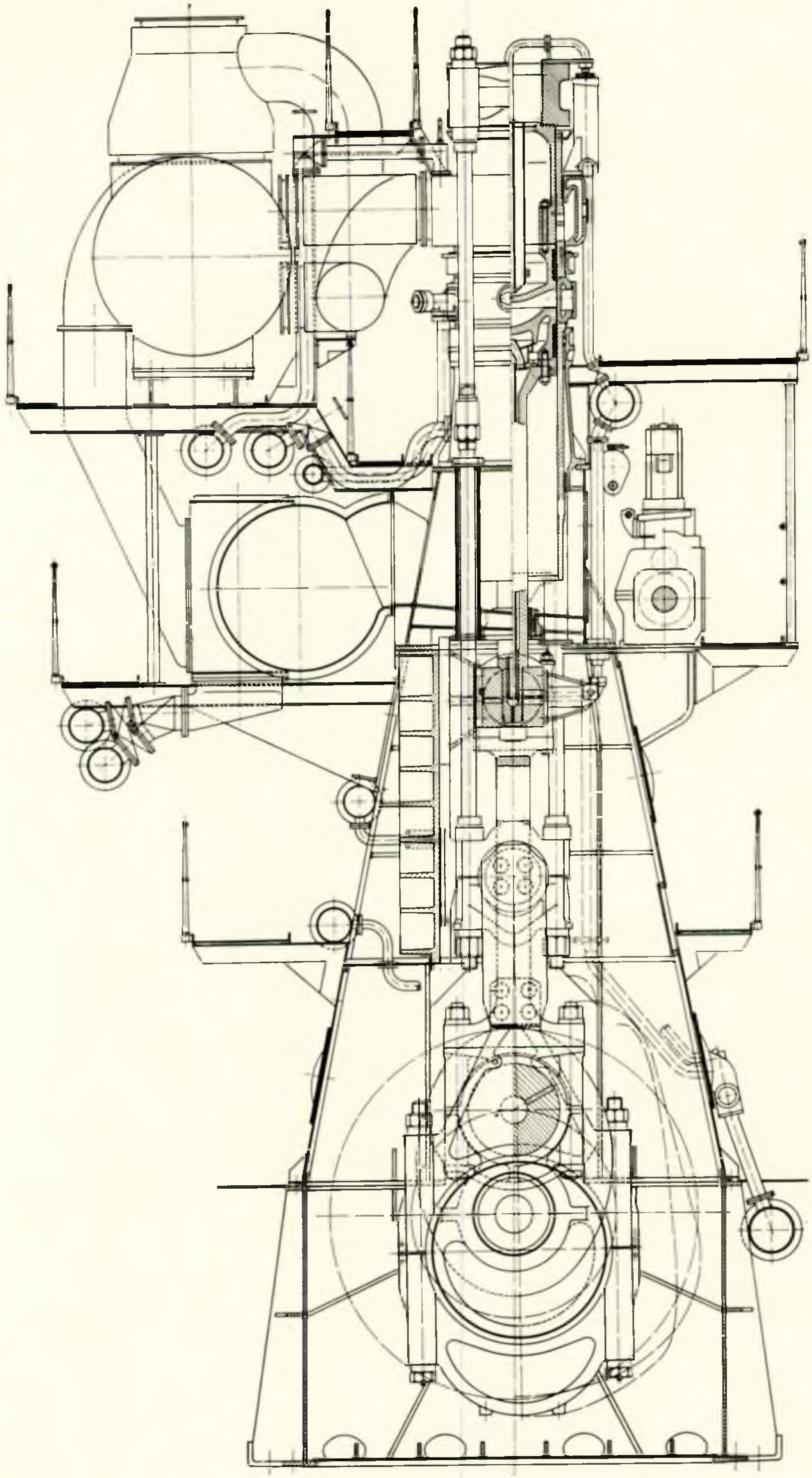


FIG. 6—*High pressure turbocharged opposed piston engine*

at least at higher revolutions, where the torques are large, and the criticals for the axial vibration should be well away from the criticals for the torsional vibration in order to avoid the axial deflexions, at or near the torsional resonance, being amplified by the axial resonance.

Figs. 5 and 6 show examples of slow running crosshead engines.

Fig. 5 shows a high pressure turbocharged poppet valve engine with a cylinder diameter of 840 mm., stroke 1,800 mm. and an output in continuous service of 2,100 b.h.p. per cylinder at 110 r.p.m. and an m.i.p. of 9·5 kg./sq. cm. (135lb./sq. in.). The engines are built with up to 12 cylinders, with an output in continuous service of 25,200 b.h.p. The weight of a 10-cylinder welded engine with an output of 21,000 b.h.p. is 815 tons or about 39 kg. (85lb.)/b.h.p.

The smaller engines of the same type yield 1,500, 1,100, 700 and 475 b.h.p. per cylinder respectively, thus together covering the outputs from 2,400 b.h.p. to 25,000 b.h.p.

Fig. 6 shows a high pressure turbocharged opposed piston engine with a cylinder diameter of 750 mm., the total stroke 2,300 mm. and the output in continuous service 2,100 b.h.p.

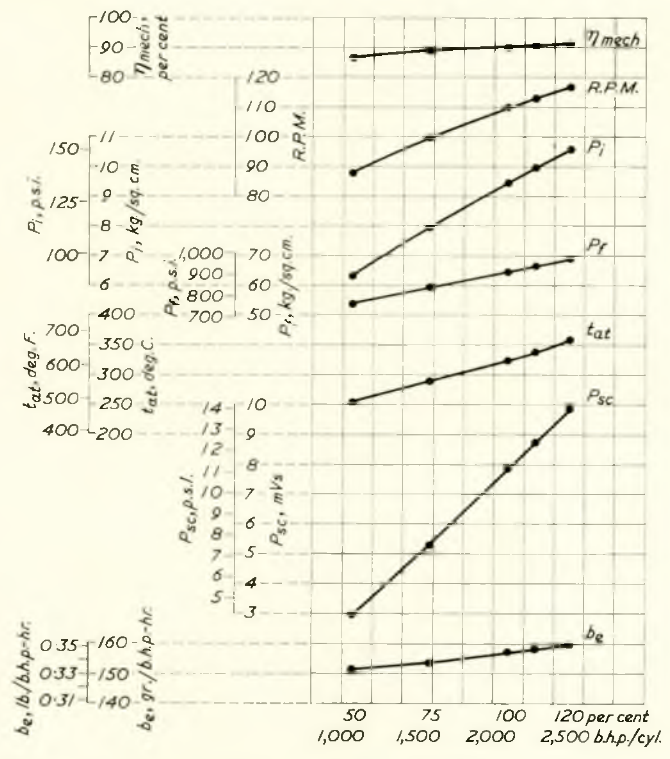


FIG. 7—*Test results from an engine with 840 mm. cylinder bore and 1,800 mm. stroke at about 50-75-100-110-120 per cent load*

per cylinder at 110 r.p.m. and an m.i.p. of 9·5 kg./sq. cm. (135lb./sq. in.). The engines can be built with up to 10 cylinders with an output for continuous service of 21,000 b.h.p.

The weight of this engine is about 860 tons or about 41 kg. (90lb.)/b.h.p.

Fig. 7 shows test results from one of the engines with 840 mm. cylinder bore and 1,800 mm. stroke at about 50-75-100-110-120 per cent load.

Together with a smaller engine of the same type the outputs from 6,250 to 21,000 b.h.p. will be covered.

A MODERN DIRECT DRIVE DIESEL MACHINERY INSTALLATION

As an example of modern direct drive Diesel machinery, Fig. 8 shows the engine room arrangement for a tanker of 53,000 d.w.t., with a trial speed of 17·3 knots.

The main engine is a 10-cylinder, single-acting, high pressure, turbocharged, poppet valve, two-stroke engine of welded design, direct-coupled to the propeller and arranged for burning boiler oil. The cylinder bore is 840 mm. and the stroke is 1,800 mm. In daily service the engine develops 21,000 b.h.p. at 110 r.p.m. and can run at ten per cent overload without a time limit. For shorter periods, the engine may be overloaded by 20 per cent.

The auxiliary machinery is electrically driven except for the shaft-driven lubricating oil pump and the hydraulically-driven cooling water pumps.

The lubricating oil pump is a screw pump driven from the intermediate shaft. The pump delivers the oil at pressures of 8-10 kg./sq. cm. (115-140lb./sq. in.) to an oil motor and leaves the oil motor at the normal lubricating oil pressure. The oil motor actuates two centrifugal pumps for cooling water, one for salt water and one for fresh water. With this arrangement, a turbo-generator, driven by the steam from the exhaust-gas boiler, can produce the necessary power for the requirements at sea.

The exhaust-gas boiler has a steam production rate of about 7,000 kg./hr. (15,500lb.), and, after deduction of 1,700 kg./hr. (3,750lb.) for other purposes, there are 5,300 kg./hr. (11,700lb.) available for the turbo-generator. At a vacuum of 90 per cent and a specific steam consumption of about 9 kg./kW hr. (20lb.) the turbo-generator can develop about 530 kW at full load of the main engine.

The installation includes two oil-fired watertube boilers, each capable of giving 26,880lb./hr. of steam. The boilers have two separate steam systems; a primary, closed system with pure water and a secondary system which produces the steam for consumption. This type of boiler is easy to operate and provides ample security against fouling of the evaporator pipes directly exposed to the fire. They are thus suitable for Diesel ships in preference to the more vulnerable single-stage boilers.

To keep a cargo oil temperature of 100 deg. F., at an ambient temperature of 40 deg. F., a steam supply of about 30,240lb./hr. for the heating coils in the cargo oil tanks is required. For other purposes about 5,600lb./hr. are required, i.e. in all 35,840lb./hr.

For tank cleaning, using three Butterworth units, each of 45 tons of sea water per hour capacity, at a temperature of 75 deg. C., 45,920lb./hr. of steam are required. Thus, the two oil-fired boilers, each with a capacity of 26,880lb./hr., cover these two requirements.

There are three cargo pumps, with a total capacity of 5,000 cu. m./hr. and a pressure of 10·5 kg./sq. cm. (149lb./sq. in.).

Two of the pumps are driven by steam turbines. The steam is slightly superheated at a pressure of 12·5 kg./sq. cm. (178lb./sq. in. gauge) and the vacuum is 66 per cent. The steam consumption for each pump is 19,040lb./hr. which together with the consumption for other purposes makes a total steam consumption of about 47,040lb./hr.

The third cargo pump is electrically driven and has a power consumption of 700 kW.

The maximum power consumption at quay will be about 1,140 kW, the consumption for other purposes being about 440 kW.

Normal at sea	280 kW
Maximum at sea	490 kW
With main engine manœuvring, normal	710 kW
maximum	870 kW
Whilst pumping cargo	1,140 kW

The installation comprises two Diesel generators, each of 670 kW, and a turbo-generator of 600 kW.

The power consumption will be covered by either one or two generators, so that at least one generator is on standby.

The total weight of the plant is 1,540 tons, distributed as follows:

	Tons
Main engine	815
Auxiliary machinery for main engine ...	104
Diesel—and turbo-generators	69
Boiler plant, inclusive of water	154

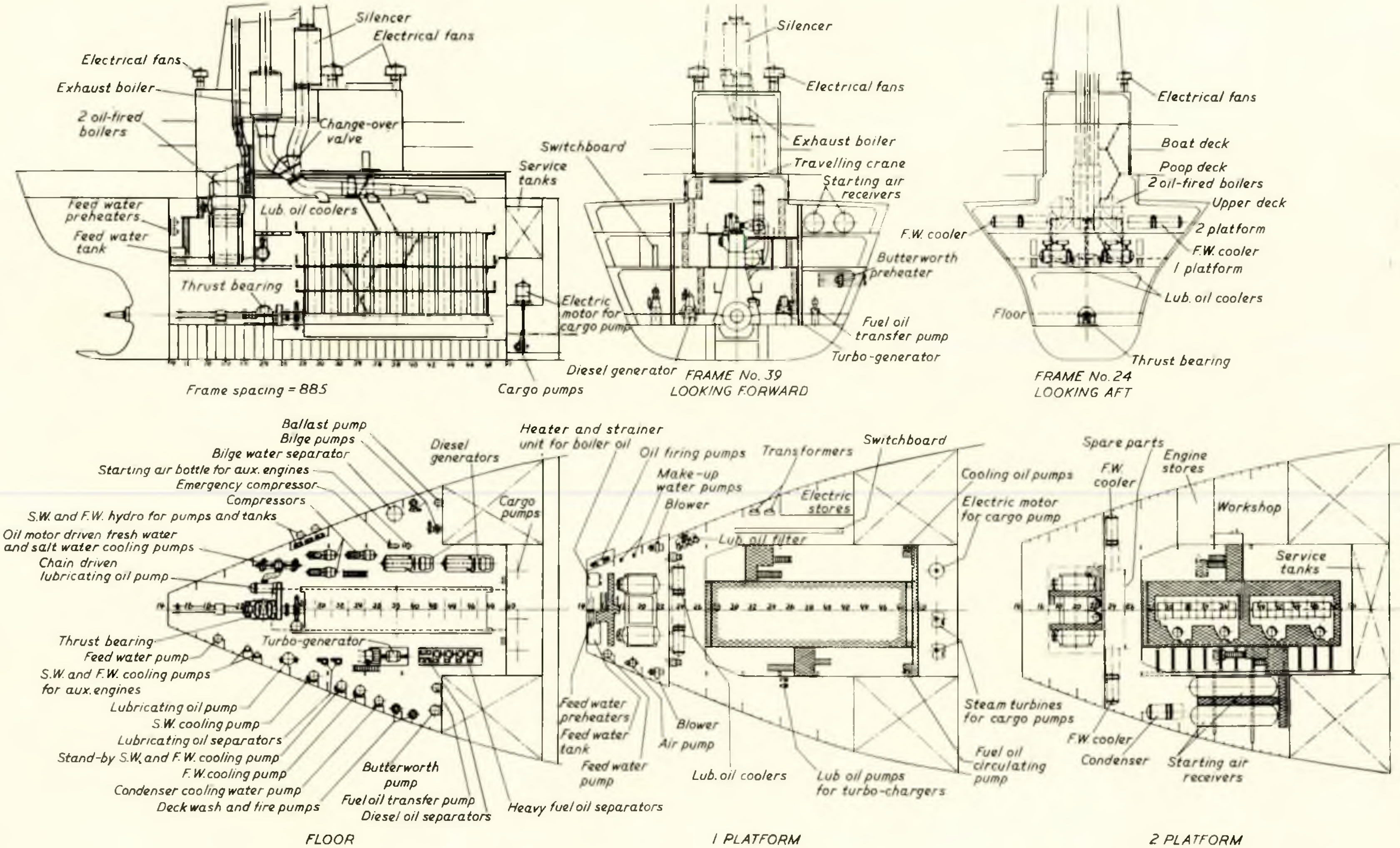


FIG. 8—*Engine room arrangement for a 53,000 d.w.t. tanker*

Cargo pumps	22
Shafting and propeller	71
Pipes in engine room	176
Remaining machinery, inclusive of spare parts	75
Water and lubricating oil	54
Total weight	1,540

CONCLUSION

In considering the possibilities for further development of the existing engines it should be pointed out that the mean indicated pressure of 9·5 kg./sq. cm. (135lb./sq. in.) for normal continuous service is very moderate and that in addition the engines can be overloaded by 10 per cent, without time limit, and by 20 per cent for shorter periods.

A possible increase of load should, however, only be considered if charging air pressure, compression pressure, and maximum pressure are increased at least proportionally, to maintain the present thermal load, and to keep the fuel consumption at approximately the same level as now. The engines must therefore be built for continuous running at the high maximum pressures.

Automation and remote control are questions in which a growing interest will be taken, in the future, due to the increasing difficulty in procuring qualified crew on board the ships.

Automation will include a greater or smaller part of the auxiliary machinery—automatic start and stop and automatic adjustment of pressure and temperatures.

For the main engines it is a matter of manœuvring and adjustment control from a special cabin either inside or outside the engine room and possibly from the bridge. Such plants are already in service.

For the main engines the most important will, however, without doubt be a centralized, remote control, preferably with visual recording. For turbocharged engines there are special circumstances, e.g. some of the most important service values vary with engine speed and others with output. The control system for the service conditions should therefore be such that, irrespective of the actual engine output, it can be seen at a glance whether the most important service values differ from the normal.

Discussion

MR. H. N. PEMBERTON (Member) said that the engines described in the two papers were representative of the intensive developments in recent years by nearly all builders of large marine oil engines, which had led to the present position when an output of some 30,000 b.h.p. was possible from one slow running single acting two-stroke engine. This, of course, was the result of improvements in scavenging and pressure charging by highly efficient free-running turbochargers. Both Sulzer and Fiat had obtained 3,000 b.h.p. per cylinder from engines during periods of sustained overload on the test bed. The time was obviously approaching when Diesel engines would be capable of providing any power appropriate to a single shaft for a merchant ship. Table I in Mr. Jackson's paper was of considerable interest and showed that Doxford's were abreast of their Continental competitors.

The economics of Diesel propulsion were governed by capital cost of machinery, fuel costs and the cost of maintenance and repair. Since the thermal efficiency of the modern Diesel engine was so near the optimum, little, if any, further gain could be expected in specific fuel consumption. Reduction of fuel costs depended, therefore, on the burning of a fuel which was cheaper than Diesel oil, and at present there was a decided advantage in burning heavy oil, such as Bunker C, in Diesel engines. This aspect was dealt with by both Mr. Andresen and Mr. Jackson, both of whom had given a lot of attention to the overcoming of some of the serious problems which arose when this type of oil was first used. Today, according to the records at Lloyd's Register, at least 90 per cent of all oil engines above 5,000 b.h.p. being constructed throughout the world were fitted to burn heavy oil.

Increase of power per cylinder meant, of course, fewer cylinders and a reduction in the number of working parts, which could be expected to result in a reduction of maintenance costs. However, it must be borne in mind that the loading of components was increased and therefore design detail became of greater significance. It was interesting to conjecture whether any further advances could be expected in the pressure charging of these large engines, bearing in mind, for example, that the very heavy crankshaft parts might present difficulties in forges and foundries. Whilst it could be expected that these large slow-running oil engines would be successful in service, it would, he suggested, be premature to contemplate any further advance at present.

Having in mind the serious problem which so many shipowners faced today in the manning of their engine rooms, Mr. Pemberton referred to one item from each of the papers which had a bearing on this problem. Mr. Jackson described the fuel system for heavy fuels which he had adopted for the Doxford engine and emphasized the need for careful maintenance and cleanliness of the centrifuges used in the system. He stated that for an engine burning more than 30 tons of fuel per day an extra junior engineer was required to maintain these centrifuges. Mr. Pemberton suggested that this was an unnecessary waste of talent, and that these duties, together with many other watchkeeping duties in a modern engine room, could be carried out by semi-skilled men rather than by trained engineers. Qualities of character and integrity, and a certain manual efficiency, were the primary needs for such duties, and the employment of men with these qualities in engine rooms might go a long way to solving the problem associated with the shortage of trained marine engineers.

Mr. Andresen, in referring to this problem, stated that automation and remote control must figure prominently in future development of oil engine machinery.

Mr. Pemberton said that in his view insufficient attention was being given to these modern control systems in merchant ship machinery, and he suggested that some research and development effort might well be directed to these features.

MR. M. YAMAGUCHI congratulated Mr. Jackson on his success in designing perhaps one of the most advanced large power Diesel engines in the world, and looked forward to seeing its appearance on the market in the near future.

Mr. Yamaguchi said that he would like to hear the author's opinion on the following points:

1) If the fuel was heated to a temperature by the heater there remained the problem of how to maintain the temperature of the fuel at the desired figure at the point of the fuel nozzle.

2) As present-day low grade oil included volatile ingredients to meet viscosity requirements, the danger of vapour locking of heated oil at the pump had to be considered. To avoid this drawback another supply pump was likely to be necessary.

3) Keeping the outside wall of the cylinder warm by maintaining the cooling water at a high temperature, led to a high temperature of the inner wall, i.e. the combustion chamber side wall, and thus to clogging of the cylinder oil. Therefore, in his company's practice, the temperature of the cooling water was maintained at 45 deg. C. at inlet, 55 deg. C. at outlet, and inner wall temperature at under 150 deg. C.

Mr. Yamaguchi then asked the following questions:

1) The fuel injection pressure of 7,000 to 8,000lb./sq. in. was somewhat lower than in Japanese practice in which a somewhat similar fuel system was employed. Was this pressure chosen to secure the most favourable combustion condition or by the limitation of mechanical strength of the components?

2) Concerning corrosion of crosshead pins and crankshafts, what was the amount of allowable oxidation of lubricating oil and how was the oil purified?

3) Advancing the lead of opening of the exhaust ports was stated to be advantageous in getting the engine to run at lower r.p.m. What were its merits in preventing the blowback of gas through scavenging ports?

Mr. Yamaguchi then dealt with Mr. Andresen's paper under four headings:

1) Mr. Andresen had shown an interesting relation between fuel oil price and its viscosity. But what his own company was now faced with was the fact that so-called "heavy" oil was often of high density, including poorly combustible components, though the mean viscosity was under, say, 3,400 seconds. How was this trouble treated in the author's case?

2) Mr. Andresen had stated that chromium plated liners were used if owners preferred them. What was his opinion as to the economical advantage of this method, and were there any data concerning the amount of wear per 1,000 hours?

3) The statement that the "critical" for the axial vibration should be well away from that for the torsional vibration in order to avoid the axial deflexion was of great interest, and the designers in Mr. Yamaguchi's company would give consideration to this.
4) There was no comment in either of the papers as to the use of controllable pitch propellers in conjunction with large powered Diesel engines. Since the use of such propellers should produce some advantages in simplification of the engine and control systems, it would be interesting to have the authors' opinions on this.

Mr. C. C. Pounder (Member) said that he did not propose to ply Mr. Jackson with questions but rather to answer one which he had recently enunciated on a slip of paper passed quietly across a table, in the course of a tedious research committee meeting, while other men discoursed verbosely upon extraneous matters. Mr. Jackson's question was: "Is it likely to be the policy of Harland and Wolff to forsake opposed piston engines for poppet valve engines?" To this question the speaker gave a categorical negative reply.

Ten or twelve years ago, when Burmeisters were developing their single-acting two-stroke poppet valve engines rather intensively, Mr. Pounder had sought the interest of all his company's superintendent engineer clients, in the expectation that a substantial number of them would favour the engine. But nobody wanted that type of engine. On the contrary, sentiment ran strongly in favour of the opposed piston engine. And so, in Belfast, he and his colleagues continued with the development of their characteristic form of opposed piston engine driven by eccentrics. It was said at that time—not by Mr. Andresen but by other Copenhagen engineers—that for engines of the size being developed the operation of exhaust pistons by eccentrics would prove impossible. Experience had shown the critics to be wrong. Harland and Wolff's policy was to continue with the development of the opposed piston engine, but more intensively than before.

Harland and Wolff, as a great firm with many clients, had always built two different types of engine, at least. Recently a number of clients—Scandinavian shipowners, be it said—had ordered poppet valve engines from them. And so they built poppet valve engines "cheek by jowl" with opposed piston engines.

Recently, in Belfast, they had had on trial a six-cylinder engine of 620 mm. bore, and clients had been agreeably surprised at the relative quietness of it. It shook a little, but not much. On their order book they had the largest single-screw Diesel engine yet to be built in the United Kingdom, a ten-cylinder unit 840 mm. bore/1,800 mm. stroke. The engine should be on test by the end of the present year.

Regarding the paper by Mr. Andresen: he suggested that the audience might like to know why it was necessary to tie the poppet valve engine to the decks of the ship. Thus, for the engine he had mentioned, transverse ties were being arranged at each cylinder. Only on two occasions had Mr. Pounder been faced with the need of tying an engine to the deck beams. That was thirty years ago, the engines being long-stroke four-cycle engines. For a twin-screw passenger ship tied engines could engender a number of difficulties.

Mr. Pounder concluded his remarks by expressing his intense regret that his old friend Andresen was leaving Burmeisters. He drew a parallel between Andresen the Dane and Hamlet the Dane.

Professor G. Aertssen said that in 1926 the Port Authority of Antwerp, who owned practically all the small floating craft in the harbour, decided to use only Diesel engines for all future designs. Then about 25 years ago, at an International Conference of Naval Architects and Marine Engineers which was held in Liège, the late Mr. A. C. Hardy had produced a note on the position of the Diesel engine on coasters, fishing craft and inland waterway ships. Speaking of the position of Holland and Belgium, he had said: "It should be placed on record that both Belgium and Holland were, perhaps naturally, the home of oil-engined small craft". And further: "One has only to make a cursory visit to ports like Antwerp to see the internal combustion engine vigorously and intelligently applied to tugs, barges, ferries, dredgers and the like".

It had been the policy of Belgian shipping companies for more than 30 years to have Diesel engines installed in their ships both for propulsion and also auxiliaries. Only in such cases as, for instance, where—with the sudden development of supertankers—Diesel builders were not in a position to satisfy the demand for high power engines, were steam turbines installed in the Belgian merchant marine. This explained how much they in Belgium followed with kindred feeling the great endeavour of important Diesel builders like Doxford and Burmeister to develop their engines in size as well as efficiency.

For more than ten years CEBERENA, the Belgian Naval Research Association, had carried out service-performance and seakeeping trials on merchant ships, and being concerned personally with these trials he had witnessed the growth of the large Diesel engine. The two striking stages in this research were the trials on the *Lubumbashi* in 1953-54, the first Belgian 10,000 ton cargo liner, driven by a heavy fuel burning Diesel engine, and the trials on a sister ship, the *Lukuga,* in 1960, driven by a heavy fuel burning supercharged Diesel engine. Both engines were Burmeister-license, Cockerill built.

In three papers* read before the Royal Institution of Naval Architects in conjunction with this Institute the performances of the *Lubumbashi* were related and analysed for hull, propulsion and machinery. The main engines were rated at 6,000 h.p., or 1,000 h.p. per cylinder, double acting two-stroke. The cylinder liner wear over 1,000 hours reduced after three years from 0·01in. to 0·007in., which were practically the highest figures given in Mr. Jackson's paper for the heavy fuel burning supercharged 10,000 h.p. engine of the *Montana*. The fuel rate of the *Lubumbashi,* after three years' service, was 0·42lb./s.h.p. hr. This figure was to be compared with the fuel rate of 0·34lb./s.h.p. given for the 10,000 h.p. Doxford P type engine. It must be said that on the cargo liner *Lukuga* the single-acting heavy fuel burning supercharged six-cylinder 7,000 s.h.p. engine had a fuel rate after three years' service of 0·36lb./s.h.p. The cylinder liner wear after the three years' service was 0·0045in./1,000 hr. These figures were comparable with the figures of the P type engine.

In connexion with the fuel rate of 0·34lb./s.h.p. given in Mr. Jackson's paper, he wished to ask the author if that figure was obtained at sea and how it was measured.

The remarkable feature of these supercharged engines was their high mechanical efficiency. On the *Lukuga* a mechanical efficiency of 0·91 per cent was obtained after three years' service, and this figure was again comparable with the high efficiency of the P type engine. This high mechanical efficiency related to the low cylinder oil consumption of 13 gal./day for the *Lukuga,* while the consumption on the double acting engine of the *Lubumbashi* was 24 gal./day. The mechanical efficiency of this engine was 80 per cent.

These figures were given to show the progress in marine Diesel engines in recent years. A fuel rate of 0·34lb./s.h.p. meant that 40 per cent of the heat value of the high viscosity fuel was transmitted to the screw, and he congratulated Mr. Jackson and Mr. Andresen not only on their excellent paper but even more on this splendid achievement.

Mr. J. F. Alcock said that the importance of thermal problems in these large supercharged engines was well known. In dealing with these the first thing essential to know was the heat flow, just as in mechanical design one started with the gas and inertia loads.

* Aertssen, G. 1955. "Sea Trials on a 9,500-ton Deadweight Motor Cargo Liner". Trans.I.Mar.E., Vol. 67, p. 395.
Aertssen, G. 1957. "Further Sea Trials on the *Lubumbashi*". Trans.I.Mar.E., Vol. 69, p. 411.
Aertssen, G. 1961. "New Sea Trials on the Sandblasted *Lubumbashi*". Trans.I.Mar.E., Vol. 73, p.1.

The company he represented had for some time been mapping the local heat fluxes in small high speed Diesels and had worked out some rough design rules. They were now trying to do the same for large marine two-stroke engines but were hampered by lack of data from modern engines.

The classic work of Nusselt and Eichelberg was done on air-blast injection engines; in these the air blast produced violent turbulence, which would affect the heat transfer at T.D.C. when the gas temperature and density were highest. Modern solid-injection engines might have quite different heat-flow characteristics; some certainly did.

He had therefore pounced on Mr. Andresen's Fig. 3 and the table on page 477, which gave some piston-crown temperatures. From those, at points B and C, it was possible, if the thermal conductivity were known, to calculate roughly the heat flux. He said "roughly" because local changes in section might distort the pattern. The metal was stated to be Cr Mo cast steel. If this were a low alloy, say about 3 per cent chromium, its conductivity would be about 0·1 c.g.s., and he was sticking his neck out by using this figure in the hope that, if he were wrong, Mr. Andresen would correct him.

On this assumption the heat flux around point C came out at about 31,000lb. calories/(sq. ft. × hr.). He had made similar wild guesses from piston temperature figures recently published for four other engines. Their bores ranged from 750-900 mm.; all were turbo-blown and running at or near their rated loads and speeds. The fluxes so calculated all came within the range 31,000 to 41,000 C.h.u./(sq. ft. × hr.). The higher figure, however, was a local value at a known hot spot; if this were excluded, the figures, including that for Mr. Andresen's engine, fell in the remarkably narrow range 31,000-34,000 C.h.u./(sq. ft. × hr.) or 151,000-166,000 kg. cal./sq. m.-hr. (more recent data shows a wider scatter). This led him to hope that his guesses had not been too wild.

In all his company's experience on small engines the local heat fluxes were roughly a function of the fuelling rate, i.e. the fuel burnt per unit piston area per hr. For all the engines concerned the fuelling rate was about the same, 0·73-0·79lb./(sq. in. × hr.) or 50-56 gr./sq. cm.-hr.

If the temperature of the piston coolant were known the thermal conductance between piston and oil could also be calculated. Again endangering his neck, he had taken this as 70 deg. C.; this gave conductance figures of 240-290 C.h.u./(hr. × sq. ft. × deg. C.), or 1,170-1,420 kg. cal.sq. cm. hr.-deg. C. The high value might be due to his point C being in a passage where the oil velocity was high.

He wished to repeat that these figures were very rough; there might easily be errors of 20 per cent or more either way. There was, however, some confirmation from tests reported by his colleague, Mr. Millington, on a Doxford single-cylinder unit*. Here heat flows were measured in the cylinder liner; there was none of Mr. Alcock's guesswork, since all relevant data, including the metal conductivity, were measured. Mr. Millington's flux figure at the injector level, where there was full exposure to combustion, came to 36,000 C.h.u./sq. ft.-hr. at a fuelling rate of 0·72lb./(sq. in. × hr.). This was rather higher than the piston heat fluxes quoted above, probably because the liner was much cooler than the pistons and therefore picked up more heat.

On page 476 Mr. Andresen gave the amount of air trapped in the cylinder. How did he measure this? From Mr. Alcock's own experience he knew that it was a very difficult task.

Rear-Admiral T. B. Bose, B.Sc. (Vice-President for India) said that his country was still behind many others as far as manufacture of large Diesel engines was concerned but they were contemplating setting up a factory for the manufacture of marine Diesel engines along with other auxiliaries. With regard to Doxford engines, as far as he knew, auxiliaries were not made by Doxfords, but he wondered if Mr. Jackson could tell him what was the smallest engine, with respect to horsepower, that Doxford had ever made and if they had made engines for generators, etc. Shipowners had to take various factors into consideration when deciding on the installation of machinery. He also enquired if Mr. Jackson had ever experienced trouble with vibration in Doxford engines and, if so, in what type of engine, as it had been known that in some engines, for example those with an odd number of cylinders, there had been cases where excessive vibration trouble had been experienced under certain circumstances.

With regard to Burmeister and Wain engines, he had no question to put to Mr. Andresen, as a few months previously there had been a similar conference in Bombay at which a paper on Burmeister and Wain Diesel engines had been read, followed by an interesting discussion.

Admiral Bose complimented both the authors on their excellent papers.

Dr. T. W. F. Brown, C.B.E., S.M. (Member), referring to Mr. Jackson's paper, said that he was on the author's side, but considered that at the beginning he made far too great a claim when he stated that it was really since the end of the Second World War that the Diesel engine had established its supremacy over the steam engine and steam turbine for powers up to 20,000 h.p. During 1960 the actual figures for world launchings gave the following data:

	Total h.p.	
Power range	*Diesel*	*Steam turbine*
10,000-20,000 h.p.	750,950	1,472,930
14,000-20,000 h.p.	168,000	1,106,370

This was hardly supremacy. Admittedly, the position in 1962 would be more favourable to the Diesel, but it was still true that there was a long way to go before supremacy could be claimed. Again, little was gained in exaggerating fuel consumption of steam turbines in comparison with oil engines. It was suggested that comparative figures were:

Oil engines	0·36 to 0·37lb./b.h.p. hr.	all
Steam turbines	0·50 to 0·52lb./s.h.p. hr.	purposes

the steam turbines being able to utilize the lowest grade of residual fuel. Even without relation to the cost of fuel per ton, the ratio of these figures was 2 : 3, not 1 : 2 as stated by the author. Moreover, there were other important factors influencing cost, namely, lubricating oil consumption, operating and maintenance costs and consequential "outages". It should also be realized that turbine engineers were now important contributors to the further development of Diesel engines as they dealt with the compressor for supplying air to burn the fuel in the cylinders and had to improve the efficiency of the turbines utilizing exhaust gas driving these compressors. The future of the heavy oil engine developing even greater m.i.p. in given cylinder sizes was intimately tied up with improvements in compressors and gas turbines.

It was noteworthy that Doxford hoped to obtain a power of 2,200 h.p. per cylinder from a 760 mm. bore engine, whereas most other manufacturers required 840 to 900 mm. for the same duty. It was very pleasing to note that in Table I the Doxford engine was shown considerably shorter and lighter than other engines which should ensure that the engine would be highly competitive.

Concerning Mr. Andresen's paper, Dr. Brown said that this was a very interesting paper, describing the large Burmeister and Wain engines illustrated in Figs. 5 and 6. Was the author's firm now convinced that the opposed piston engine was the better, giving an equal power to the poppet valve engine at 90 mm. less bore?

An interesting feature of the installation in Fig. 8 was the use of two-stage boilers, presumably equivalent to an oil-fired boiler in series with a steam/steam generator, no doubt because the secondary steam was used for cargo oil heating amongst other things. Perhaps the author could elaborate on experience with this system.

In the conclusions the author referred to possible develop-

* Millington, B. W. 1960. "Temperature Measurements in a Doxford Cylinder Liner". The Engineer, Vol. 210, p. 871.

ments in the direction of automation, and mentioned that plants with remote control of the main engines were in operation. Any further information which the author could give on the automation of main propulsion Diesel engines would be of great value.

CAPT. W. S. C. JENKS, O.B.E., R.N. (Member) said that the two papers gave an interesting description of some of the features of two important types of Diesel engine and of the factors that had influenced their development.

Dealing first with Mr. Jackson's paper, on page 464 he referred to the efforts being made to time accurately the injection of cylinder lubricating oil so that it could be used to greater effect. If these arrangements proved successful there would presumably be reduced oil consumption for an equivalent rate of cylinder wear as compared with more haphazard injection. Could the author give some information as to the percentage reduction of lubricating oil which was anticipated for equivalent wear conditions?

On page 470 reference was made to the fact that with an opposed piston engine the combustion loads were transmitted through the running gear, thereby relieving the stressing of the entablatures and columns. It was not clear, however, that this situation was materially different from that in a single-piston engine with through tie rods connecting the entablature to the bedplate, which would seem to have a more or less equivalent effect.

It appeared from Fig. 24 that the J type engine was to be built with 670 mm. bore as well as 760 mm. bore, and would presumably, therefore, supersede the P type engine which had so recently been introduced. In view of the good reports of the P type engine performance given in this paper this was rather surprising, and it would be interesting to know the principal differences between the P and J types and the reasons why there was such confidence that the J would be superior. Clearly, from the diagrams, it was much smaller and lighter, but the reasons were not clear.

In case there should be any danger of the following remarks being misconstrued he wished to say that he had been in the position of recommending to his company a change from steam turbine practice, which they had had for 40 years, to direct drive Diesel practice, so he did not think he could be accused of being biased against the big Diesel, but in a review of this nature it was very interesting to hear the views of the authors on future trends. Neither had gone beyond the present or immediate future engines.

For any form of transport, large weight and size were bound to be a disadvantage, and merchant ships were no exception to this rule. This was particularly the case with very large single units, which severely limited the flexibility of machinery layout and prevented the optimum arrangement of cargo spaces. There was bound to be pressure, therefore, for the development of smaller and lighter machinery; and the personnel problem (to which he had alluded during the discussion on "Post-war Developments and Future Trends of Steam Turbine Tanker Machinery") was another factor which could have some significant effect.

There seemed to be some kind of law which limited the life of any large mobile object. The dinosaur died out because it could not survive the changes in its environment; the whale had found its environment more sympathetic but was in danger of extermination through the malicious or unthinking acts of man. He did not know which factor would kill off the big Diesel but its day would surely come, and he wondered whether its present pre-eminence (in the range up to at least 10,000 h.p.) would last for more than another decade. Whether it would be a development of the high speed Diesel or the gas turbine or a change in the economics of the fuel situation that would provide the challenge it was difficult to say, but the challenge was bound to come. It would be very interesting to have the views of the authors on how confident they were of a really long-term future for the large direct drive engine and on the type of development most likely to produce a serious rival.

MR. G. VICTORY (Member) felt that at this International Conference it was fitting that overseas guests should be left in no doubt that in the Doxford engine, which was entirely British, this country had strong claims to have the finest marine Diesel engine in the world. Mr. Jackson followed the noble pattern of Mr. Keller and Mr. Purdie in ensuring that, good as others might be, there were none better.

Of course the Doxford engine always could burn heavy fuel, though it might be wondered what viscosity was referred to as "heavy" in Mr. Keller's days, but the success of any engine was now dependent on burning it without extra wear or maintenance and these problems appeared to have been reasonably accounted for. At the same time it would be wrong to assume that this ability to digest the indigestible would always be the prerogative of the large slow speed engine and it was hoped that the author had not lost sight of the weight handicap under which his entry would be running when smaller high speed engines demonstrated equal success with heavy fuel.

In Table I, where the author gave lengths of engine "over bed and thrust block", he felt Mr. Jackson might have been a little unfair to certain competitors who did not have integral thrust blocks. This feature of the integral thrust block had, he believed, been blamed for initiating a number of crankcase explosions, and perhaps Mr. Jackson would say what were the overriding disadvantages of the separate thrust which made him willing to decide on this extra hazard, though it was admittedly small. He was not surprised to hear that the double cylinder engine in Fig. 18 got short shrift for it was obviously a complication to put the present engine upside down with the crank on the wrong end in order to use two cylinders when one would do.

He noted also that the well-known question, "How do you adjust a spherical bottom end on a Doxford engine?" would soon be out of date. He was very sorry about this, because it had produced some wonderful answers in Engineering Knowledge examinations. However, from the *Canberra* paper it appeared that the headache lost to Doxfords and to Mr. Jackson, had apparently fallen upon the turbine manufacturer and designers in the turbine field, so it might still be possible to use the question in the Examinations of engineers if it could be transferred from the Motor paper to the Steam paper.

On pages 460 and 461 it was stated that the engines now developed with 8 deg. lead on the exhaust cranks would run ahead and astern down to 25 r.p.m. and pick up from this speed. It seemed reasonable to expect some difference in the torque available on the very different port timings and perhaps Mr. Jackson would indicate the times taken from dead slow to full in the ahead and astern direction.

The model of the new J engine had a very smooth, clean, look, but he still felt that it would look better without the camshaft and chain drive casing. He hoped they would not have to wait as long for the engine to do the work of the fuel pump by using cylinder compression on the Archaeloff system, as they had waited for the engine exhaust to do the work of the scavenge pumps.

There was one feature of the new engines which the author had not mentioned but which Mr. Victory felt to be very important. He believed that Doxfords were now prepared to undertake planned maintenance and intended having a travelling quizmaster who would visit ships and check on the running records and find out what chief engineers thought of the engines, and which items were giving the most trouble. This policy could not but result in finer engines, for the knowledge accumulated by the operational staff had for too long been ignored. In this respect he hoped that, in the new engines, Mr. Jackson had disposed of Doxford's built-in exonerator, the fusible pellets which were inserted in pistons and (he believed) liners. This was always a good get-out, but it reminded him of the insoluble question of the chicken and the egg, for if the engine could not be stopped immediately, how anyone could decide whether the pellets fused before or after the piston or liner cracked he would never know. Gas getting through the crack could interfere with the normal cooling sufficiently to cause overheating after the event, but this appeared to be ignored.

Referring to Mr. Andresen's paper, on the old single-acting four-stroke B. and W. engine there had been no difficulty in ascertaining that the cylinder lubrication was satisfactory. Regularly six or eight times a watch one would twist one's neck like a swan and, with the aid of a torch, peer up the open end of the liners. A nice overall shiny reflection was a thing of joy; a dull dark gradually spreading area was the sign for action stations. Of course, in the meantime the usual fit (or lack of fit) of the pistons ensured a strong waft of sulphurous exhaust gas in the face resulting in streaming eyes, which probably accounted for the fact that he now wore spectacles. However, on page 477 Mr. Andresen stated that the cylinder lubrication was now much superior because it was "timed". Mr. Victory wished to know how, when and where? He had never believed that at the precise instant of the pump plunger receiving a infinitesimal bump from a cam, an equally infinitesimal globule of oil obediently left the security of the lubricator quill and took up residence somewhere between the piston rings on the upward stroke of the piston. Surely most of it dribbled out of the open end after the piston had passed. Mr. Jackson had confirmed this opinion and had given details of the special arrangements made to ensure proper distribution of the cylinder lubricating oil insofar as it was possible. Mr. Andresen would perhaps state what experiments had been carried out to ensure optimum distribution of the cylinder oil now that visual inspection via the open end of the liner was not possible. Perhaps an entirely new technique was overdue in this field.

Mr. Victory also thought that the adoption of oil cooling for pistons was largely an admission of defeat. Surely it was obvious that a separate system using either water or else a glycol or oil cooling medium specially developed for cooling and not for lubrication was the optimum requirement. To adopt any other system meant that the designer could not evolve a system where mixing could not take place. To say that there was not much difference in the various media and then to show on page 477 of his paper that water cooling gave maximum crown temperature 30 deg. C. less on a piston of 840 mm. diameter than oil cooling on a piston of 740 mm. diameter (which in fact should be appreciably lower because of the reduced diameter) was surely begging the question. His experience was that on older oil cooled engines the crankcase was usually filthy with thick greasy scum, and the inside of the pistons was worse, which was a fair reflection on a design which required that the same fluid should perform its functions under two widely differing conditions.

Mr. Andresen touched on "automation", which was really an omnibus word for a process which had been going on for a long time, and embraced all instrumentation and safety devices. Progress in this field could only follow reliability, and could not entirely dispense with the requirement that so long as engines and monitoring installations broke down, well qualified personnel whose normal duties were to interpret the information fed in from various sources would still be required to repair and husband the machinery into port after one or other of the all-seeing gadgets had a bit of "shut-eye".

Mr. Lars Nördstrom said that he felt that some mention should be made of developments which were taking place in the sphere of Diesel machinery as a whole.

In the paper, "Post-war Developments and Future Trends of Steam Turbine Tanker Machinery", reference had been made to the "austerity ship" and the "automated ship", with regard to steam ships. However exactly the same sort of development was taking place in Diesel machinery. Manual operation had been normal so far but, for instance, the junior engineer, mentioned by Mr. Jackson as being needed to care for the fuel treatment system, would eventually be replaced by an automatic device and instead of daily cleaning, the purifiers would need only a monthly overhaul.

Automatic valves would be installed in many auxiliary systems, new safety devices introduced and warning systems improved, whilst central control positions, for manœuvring and supervision were already in service. Mr. Andresen had referred to this with regard to main engines but it also applied to auxiliaries. Arrangements for bringing about easier maintenance work on main engines had been mentioned, but it should also be noted that similar arrangements were made for the other parts of the installation. Altogether, a mechanized ship, referred to by some as an automated ship, was now feasible.

It was essential that this work should be carried out with co-operation between shipbuilders; such co-operation already existed between Scandinavian shipyards and engine builders. Without co-operation the same state of affairs would arise with regard to Diesel machinery as existed with steam machinery, where each turbine manufacturer was fighting for the acceptance of his own steam and feed cycle, with the result that the market was completely confused.

Mr. J. McNaught (Member) said that he would like to know more about the effect of the timing of cylinder lubrication, especially on the B. and W. opposed piston engine, where the incidence of scavenge fires was rather more than was normally experienced, and where it was a design fault rather than a matter of faulty operation. The effects were much more serious than on the older types, because with a common scavenge belt the fire spread very rapidly and could cause serious damage. He sometimes wondered whether designers took scavenge fires seriously and it would be interesting to know whether any attempt was made to reproduce one on the test bed, or whether one had actually happened on any of the sea trials. He had known of one happening on a dock trial and he could not see why it could not happen on a test.

What experimental work had been done on timed cylinder lubrication and what was the degree of lag between the pump operation and the oil injection into the cylinder? Also, what design features were there in the opposed piston and poppet valve engines which were intended to prevent scavenge fires?

With regard to the reduction of manpower, he would have liked to see designers doing something about reducing the number of hand oiling and greasing points around most of the large engines. One six-cylinder engine he knew had more than 90. He did not know how many there were on the poppet valve engine but he thought the number was about the same.

Mr. C. L. G. Worn, B.Sc.(Eng.) (Member) said that Mr. Jackson had referred to trials on a four-cylinder P type engine fitted with a single turbocharger. It might be of interest to give some details of the engine performance under these conditions when compared with the more normal two turbocharger arrangement.

The advantages of using the single turbocharger were not only reduced first cost and simplification but improved performance as a result of reduced losses in the exhaust ducting, including elimination of the "split pulse" and reduced losses in the turbine—windage, friction and partial admission losses being considered in the case of the latter.

A standard H.S.B.T. 50 Series turbocharger fitted with special three-entry turbine casing was used (Fig. D.1). This casing was, in effect, the equivalent of a standard twin-entry casing but, apart from achieving a more compact installation with smaller exhaust pipe volume, a better aerodynamic shape was possible with the elimination of right-angle bends. The centre cylinders Nos. 2 and 3 discharged straight into 180 deg. segments of turbine nozzle annulus through the axial entry in an ideal manner, whilst Nos. 1 and 4 exhausted into the remaining segment alternately through the two side entries. Between these two opposed side entries a curved deflector shape

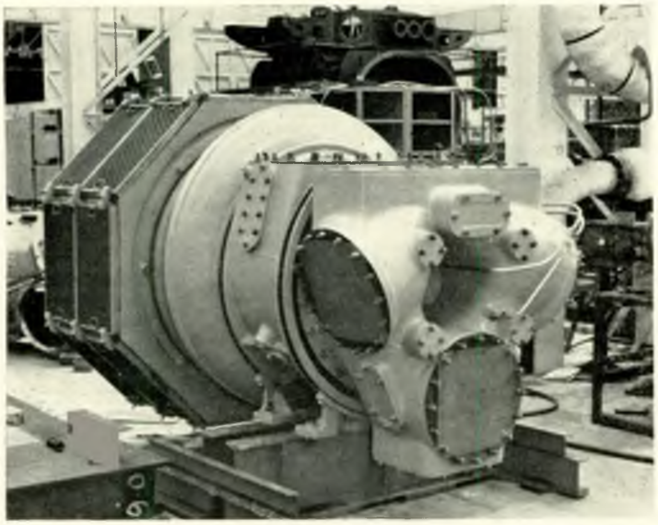
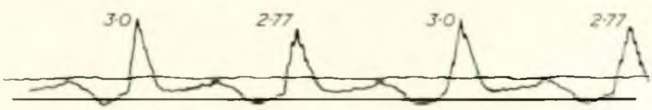


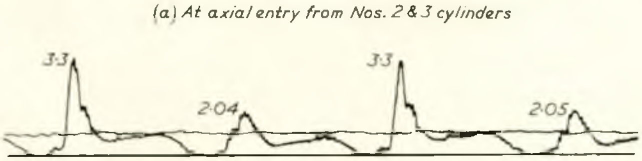
FIG. D.1—*H.S.B.T.—50 series turbocharger fitted with special 3-entry turbine casing*

was cast to direct the flow into the turbine and restrict gas flow between branch pipes to Nos. 1 and 4 cylinders.

Full load pulse diagrams were taken at the cylinder exhaust belts and just upstream of the three turbine inlets. Two were reproduced (Fig. D.2) illustrating, in the first case,



(a) At axial entry from Nos. 2 & 3 cylinders



(b) At side entry from No. 1 cylinder

FIG. D.2.

the conditions at the axial entry from Nos. 2 and 3 cylinders, and in the second case the effect of the deflector on a pulse from either Nos. 1 or 4 cylinders. In this case, the first pulse had a high peak pressure which effectively passed through somewhat over 90 deg. of nozzle segment, whilst the remainder of the segment had a reduced peak pressure due to throttling across the deflector. In the second diagram this was illustrated by the smaller pulse which had come through the deflector from No. 4 cylinder.

Engine performance showed measurable improvement in scavenging conditions when compared with a two-blower build. It was not possible for the comparative tests to be performed on the same engine, and as the single-blower trials took place during the running-in period of a new engine, it was considered

TABLE D.1

	Single Turbochargers	*Twin Turbochargers* (Figures from Fig. 17)
Boost pressure lb./sq. in. gauge	6·7	7·4
Specific air flow lb./b.h.p. hr.	14·6	13·6
Mean cylinder exhaust temperature deg. F.	675	710
Maximum combustion pressure atmos	58·5	58·5

that the only valid comparisons which could be made were those concerned with air flow, pressures and temperatures.

From Table D.1 it could be seen that a lower boost pressure was achieved with the single blower; unfortunately there was insufficient time to make an adjustment to the turbine nozzle area. Even so, the total air flow through the engine was increased with a significant reduction in mean cylinder exhaust temperature; i.e. assuming the same turbocharger internal efficiencies, improved flow conditions of the exhaust system had resulted in a lower resistance to flow across the engine. Included in this improvement was the effect of the reduced partial admission losses of a single turbine.

The effect of the improved flow conditions for the tests performed was shown in exhaust temperature and not in fuel consumption. Rematching to optimum boost pressure should permit a higher load to be carried per cylinder for given limiting conditions.

As a means of comparing the performance of turbochargers operating under different conditions and with different exhaust arrangements, a so-called "apparent turbocharger efficiency" had been adopted. This was defined as the efficiency a turbocharger would need to have if it operated on a purely constant pressure system and given the same engine conditions of air flow, pressure differential during scavenge, exhaust back pressure and exhaust temperatures as were actually obtained with the pulse system employed. This apparent efficiency was then higher than the true steady flow value owing to utilization of the energy of the blow down pulse; it could be considered as the product of true turbocharger efficiency as measured on a steady flow test rig and a "pulse factor".

Calculations from the figures obtained during the tests referred to showed that the single blower build had an apparent efficiency of 83 per cent as compared with 71 per cent for a twin-blower build fitted to a similar engine. If a value of true steady flow efficiency for the turbochargers was assumed to be 62 per cent in each case, then the pulse factors became 1·34 and 1·15 respectively. That was twice as much pulse energy utilized with the single-blower arrangement.

The adoption of an optimum exhaust system was, therefore, of some importance and could yield measurable economies in operation. It was particularly important that those who chose to use the pulse system should take full advantage of the fact, especially in relation to their competitors who adopted the constant pressure system.

MR. BRYAN TAYLOR, B.Sc.(Eng.) (Member) referring to cylinder liner wear and the effect of chrome deposit on the cylinder bores, said that this subject had been touched on by both the authors but they did not come down conclusively for chrome bores or plain cast iron liners. From data collected from medium speed direct coupled engines of the four-stroke type, using marine Diesel fuel, it now seemed that with the recent developments in cylinder lubricating oil there was hardly a case for using chrome bores.

When the Series 2 type detergent oil was used for cylinder lubrication the average rate of liner wear was found to be between 0·0015in. and 0·003in./1,000 hr. in engines of 390 mm. bore with plain liners. In the case of chromed bores the rate of wear was reduced to something of the order of 0·00025in./1,000 hr.; in other words, the rate of wear in the chrome bore was about one-eighth of that in the plain bore.

With the introduction of the alkaline additive type of lubricating oil they had found considerable improvements in the rate of wear, and compared with that obtained with the Series 2 type of oil the wear rate had been reduced to about one-quarter of its previous value. What had been the authors' experience in the large slow running engines? Had they had a similar experience?

The other point he wished to deal with was that similar or even better results had been obtained in regard to piston ring wear. The limiting factor on the running hours between piston overhaul had in the past been the rate of wear of the top ring. It had been found that in the case of a chrome top ring running in a plain bore, using a Series 2 oil, the piston had to be overhauled at about 2,000 to 3,000 hours, because the ring gap had opened out something like 1 per cent of the bore, which he usually gave as the limit on the allowable wear. By using an alkaline additive oil the rate of ring wear had been cut to approximately one-twelfth of its previous value so that the time between piston overhauls could be extended considerably.

Mr. Andresen had mentioned that where particularly requested his company did fit chromed liners and he was rather interested to know that Mr. Andresen adopted a thicker chrome deposit over the top part of the liner, up to 1·8 mm. Did Mr. Andresen consider such a thick chrome deposit to be economical in the long run?

Mr. K. V. Menon (Associate Member) said that in speaking of Diesel engines he recalled the nightmares concerning the number of exhaust valves overhauled and liners changed in the heat of the tropics. It was very satisfying to see that an immense amount of research and development had been carried out for the improvement of the marine Diesel engine and he congratulated the authors and their colleagues on the great work that they had done and were still doing in this field.

The marine Diesel engine owed its existence to the older generation who had worked hard under adverse conditions seldom found today.

With regard to the item about the marine engineer being allowed to carry out repairs, it was true that, in contrast with service on a steam turbine ship, a Diesel engineer was expected to do the major part of the repairs himself, during the trip and under conditions not necessarily of the best. The research into the efficient operation of the engine, having due regard to the length of service required from marine conditions, should not end on the shop floor. Unfortunately, it seemed that while development of the Diesel engine itself had gone on apace, scant attention had been paid to the improvement of the maintenance equipment. If Mr. Jackson would excuse him, he thought that it was high time that, for Doxford engines, hydraulic torque spanners and other tools for the perfect maintenance of the machinery were available. These should be available on board ship.

What were the authors' opinions as to the work which had been done to make engine maintenance easy? Even these days, old hammers and chisels could be seen in use in shipbuilding yards.

Mr. A. G. Arnold (Member) said that he joined the many who would congratulate the two authors upon their excellent and informative papers.

It was to be very much regretted that Mr. Jackson had once again had to report annoying failures of "big end caps" as described on page 465 of his paper and illustrated in Fig. 14. There was no doubt that flame cutting of such important parts had caused a considerable amount of trouble in the past. This was referred to emphatically when Mr. Jackson read his excellent paper* some two years ago, and, no doubt with many others, he thought that this was at least one mistake that would not be repeated.

Cylinder lubrication and liner wear had been the subject of discussion since the first marine Diesel engine, but he could not recall any more interesting diagrams illustrating the subject than those Mr. Jackson had given in Fig. 9, page 464. He looked forward to reading Mr. Butler's paper, when the subject would be gone into more thoroughly.

He had known Mr. Andresen since he first came from university in Denmark, and had worked closely, and agreeably with him. Mr. Andresen was a dedicated Diesel engineer, and Mr. Arnold agreed very much with Mr. Pounder that he would be very difficult to replace.

Mr. Arnold was very familiar with the opposed piston engine illustrated, but not familiar with the poppet valve type, in service, but it appeared to be gaining a good reputation.

Mr. Andresen would know that one of the main reasons for the opposed piston engine being developed, was the very serious troubles that were experienced with cast steel cylinder covers in the earlier type of his firm's engine, and Mr. Arnold wondered if there had been any trouble in this respect with the current type of engine described, if not, what was the difference either in the analysis and/or foundry technique. He wondered if the exhaust valves sometimes stuck in the manner that they had done in the earlier four-stroke engines, if so, what effect did this have on the exhaust cams, etc. and was there any record of the fuel pump cams wearing excessively?

Had Mr. Andresen given any serious thought to adopting a hydraulic mechanism for operating the exhaust valves? This would make it possible to eliminate the camshaft entirely with the resultant saving in weight and money, etc.

There was nothing new in this system, Messrs. Scotts applied it to their Scott-Still Engine in 1922, and it worked satisfactorily for 22 years, until the vessel in which it was fitted was lost.

Professor G. H. C. Chambers, D.S.C. (Member) said that he would like to put one brief question to both the authors. As the size of these Diesel engines increased, the amount of intrinsic work done in building them up in the shop and taking them to pieces again and building them up in ships also became greater. As the number of Diesels produced to the same design, and their degree of development, increased, what were the prospects of this erection and dismantling being dispensed with?

* Atkinson, R. and Jackson, P. 1960. "Some Crankshaft Failures: Investigations, Causes and Remedies". Trans.I.Mar.E., Vol. 72, p. 269.

Correspondence

Mr. S. Archer, M.Sc. (Member) wrote that with the keen competition currently existing within the marine Diesel engine industry, Mr. Jackson's paper describing the Doxford direct drive engine in its "new look" was very welcome in that not only did it give a general review of the research and development work leading up to the P and J engine types, but also frank and instructive details of the early teething troubles with the former, inevitable in any new design. In addition, he had given what some might say was a somewhat tantalizing peep into the future so far as the large Doxford engine was concerned.

All this, in Mr. Archer's view, augured well for the competitive position of the British large Diesel engine industry in the years ahead and all credit was due to Mr. Jackson and his associates for their share in this resurgence.

In his introduction the author referred to the comparative efficiencies and fuel consumptions of steam and Diesel machinery. A point sometimes lost sight of, however, was that the efficiency of any steam plant deteriorated after entering service, due of course to falling transmission efficiencies in boiler heating surfaces and thermodynamic efficiencies in, for example, turbine blading, whereas the overall thermal efficiency of the Diesel engine remained almost constant in service.

In the list of factors mainly responsible for the rapid progress of the marine Diesel engine during the past fifteen years, he felt that Mr. Jackson might justifiably have added two more, namely 6) welding of engine structures and 7) the high pressure supercharging of the four-cycle engine by means of exhaust turboblowers. In his view, it would be wrong to ignore altogether the progress made in the latter, especially in Germany and Japan.

Mr. Jackson had given a most lucid account of the Doxford system for burning heavy fuels and Fig. 1 was a very useful guide to present practice. As regards Mr. Jackson's claim (tentative though it was!) of complete freedom from scavenge fires on any turbocharged Doxford engine since *British Escort,* all he could say was that the survey records of Lloyd's Register supported this claim 100 per cent.

From Fig. 4, showing the interesting changes in port timings for the various engines, it appeared that the improvements had resulted from lengthening the period of exhaust opening whilst reducing the period open to air and also advancing the commencement of exhaust opening by some 10 deg. It would be interesting to know what parallel changes in port area, if any were made. Incidentally was there any adverse effect upon full load power due to the changes?

Fig. 6, showing the modification to the stuffing box for the water-cooling of the upper piston, was a neat design and should be about as "foolproof" an arrangement as could be imagined.

Fig. 8 indicated much higher rates of wear on the lower cylinder liners of m.v. *Montana* compared with those on the upper liners. Presumably, this could partly be ascribed to the lower mean temperature of the scavenge air-cooled lower liner and the consequently greater tendency to condensation of H_2SO_4, etc. on the cylinder walls on the lines indicated by the author in Fig. 3. Also, of course, any such acids would tend to run downward away from the upper liners. Could the author please comment on this point?

The full size *ad hoc* experimental work on the scavenge ports to determine the optimum amount of swirl, etc. seemed ingenious and effective. It was interesting to note that the conditions in the cylinder were presumably sufficiently turbulent to rule out the possibility of stratification in the shape of a central core of inert exhaust gas.

The failures of the side rod bottom end keeps were disappointing and tended to confirm the undesirability of using flame-cut components in highly stressed locations, especially in the presence of notch effects. Incidentally, it was understood that such flame-cutting was usually done hot in the steel works and presumably this precaution was not taken on the keep material which failed?

He was particularly interested in the reported good experience to date with the solid arrangement of transverse beams, etc. Rough calculations showed that the upper piston side clearance should be sufficient to absorb even as much as ⅛in. differential wear-down between forward and after side rod "legs" by virtue of its ability to tilt in a fore-and-aft plane. However, it would still seem desirable to retain some means of adjusting the relative alignment of piston and beam, e.g. by means of the spherical pad, in order to minimize upper piston ring and liner trouble, also probably side rod top end bearing trouble in the event of excessive differential wear occurring on the side rod bottom end bearings. Had there been any evidence of fretting at these connexions?

As regards the calculated torsional characteristics of the 85 P.T.6 engine given in Fig. 20, it was presumed that the crankshaft stresses shown due to the 9th order, 3-node criticals were for the engine without a detuner or damper? If so, it should surely have been possible to tune the 9III critical below, say, 100 r.p.m. and supply some form of additional damping, using a crankshaft of 690 mm. diameter. What was the difficulty involved? Was the author afraid of bringing down the 7th order too near the service speed?

It was interesting to note the typical performance curves of P type engines given in Fig. 17, including the remarkably high mechanical efficiency figures of 95/96 per cent. Were these based on "Farnborough" indicators or were normal type instruments used, and what limits of accuracy would the author claim for the indicated power measurements?

In conclusion, a study of Table I, summarizing present day engines capable of developing 20,000 b.h.p., showed very clearly that, certainly so far as overall dimensions and weights were concerned, the Doxford engine of the future should be in a very competitive position.

Mr. Archer also wrote that Mr. Andresen's paper was so largely factual that it left little scope for discussion other than to congratulate him upon his ability to compress such a lot of valuable information into so relatively small a space.

There were just one or two comments he would like to make.

As regards the fully-built crankshaft design, it was thought that with a minimum eyehole bridge thickness of 27 per cent of the shrinkage diameters and a web material having a lower yield point of about 15 tons/sq. in. this was about as far as the fully-built design should be pushed and thereafter recourse should preferably be had to the semi-built type of shaft, since otherwise the uniformity of grip pressure around the eyeholes was likely to be adversely affected.

It was noted that the author felt that, at least for the present ratings, there was no compulsion to go to water cooling, for which he cited experimental evidence in the form of comparative temperature measurements. On the other side of the ledger, however, there was the continued possibility of carbon deposits in the pistons and in the crankcase lubricating oil. The suggestion had also been advanced that one of the reasons why crankcase explosions seemed to be more prevalent today than in earlier days of water cooling was the mixing of piston cooling oil and crankcase oil, leading to relatively higher temperatures in the crankcase than in former years. Did the author think that any worthwhile increase in security could be achieved by, for example, arranging separate systems for piston cooling and bearing lubrication?

On a recent ship, not fitted with B. and W. engines, although nothing untoward had been noticed on the trials, trouble developed with the turboblowers on the maiden voyage. It was a nine-cylinder, two-stroke engine having three turboblower aggregates and at a certain speed the centre blower began to emit a periodical high-pitched scream which could only be eliminated by reducing speed or by blowing off air from the scavenge trunk. Had the author ever experienced this phenomenon and could he please suggest its cause and possible cure?

Mr. R. Beattie (Member) wrote that his contribution to this discussion must of necessity be limited owing to the fact that the range of powers with which he had been in continuous contact over the last ten years were limited to the 1,000 to 2,000 b.h.p. class. Nevertheless, he tried to keep abreast of the latest developments in the higher power range. Mr. Jackson and his colleagues had succeeded in producing two excellent papers on the new Doxford engine, the first* of these, published in the July 1961 issue of Transactions, was in his opinion a masterpiece of constructional illustration which he had not seen bettered so far in any technical journal. The wealth of detail contained therein enabled individuals such as himself to comprehend the simplicity of the design of the new engine.

One point which had always puzzled him was why the large engine builders had avoided until comparatively recently the use of turboblowers, especially with the introduction of low grade fuels. In the early stages of this development, i.e. around 1929-30, four-cycle engines of 800-1,200 b.h.p. were being supercharged up to 30 per cent. Today it was becoming more or less standard practice in the lower power group to supercharge both the two and four-cycle engines to 80 per cent and even 100 per cent and over in some cases, admittedly using

* Jackson, P. 1961. "The Future Doxford Marine Oil Engine" Trans. I.Mar.E., Vol. 73, p. 197.

marine Diesel fuel, but surely what could be achieved in this field could also be achieved with the heavier units. It could be done, as both Doxford and other builders had proved.

He would say that the advantage in this respect definitely lay with the two-cycle rather than the four-cycle engine and more so with the opposed piston engine, where ports offered much better facilities for exhausting the waste products of combustion than valves; nor could the uniflow engine be left out of the reckoning and it had been adopted for at least two types in the high powered range.

Mr. Jackson referred to the experiments carried out with regard to the desirability of using one, two or three turbochargers and the accompanying remarks made interesting reading. The desirability of retaining the scavenge pump for manœuvring purposes made him wonder if it would not be advantageous to fit the blowers in series with the scavenge pump, as had been done on a well known make of smaller propulsion unit where loop scavenging was employed. The blower in this case supplied air to the scavenge pump under pressure. Would this be practical in a Doxford engine?

The alternative method, upon which comment was made in the paper, was the fitting of an independent blower for manœuvring purposes. He knew of at least two normally aspirated, two-cycle engines, fitted in seagoing ships, where the scavenge pumps were entirely eliminated and electrically driven blowers installed, with very successful results.

He was sure that the section dealing with liner lubrication was of interest to all engine operators, especially with regard to the differential needle control fitted to the quills, which must have obvious advantages, but he felt personally that the introduction of the Alan Muntz ball type flow indicator had been one of the most advantageous innovations in so far as it enabled a reasonably accurate measurement of the amount of lubrication being delivered to the cylinder walls. He believed that he was one of the first to fit these flow indicators to plant in his charge. The fact that the amount of oil being delivered could be regulated by means of the graph supplied, enabled the flow of oil to be cut to a reasonable quantity without affecting liner wear. Engine builders were always on the generous side with a new engine and few operators would risk altering the original setting, just as few engineers would run the Diesel engine cooling water at temperatures above 160 deg. F. on the outlet side, whereas, as Mr. Jackson rightly pointed out, temperatures of 180 deg. F. and over would give far better results. He thought back to his early days at sea and recalled the old fashioned idea of sacrificing vacuum for a hot feed by wire drawing the sea inlet, to obtain the maximum temperature in the feed water before entering the heater. He had found that even with thermostatic controls on fresh water regulators, settings were frequently altered to give much lower temperatures than were desirable.

Pre-heating of the cooling water had always been attractive to him and in the smaller class of engines which he had under his charge the central heating system and generators were coupled to the main engine cooling system so that the temperature never dropped below 130 deg. F. when the engines were lying stopped.

One of the features of the old Doxford engine was the amount of deflexion in the crankshaft. Had the elimination of the spherical bearing reduced this deflexion to reasonable limits?

In the earlier Doxford engines the transverse beams were manufactured from cast steel in the majority of cases, and several of these beams fractured in service causing severe damage, such as bending of side rods and damaging the pistons, etc. Investigations into the cause of this fracturing seemed to indicate that the initial cause was fatigue and the fatigue fracture usually began in way of one of the holes in the beam. He believed that all transverse beams were now made of steel and he would be interested to know if this had completely cured the trouble.

Had crankcase corrosion now been completely eliminated as a result of oil cooling the low piston and fitting a gland?

Finally, there had been a few cases where fractures had occurred in fabricated bedplates in the main bearing pocket. Fracturing in this region could be very difficult to repair effectively and he had heard it suggested that the fracturing was caused because the fabricated bedplates were not stress-relieved after welding. Had this trouble now been cured with the fitting of double transverse girders and were the bedplates stress relieved after welding?

In conclusion he would like to add his congratulations to Mr. Jackson and his colleagues on producing such an excellent paper on the present and future engines which Mr. Jackson's company were producing and hoped to produce.

Mr. F. C. Bown (Associate Member) wrote that he was rather surprised to see that on page 458 of his paper, Mr. Jackson made a general statement that usually an extra junior engineer was carried to maintain the centrifuges and clean them as necessary, when more than 30 tons high viscosity fuel were burnt per day. This might have been the case in the early days of the burning of high viscosity fuels, but it was certainly not the case today. Mr. Bown personally knew of many cases where existing ships were converted to high viscosity fuel oil with no addition at all to the personnel in the engine room. In any case, the development of the self-cleaning centrifuge had made frequent cleaning of purifiers unnecessary, and, in many cases, ships went for several months without stripping out self-cleaning purifier bowls.

When the author was replying to the discussion, he mentioned scavenge fires and the probable causes. In Mr. Bown's experience of scavenge fires, in all cases the cause was found to be bad combustion caused, or contributed to, by broken or worn piston rings together with worn liners. It would be interesting to hear the author's view on whether this was a more probable cause of igniting the sludge in the scavenge than that mentioned.

Mr. A. M. Bennett (Member), in a written contribution, pointed out that the pre-occupation of Doxfords with the production of current engines up to 1958, to which Mr. Jackson referred, was observed by many shipowners and widely regretted. The Doxford engines up to 7,000-8,000 brake horsepower which were produced in that era were second to none and, therefore, it was most gratifying to learn from recent papers by the author and his colleagues of the endeavours that Doxford's were making to develop an engine of considerably higher power without sacrificing the opposed piston principle.

His own company's experience was that the Doxford engine, prior to the P type of which they had no experience, had a much lower specific fuel consumption than other types of engine and, moreover, it was generally quieter running and better balanced; furthermore, it was difficult to accept the suggestion that loop scavenge could be so efficient thermodynamically.

Was the author correct when he stated that the first cost of a Diesel plant was greater than the modern steam plant? From their investigations for moderate powers the two broke about even.

He thought it was a reflection, that Mr. Jackson admitted by implication that it was left to the shipowners to pioneer the burning of high viscosity fuel, and it was most deplorable that even now so many engine builders still refused to run the shop trials on the grade of fuel which the engine was designed to burn. A further point of disagreement was found in the author's statement that an extra engineer was needed to maintain the heavy oil centrifuges and clean them as necessary. With many heavy fuels the extra work of purifying could not be said to justify additional staff, and in any case the self-cleaning centrifugal separator was available.

Mr. Bennett thought that the author had a point when he made the observation that Doxford engines were free from scavenge fires. He did not know of an engine less prone to this irritation.

He would disagree with the advice of the author to change from high viscosity fuel to Diesel fuel when manœuvring. In his opinion the advantages of not doing so far outweighed the disadvantages of the complications of the actual change-over, with the greater risk of gassing. The author made the point on page 459 that the fuel pumps suitable for high viscosity fuel were not suitable for Diesel oil. It had been his company's practice, on twenty-five ships operating on fuel oil at 1,500-3,000 seconds Redwood No. 1 viscosity at 100 deg. F., to manœuvre entirely on this range of fuel. The voyages on an average were of three months duration and ships might call at anything up to thirty ports and navigate several rivers in that period of time. The average cylinder liner wear rate was 0·0075in./1,000 hr., which compared with similar ships operating on Diesel oil, with a cylinder liner wear rate of 0·005in./1,000 hr. It was difficult to believe that the former wear rate could be improved by manœuvring on Diesel oil, but if it were it was more than compensated for by the saving in cost of the heavy fuel. It must be appreciated that the change-over had to take place long before the normal standby, and therefore could amount to a considerable quantity of Diesel oil in a round voyage. However, he accepted the fact that it was important to have an efficient heating and circulating system.

The author was right in saying that there were many unfavourable rumours about P type engines, and it was gratifying to read that Mr. Jackson had been able to repudiate them all. The failure of the side rod big end caps received much prominence but was directly attributable to bad practice. The square-cut corner needed no comment but it was well known that flame-cutting very often started cracks in the vicinity of the cutting surface, and his company had experienced a Doxford crankshaft web failure due to the hole being flame-cut instead of trepanned. Mr. Jackson mentioned that the crankshaft deflexions, with the new type of bedplate, had been more than halved. He would like to ask half of what? Surely half of what was acceptable on the previous Doxford engine crankshaft was not acceptable for the type of crankshaft fitted to the P type engines.

The author mentioned that there had not been any cases of carbonization of the piston cooling oil in Doxford engines, but Mr. Bennett's company had experience of lubricant breakdown of a less drastic type due to general oxidation of the oil. This was not common only to this particular type of engine, but with all piston cooling oil it took place gradually and could be monitored by the increase in the organic acidity in the oil. It had been overcome largely in their fleet by a suitable inhibited oil, but he presumed that the best practice was to separate the piston cooling oil from the lubricating oil system.

The information given on the latest engine was insufficient for any reasoned comment, but he looked forward to 1963 when perhaps it would be possible to see the new engine running and to learn something of the details of its construction. It was to be hoped that the design and details of the new engine would be given the attention which had been such a feature of competitive engines in recent years.

Mr. Bennett and his company wished the author every success in his endeavours.

MR. JUST FR. STORM noted that Mr. Andresen had stated that the mechanical noise from a modern two-stroke, low speed engine was normally below the permitted level. Measurements on board 15 new vessels during the year 1961 showed that this statement was far too optimistic.

The new vessels subjected to these surveying measurements were of different manufacture, and might be considered representative for the general situation. The survey covered different locations in the engine room, and for each location, and each octave band, the median value for the group was determined. Fifty per cent of the ships had octave band levels higher than this median value. In no location were these median values below the reference curve, presented by the author in Fig. 2, in all bands, except in the workshop. In the neighbourhood of the supercharger the median values were about 5 db above the reference curve in 4 high frequency bands, and for some of the ships the readings here were 15-20 db higher.

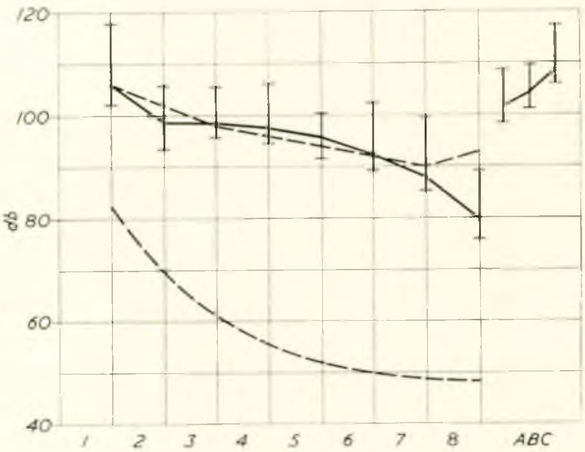


FIG. D.3—*Octave band noise on manœuvring platform. Median and range for 15 ships with slow running Diesel engines*

The diagram (Fig. D.3) showed the median values for the measurements on the manœuvring platform (solid line) superimposed on the reference curve (dotted line). The diagram also gave the maximum readings and the minimum readings for the group of 15 ships; these values were connected with a solid vertical range line for each octave band. The band limits were almost the same as in Fig. 2 of the paper, except that the lowest and highest bands (band 1 and 8) covered the intervals 20-90 c/s and 5,600-16,000 c/s respectively.

It should be noted that the noise level ought to be well below the reference curve in all bands in order to avoid the use of ear protectors.

COMMANDER C. M. HALL, R.N. (Member) wrote that amongst the items in Mr. Andresen's most instructive paper, he was particularly interested in the author's description of the pumps hydraulically driven from the main engine, the economic value of which coupled with the exhaust gas boiler was a most important feature. He would be glad to know Mr. Andresen's estimate of the overall efficiency of this drive together with the capacity of the generator pump, as this appeared to run at a lower pressure than would be expected. It would be interesting to know whether any troubles in connexion with cavitation or aeration had been experienced and whether the author thought it would be practicable to apply this system to engines where the oil requirements were relatively low, e.g. where water cooled pistons were used.

MR. L. PARODI (Member) congratulated Mr. Andresen on his very interesting paper.

The author had given an example of the machinery installation of a modern motor tanker. In this ship the electric power for the requirements at sea was obtained from a turbo-generator, driven by steam from the exhaust-gas boiler which had a steam evaporation rate of 7,000 kg./hr. (15,500lb./hr.). The pressure was not given but it could be supposed that it would be about 7-8 kg./sq. cm. It would be of interest to know whether the author had been able to calculate in advance the relationship between the area of the evaporating surface of the boiler and the consequent back-pressure on the exhaust gas duct.

In Italy they were looking into a similar arrangement and at the first attempt the back-pressure was not very high, but the steam output did not reach the figure suggested by Mr.

Andresen. It was feared that, by increasing the heating surface of the exhaust-gas boiler, the back-pressure would become too high. Might he have some more information upon this point?

Mr. H. Andresen wrote that he wished to congratulate Mr. Jackson and to thank him for his valuable paper, which he had studied with great interest—not only the information on the existing Doxford P type engine, but also what might be expected from the next step in the development, the J type engine.

In connexion with the performance curves Mr. Jackson mentioned that the cylinder rating was independent of the number of cylinders and he fully agreed with this important statement, which confirmed that engines with longitudinal scavenging could be turbocharged in such an effective way that the small variations in air flow for different cylinder numbers were of no importance.

From the performance curves it was seen that the air flow for the P type engine was only 6·2-6·8 kg./b.h.p.-hr. It was, however, difficult to judge what influence this might have on the reliability of the engines as the exhaust temperature before the turbines varied between 325 deg. C. and 465 deg. C. and the temperatures of the scavenging air were not indicated. It should, therefore, be interesting if Mr. Jackson could give some more detailed information regarding this question.

Regarding the proposed new J type engine Mr. Jackson mentioned that the weight of the crankshaft for a 9-cylinder engine would be 125 tons. From this Mr. Andresen understood that the crankshaft was of the semi-built type, with eccentrics, and that the crankpin diameter corresponded to a m.i.p. of about 9·3 kg./sq. cm. and to Lloyd's existing rules. This seemed reasonable when only 90 per cent of the rated output was recommended to be developed in normal service. For these conditions the weight of the crankshaft corresponded very well to what he would consider normal.

The weight of the engine seemed, however, to be extremely low compared with the weights of other engine types and the explanation might be that special materials would be used to a great extent. He would welcome any clarification Mr. Jackson might be able to give in this respect.

Author's Reply

Reply by Mr. Jackson

Mr. Jackson thanked the many speakers who had contributed to the discussion and thus helped to enhance the value of the paper. He much appreciated the kind remarks of Mr. Pemberton and was grateful for his good wishes for the future success of the Doxford engines. He was in agreement with Mr. Pemberton that while there would be advances in the turbocharging of large Diesel engines, and higher mean pressures would be achieved, these could only be obtained by higher maximum pressures which entailed higher stresses, higher bearing pressures and larger parts for the crankshafts of a given size of engine or equal parts for the crankshafts of engines of the same power. He therefore did not expect any further rapid development in the turbocharged engine until better materials were found for crankshafts or bearings capable of withstanding higher loads, or an improvement in the shrinking together of large crankwebs and journals which would permit Mr. Pemberton to reduce Lloyd's requirements for these crankwebs. A number of speakers seemed to disagree with the statement that an extra junior engineer was required on ships burning more than thirty tons of high viscosity fuel per day and it would appear that with modern self-cleaning centrifuges this was no longer necessary.

Mr. Andresen had raised the question of air flow for engines of the various numbers of cylinders. One maker of large marine engines had increased the power of those of his engines having multiples of three cylinders, viz. the six, nine and twelve cylinder engines. Neither Burmeister and Wain nor Doxford's had done so. It was true that the amount of air delivered on engines having multiples of three cylinders was higher than for other cylinder combinations and this would permit of a higher power being developed for a given exhaust temperature but, on the other hand, maximum pressures would increase with mean pressure and this would result in greater liner and piston ring wear. Thus, if the power of the engine was increased by ten per cent, the maximum pressure would probably increase by an equal amount and the rate of wear correspondingly.

Mr. Pemberton had raised the point of fuel consumption. There had been little improvement in the fuel consumption of Diesel engines over the past ten years apart from the fact that the turbocharged engine had enabled a reduction in fuel consumption of the order of four to seven per cent to be achieved relative to that of the normal aspirated engine, but this was entirely due to the improvement in mechanical efficiency of the turbocharged engine. This gain in mechanical efficiency was due to the elimination of the scavenging pumps of the engine and the power which they absorbed, since with a turbocharged engine the energy required for the pumping of the scavenging air was obtained from the energy of the exhaust gases from the engine. Apart from this, he fully agreed with Mr. Pemberton that there could be little improvement in the fuel consumption of the Diesel engine.

Mr. Yamaguchi had asked a number of questions regarding the temperature of the fuel oil at the injector nozzle. On the Doxford engine tracer pipes were run alongside the fuel pipes from the pump to the injector and the steam in these tracer pipes maintained the temperature of the fuel or even increased it at the nozzle. He agreed with Mr. Yamaguchi that a booster pump was required to pressure feed the engine fuel pumps and thus avoid gasification of the volatile constituents in the fuel. He had not experienced any clogging of the cylinder oil on the cylinder walls due to increasing the cooling water temperature, in fact, it was usual with these higher temperatures to use a heavier oil. The fuel injection pressure of 7,000-8,000lb./sq. in. was adopted to maintain the reliability of the fuel pipes. Doxford's had used an injection pressure of 10,000-12,000lb./sq. in. pre-war, but there had been difficulties in maintaining pipe joints and there had been cases of fuel pipes splitting. The corrosion of crosshead pins and crankshafts had been caused by the sludge from the combustion of the heavy fuels getting past the piston skirts into the lubricating oil in the crankcase and mixing with water leaking from the cooling water service. This sludge contained a high percentage of sulphur. The lubricating oil had been purified in a centrifuge but this had not been sufficient. Mr. Jackson said he agreed with Mr. Yamaguchi that advancing the opening of the exhaust ports was advantageous in preventing blow-back of exhaust gas through the scavenge ports, but he did not agree that the use of controllable pitch propellers would simplify these large Diesel engines. The manœuvring and reversing system of the Doxford engine was very simple and the mechanism much simpler than that of the variable pitch propeller and much less costly.

Mr. Jackson said he was pleased to hear from Mr. Pounder that the famous firm of Harland and Wolff proposed to continue the manufacture of opposed piston engines alongside poppet valve engines. He, also, had been surprised that it was necessary to tie the cylinder heads of the large ten-cylinder poppet valve engine to the side of the ship, and to his knowledge, this had only been done on two Doxford engines which were both in bulk carriers with the engines installed right aft in a very sharp stern which had practically no strength and there had been some vibration due to the torque reaction. This had been overcome by two very simple ties between the engine and the ship's structure.

Mr. Jackson found the comparisons made by Professor Aertssen very interesting. It was appreciated that the rate of liner wear of Diesel engines was much the same as pre-war but, in the meantime, the engines had been turbocharged resulting in fifty per cent greater power from a given size of engine and the type of fuel being burned today was certainly heavier than that of pre-war days and contained much more sulphur. With regard to the fuel consumptions given in the paper, these figures were obtained on the test bed since it was not possible to obtain really accurate figures at sea, due to difficulties of measuring the fuel used and, in addition, it was difficult to measure exactly the shaft horsepower developed by the engine.

The analysis of the heat flow through the piston head and cylinder liner, given by Mr. Alcock, was very interesting and these experiments did increase our knowledge. However, when, anything goes wrong with the parts of an engine, such as the piston head or the cylinder liner, the designer gives his utmost thought to the question and makes improvements in either the design, the materials or the cooling, or all these and thus cures the problem without, he feared, resorting to calculations of the heat flow.

In reply to Admiral Bose, Doxford's did not make auxiliary engines and the smallest engine being considered at present was of 600 mm. bore to give about 1,400 h.p. per cylinder in turbocharged form. Engines as low as 400 mm. bore had been built in the past giving about 350 h.p. per cylinder, but such engines were now too small for Doxford's requirements. With regard to the question of vibration, there had been no trouble on Doxford engines due to vibration over the past fifteen years. The opposed piston engine was the best balanced engine and seven and nine-cylinder engines were really good in this respect. With regard to the five-cylinder engine, the firing order was so chosen that the secondary couples were very small, under 50 tons/ft. in magnitude and the rotary and primary couples were very low due to the fact that there were three cranks per cylinder, the outer two running opposite to the centre crank and thus balancing it; in the same way, the motions of the upper and lower pistons were opposite to each other and tended to neutralize their inertia forces so that the out-of-balance couples were very low and, in most engine sizes, the magnitudes of the couples were less than one-third of those of equal single piston engines. There had been several cases of vibration during the past year on bulk carriers with other engines when fitted aft, due to the engines being mounted over the node of the 4th order vibration form of the ship and thus the out-of-balance couples excited the vibration form when the engine and ship frequencies coincided.

Mr. Jackson acknowledged Dr. Brown's figures regarding the relative horsepowers of Diesel and steam turbines for the year 1960 but, as he remarked, 1962 onwards would be more favourable to the Diesel engine. He could not, however, agree with Dr. Brown's figures regarding the fuel consumption of steam turbine installations. There was not a single example at sea where a performance of 0·52lb./s.h.p. was being obtained and, generally, 0·56 and even 0·6 was a common figure. In this connexion he would refer Dr. Brown to the contribution of Mr. Archer of Lloyd's, who was in a position to obtain accurate comparisons of the consumptions of Diesel and steam turbine installations.

In reply to Captain Jenks, Mr. Jackson stated that the timed lubricating system had been operated with a consumption of only two-thirds that of the sight feed lubricator system, but the aim was not so much to reduce the consumption of lubricating oil, desirable as this would be, but to reduce cylinder liner wear by ensuring better lubrication. The running gear and upper and lower cylinder construction of the P and J engines were identical but the crankshaft of the J engine was so much shorter and more rigid that it resulted in a shorter engine and it would be considerably less costly. Since it employed the same running gear and was advantageous in other respects, it should be of advantage to the shipowner. Mr. Jackson said that he believed that the J type engine was approaching the limit of present development and he anticipated stablization of engine design over the next decade. He agreed with Captain Jenks that the large slow speed engine was not likely to be the ultimate form of propulsion for seagoing vessels but at the present time he could not see any other which would oust it from its present supremacy. The gas turbine and free piston engine depended upon new materials being developed; the geared Diesel engine had not yet reached a sufficient stage of reliability; nuclear power was in its infancy and would doubtless have many teething troubles before it was recognized as a reliable means of propulsion for ocean-going vessels.

Mr. Victory's contributions to these discussions were always pertinent and enlightening and were so in this case. All the figures given in Table I were similar dimensions of the various engines over the beds and thrust blocks and they were an equal comparison for all types. Mr. Jackson said that he favoured the integral thrust since it was directly coupled to the engine bed and, in addition to being bolted to the seating, was supported by the bolting down of the bed and thus formed a much more rigid structure for taking the variations of propeller thrust. It was also less costly and was better lubricated than the external thrust. With regard to the risk of the thrust pads running out and the bronze backs running on the thrust collar, it was now usual to fit a warning device of the failure of oil to the thrust pads, also, it was now the practice at Doxford's to fit a steel plate to separate the thrust chamber from the engine crankcase. The six-cylinder turbocharged engines would now run reliably both ahead and astern down to 25 r.p.m. but the four-cylinder engines required some assistance from a small motor-driven fan to provide good acceleration. It was possible by this means to accelerate the engine from 25 r.p.m. to full speed in under one minute, with the propeller fully immersed, but it was very rare that such an eventuality was necessary. The turbocharged engine would not give full torque astern but it was possible to run at 95 r.p.m. astern for long periods and up to 100 r.p.m. or more for a few minutes, though most ships were unable to withstand such high engine speeds astern due to the vibrations excited by the propeller. Mr. Jackson could not see why Mr. Victory was so critical of the fusible pellets which Doxford's built into the studs of the pistons and cylinder liners in order to indicate when there had been high temperatures. This was not "a built-in exonerator" to provide a "good get-out" but it did give a good indication of the truth and without it the engine builder was always held to blame; defective operation was never acknowledged.

Mr. McNaught's remarks were very much to the point. Mr. Jackson stated that he personally had never seen a scavenge fire but he had heard vivid descriptions of them and had done everything possible to eliminate their occurrence on the Doxford engine. For example, the opening of the exhaust ports had been advanced so that the exhaust gases were evacuated from the cylinder before the scavenge ports opened. This was a necessity since it was the main cause of the ignition of any sludge in the entablature. The main cause of the accumulation of such sludge was bad combustion, particularly with high viscosity fuels and this could hardly be detected by a difference in fuel consumption since he had known an increase in fuel consumption of less than 1 per cent cause large quantities of sludge. This sludge was generally due to the fuel injection being late or defective spraying through defective injectors, insufficient heating of the heavy fuel, or injection at too low a pressure. He did agree with Mr. Brown's written contribution that piston blow-past, due to badly worn rings and enlarged ring gaps, could be the cause of ignition of the sludge but, in general, he would consider that it was blow-back of the exhaust through the scavenge ports which was responsible for starting a fire, due to the exhaust ports not opening early enough or due to some blockage in the exhaust system such as the safety grids being choked or due to very late injection and late combustion. With regard to the timed cylinder lubrication, the degree of lag depended upon the type of pump, the length of pipe between the pump and the quill, and also on the viscosity of the lubricating oil. Doxford's measurements had indicated a lag varying between about 5 deg. and as much as 16 deg. under adverse conditions. There were no hand oiling points on the Doxford P or J type engines and only about six grease cups, but, on the other hand, it was necessary to ensure that the lubricating oil tanks supplying the lubricators were kept full and, similarly, the troughs of the timing valves.

Mr. Jackson said that he fully agreed with the remarks of Mr. Worn and his comparisons of the single turboblower versus the twin turboblower on the four-cylinder engine. He would like to have seen a greater use made of the single turboblower system but, unfortunately, most superintendents preferred to have two turboblowers on an engine because of the greater manœuvrability and increased reliability should one turboblower break down.

In reply to Mr. Taylor, Mr. Jackson stated that in general he had found the liner wear of large engines to be reduced with chromium bores to about one-half when using high viscosity fuels. On the other hand, the cost of chromium plating almost doubled the cost of the liner. He had not found Series 2 type of detergent oil to be very beneficial but alkaline additive oils were of considerable benefit in reducing cylinder liner wear. Mr. Jackson assured Mr. Menon that the maintenance of Doxford engines was under continuous study and improvement and means were being developed for improving the tightening up of the very large nuts which were now used for the side rods and side connecting rod bottom ends. Of the various methods being developed, he personally considered that the heating of the top ends of the side rods and side connecting rod bottom end bolts by means of an electric element was the simplest and easiest method. With regard to the question raised by Professor Chambers, Mr. Jackson felt that the time was not yet opportune for large engines, of turbocharged type, to be installed in ships without prior testing in the engine works. The normal aspirated Doxford engine had reached a stage of development and manufacture such that it could be placed on test on the afternoon of one day, the owners' trials taking place on the following day and then the engine was dismantled. He knew that it was common practice on the Continent to install engines in ships without prior building and testing in the works—for which £1,000 extra was asked—but this only transferred the cost from the engine builder to the shipbuilder since installations took longer and he had heard of cases of quay trials taking three days, whereas with prior testing the average Doxford quay trial was not more than four to six hours. Mr. Jackson thanked Mr. Archer for his kind remarks and for his support in his capacity at Lloyd's, of approving plans and components of new type engines. Mr. Archer was correct in assuming that the advancing of the exhaust openings had also increased the area of the ports from 17·3 sq. in./cu. ft. of cylinder volume, in the case of the *Escort* engine, to 21·3 sq. in. on later engines of similar strokes. While Mr. Jackson had feared that there would be an adverse effect on both power and fuel consumption by the progressive robbing of the expansion stroke in the cylinder in order to feed the turbine with exhaust gases at a higher pressure, this had not been the case, due, presumably, to the better combustion from the larger amount of air being supplied. It was suspected that the greater rate of wear of the lower liners of the *Montana* was due to the condensation of the sulphurus products of combustion on the cooler cylinder walls and experiments were in hand to increase the temperature of the lower jackets by reversing the flow of the cooling water, i.e. taking the hot water out from the low end of the lower liner. He appreciated Mr. Archer's remarks regarding the alignment of the upper piston and transverse beam and there had been slight fretting of the spherical pads. This fretting had, however, been in the nature of an oily red paste and had been very slight. The calculated torsional characteristics of the 85PT6 engine had been made assuming a detuner, since the 9th order shown was of the three node type, whereas without a detuner it would be of the two node type. The 9th order two node vibration was of a considerably higher magnitude than that of the 9th order three node with a detuner and, as Mr. Archer remarked, it was not possible to bring this order below 100 r.p.m. in view of the 7th also being of considerable magnitude. The performance curves and mechanical efficiencies were based on normal indicator diagrams taken by the normal indicator gear fitted to the engine, but these had been checked by a Farnborough indicator on the first *Montana* engine and the accuracy of these in comparison with the Farnborough results had been proved on a number of occasions. However, the mechanical efficiency figures given were not likely to be accurate within closer limits than three per cent.

With regard to Mr. Beattie's question concerning the earlier application of turboblowers to the four-cycle engine, and the relatively late application to two-cycle engines, this was entirely because the turboblower had a more onerous duty to perform with a two-cycle engine in that it had not only to supply the scavenging air, but had to supply this air at a higher pressure than the exhaust back pressure, in order to clear out the cylinder, whereas, on the four-cycle engine, the cylinder was cleared out by the upward exhaust stroke of the piston. The early turboblowers were not so efficient and it was not until after World War II that the higher efficiency turboblower, in conjunction with an intercooler, had given the performance necessary to enable the two-cycle engine to be turbocharged. It was possible to turbocharge the two-cycle engine to a higher degree than at present, but Mr. Jackson doubted whether it was wise to apply higher pressures as yet until more experience had been gained and, in any case, higher mean pressures meant higher maximum pressures which would increase the size of the crankshaft and running gear on a given size of engine. On the early Doxford turbocharged engines, the scavenge pump had been put in series with the turboblowers, but after the first three engines, it had been found possible to eliminate the scavenge pumps which was a considerable simplification and reduced the cost of the engine. The independent blower used for manœuvring purposes was a small single-stage fan requiring a motor of no more than 30 h.p. to drive it, whereas, the separate electrically driven scavenge pumps mentioned by Mr. Beattie would require motors exceeding 100 h.p. to drive them. There would be a considerable drop in the overall efficiency of the engine due to this. Mr. Jackson confirmed that the elimination of the spherical bearings, in conjunction with the more rigid crankshaft with large diameter main bearings, had reduced the deflexions between the webs to less than one-third of the deflexions of the older type of Doxford crankshaft. All transverse beams of present engines were made of steel forgings and there were now no failures due to fatigue of the cast steel transverse beams. Crankcase corrosion had been completely eliminated as a result of fitting a gland or diaphragm between the engine cylinders and the crankcase. Fractures at the main bearing pockets of fabricated bedplates were not entirely due to the fact that they were not stress relieved after welding but, nevertheless, the new design of bedplate, in conjunction with stress relieving, had so far removed this type of failure.

Mr. Jackson much appreciated the kind remarks of Mr. Bennett who had many Doxford engines under his care. Mr. Jackson agreed that the first cost of Diesel plant was no greater than that of modern steam turbine plants but was, however, greater than that of the reciprocating steam engine. Doxford engines would run and manœuvre satisfactorily on high viscosity fuels and the engines had been tested in the works on these fuels on many occasions. It was, however, difficult to obtain fuels of higher than 1,000 sec. viscosity and fuels of 3,000 sec. had to be obtained specially and the transport cost was very high. Mr. Bennett's observations on the advantages of manœuvring on high viscosity fuels were appreciated. He also asked regarding crankshaft deflexions. It was quite common on the older engine, with spherical bearings, to have deflexions of the centre crankwebs of 0·025in. and even more than 0·040in. had been known and the engine had run satisfactorily for years. On the other hand, the more rigid crankshaft of the P type engine, mounted in large diameter short bearings, had deflexions of no more than 0·008in. on the worst cylinder.

Mr. Jackson stated that he did not know of any case where it had been necessary to use an inhibited oil for the piston cooling on Doxford engines, and there had been no cases of carbonization in the cooling spaces in the piston heads. With present day oil cooling systems and the type of glands being used on telescopic pipes, there was such leakage from the piston cooling system to the lubricating oil system that it was not possible to separate these systems.

The Doxford turbocharged engine was very quiet and it was possible to conduct a conversation in the vicinity of the turbochargers without any discomfort. The common rail type of injection system was also very quiet and there was no noisy valve gear.

With regard to the written contribution from Mr. Andresen, the exhaust temperatures before the turbine were higher on certain cylinders, e.g. the middle cylinders of the four-cylinder engine, due to the fact that the exhausts from these cylinders were joined together and thus there were double the number of exhaust pulses passing the exhaust pyrometers than on the other cylinders. Since this date, the pyrometers had been moved to a position in each individual exhaust belt where the flow from the adjacent cylinder did not interfere and the resulting exhaust temperatures were as given for the six-cylinder engine, i.e. about 600 deg. F. at full load. The pyrometers in these positions passed through the water cooled exhaust belt into the exhaust passage.

With regard to the crankshaft of the nine-cylinder J type engine, this was of the semi-built type but had crankpins in accordance with Doxford's normal construction, and not eccentrics. The crankpin was designed for an m.i.p. of 9·5 kg./sq. cm. or 135lb./sq. in., with a maximum pressure of 900lb./sq. in. It was intended, however, that the engine should be capable of delivering the full rated output continuously at sea in normal service with an overload capacity of 10 per cent for sea trials and similar conditions, and 15 per cent overload for one hour.

With regard to the weight of the engine, it was usual for Doxford engines to weigh about four times the weight of the crankshaft and the company was quite satisfied that its estimated weights for the nine-cylinder engine would be proved accurate, e.g. the bedplate was estimated to weigh 60 tons and it was turning out to be within one ton of this figure. The crankshaft was the heaviest part of the Doxford engine and the structure of the engine was light compared to other engines due to the fact that it did not have to carry any combustion loads.

In conclusion, Mr. Jackson expressed his appreciation for the opportunity to read his paper in company with Mr. Andresen, who presented his paper on the well known B. and W. engine.

Mr. Jackson also once again thanked the many contributors to the discussion for their kind remarks and good wishes for the success of the Doxford P and J type engines.

Author's Reply

Reply by Mr. Andresen

Mr. Andresen wished to thank all those who had contributed to the discussion thereby providing information about experiences and results of investigations from research and service, information which was very important and useful in the endeavour to further improve and develop the Diesel engines. He also expressed his thanks both for the criticisms and for the encouraging remarks—including those from two old friends.

Mr. Pemberton had mentioned that an increase of power per cylinder meant fewer cylinders and that a further advance in pressure charging might present difficulties in forges and foundries with, for example, the heavy crankshaft parts. If however, the power range of the existing engines was regarded as sufficient a further increase of the mean pressure might be used to reduce the cylinder dimension for a given power, which would result in a small reduction of the weights of the heavy parts.

Mr. Yamaguchi's information regarding the heavy oil of poorer quality than expected for the viscosity given, seemed to indicate that the oil was a mixture of a very heavy oil with correspondingly poor qualities with one or another light oil to reduce the viscosity. This showed that engine builders should never indicate any specification with maximum figures for fuel oils which their engines were able to use, even disregarding the viscosity. The viscosity should only be used to decide the capacity of the fuel oil heater. To Mr. Yamaguchi's question regarding chromium plated liners, the answer was that the wear rate was generally reduced by about 50 per cent as long as the chrome was not worn through, but when that happened the wear rate would increase and this was the reason for using the thicker layer of chrome. It should however, in this connexion, be mentioned that the introduction of the neutralizing cylinder oils, which had reduced the rate of wear considerably, had made the owners less interested in chromium plating and this was an advantage to the engine builders who thus avoided the risk of delayed deliveries owing to the limited facilities for chromium plating and also the corresponding interruptions in their production programme.

Regarding the use of controllable pitch propellers in conjunction with large high powered Diesel engines, it was the author's opinion that such arrangements were only of interest in regard to special services, for example, for ships calling at small ports where manœuvring was difficult. In such cases the owners preferred to have the normal reversing equipment on the engines, not only as a safety precaution but to be used in connexion with the controllable pitch propeller thus ensuring that the side thrust on the ship could be changed from starboard side to port side or *vice versa,* when getting in and out of a small port without any help from tugs. Several engines for such service had already been delivered, but it meant no simplification of the engines which furthermore, in the case of turbocharging needed auxiliary blowers for running with the propeller blades in zero position.

Mr. Pounder had asked why transverse ties were now normally being arranged from the engine top to the deck of the ship as this had only been necessary in a very few cases. It was true that stiffening of the engines was not generally necessary, but in case of resonances between the guide pressure and the frequency of the engine, undesirable movements of the engine top might occur and at the same time increase the impulses which might incur vibrations in the ship's accommodation. The engines might move as a unit, or for a long engine with, for instance, nine cylinders, the two halves of the engine might move in opposition to each other. When the turbocharged engines were introduced at the same time the top stiffenings were introduced as a standard fitting and accepted as an advantage for the roller bearings in the turbochargers, even by owners and shipbuilders who, had not hitherto liked them. The top stiffenings which should only stand very small forces, were simple, inexpensive and completely prevented transverse vibrations. For twin-screw installations each engine should be tied to the deck, but not together.

Mr. Andresen thanked Professor Aertssen for his interesting information which confirmed the interest and support that Belgian owners had always given to the engine builders in developing the modern Diesel engines. The results from the turbocharged 7,000 b.h.p. engine in the *Lukuga*: consumption of heavy fuel oil 164 gr./b.h.p. hr., consumption of lubricating oil 0·35 gr./b.h.p. hr. and cylinder wear rate of 0·12 mm./1,000 hr. corresponded well to the good service results obtained with other modern engines.

The author thanked Mr. Alcock for the information regarding his investigations in connexion with the thermal problems in large turbocharged engines. The figures Mr. Alcock had shown corresponded very well to the figures used in Copenhagen.

The steel used for piston crowns was a low alloy Cr Mo steel with less than one per cent Cr + Mo and according to the information available its thermal conductivity was about 38 kcal/m. hr. deg. C. or about 0·105 c.g.s. The heat flux—calculated from the temperatures measured—was about 140,000 cal./sq. m. hr. or about 285,000lb. cal./sq. ft. hr.

The thermal conductance between piston and oil was calculated to 1,000-1,200 kcal/sq. m. hr. deg. C. or about 240lb. cal/sq. ft. hr. deg. C.

The amount of air trapped in the cylinder was not measured but calculated from a scavenging efficiency of about 92 per cent and a temperature of about 350 deg. C. at the beginning of the compression in the cylinders.

In reply to Dr. Brown's remarks regarding opposed piston contra poppet valve engines, the author—with reference to Mr. Pounder's contribution—mentioned that at Burmeister and Wain they were of opinion that each of the two engine types had advantages which would satisfy not only the technical demand for modern high powered Diesel engines, but also the preferences of any owner and superintendent engineer.

The advantages of the opposed piston engine were that they had no cylinder covers and exhaust valves, probably a somewhat shorter time necessary for piston overhaul and in most cases a better balancing with less free moments.

The advantages of the poppet valve engine were the smaller number of pistons and other moving parts, the simple crankshaft and the fact that no auxiliary blower for manœuvring and slow running was needed due to the reversing of the camshaft and the exhaust valve timing.

As mentioned by Dr. Brown, the secondary steam from the two stage boilers was used for cargo oil heating and with this type of boiler there would be no risk of break down of the boiler due to water contaminated by oil. Some years ago this type of boiler was introduced in the Diesel tankers as owners claimed that other types of boilers often gave trouble. The boilers had shown good results in service and today they were delivered for tankers by many firms.

Regarding the development of automation and remote control, reference should be made to the m.s. *Kinkasan Maru,* built in Japan by the Mitsui Shipbuilding and Engineering Co. which was the first large cargo liner with such an installation. For remote starting, stopping, reversing and regulating of the main engine from the bridge, or the control cabinet in the engine room, an electro-hydraulic system was used. All the instruments for the main engine as well as for the auxiliary machinery were concentrated in the control cabinet and a number of them were self-recording. Furthermore a part of the auxiliary machinery was automatically controlled.

Captain Jenks had been kind enough to point out the risk that within the next decade the large Diesel engine would die out like the dinosaur and the whale which already was in danger.

The difficulty with the dinosaur and the whale was that only in the course of a great many generations were they able to adapt themselves to resist better the changes in their environment and the malicious and unthinking acts of man. The large Diesel was, however, in a better position as it was possible within a few years to introduce necessary modifications to suit new conditions and in this task the experience of engineers in such conditional changes would no doubt be helpful.

When a purely economical point of view was available as was general in Europe, the large Diesel engine was an excellent solution, but subsidies in shipbuilding and shipping introduced in many parts of the world might give other prime movers an opportunity irrespective of the economy. In any case the large Diesel engines had for many years been leading in the development towards high power, high efficiency, reliability, use of cheapest fuel, etc., and the results and experience of these engines would without doubt be used to improve the medium and high speed Diesel engines and thus develop them as an alternative to the large Diesel engine.

Replying to Mr. Victory's remarks regarding cylinder lubrication, it should be mentioned that the oil was introduced in the lower part of the cylinder liners just above the scavenging air ports and that lubricators of the Jensen type were used. That the timing—even if not absolutely exact—was useful, was confirmed by the good wear results generally obtained and also by a case where the adjustment of the lubricators on an engine was faulty—correct adjustment on some cylinders, extreme maladjustment on others and varying degrees of maladjustment on the remaining cylinders. The wear rate followed the timing very well, normal, excessive and medium.

Mr. Victory had apparently misunderstood the information given regarding piston temperatures on page 477. The first line gave the temperatures measured in a piston of 740-mm. diameter and the second line the temperatures in the same piston, not measured but calculated from the higher temperature measured in a piston of 840-mm. The results—30 deg. C. reduction by water cooling—corresponded very well to the calculations, when water cooling was compared to an effective oil cooling.

In reply to Mr. McNaught's remarks regarding scavenge fires, Mr. Andresen said that it should be mentioned that the two important design features in this respect were the pre-exhaust period and the scavenging air pressure which should correspond in such a way that blow back into the scavenging air box was avoided. For new engines tested in the shop or on trials, blow back would not occur when the pre-exhaust period and the scavenging air pressure corresponded well to each other and blow back would then only occur when the piston rings were worn, broken or sticking, or if the cylinder liners were worn out. For opposed piston engines it was important that the advance of the exhaust piston to the main piston was sufficient and that the port heights and the setting of the pistons were correct to obtain a sufficient pre-exhaust period. Regarding the question of cylinder lubrication, this was answered in the author's reply to Mr. Victory.

Mr. Taylor had mentioned that a reduction of the cylinder wear in a 390 mm. bore engine to one quarter of its previous value had been obtained by using alkaline additive types of lubricating oil. In large slow running engines similar results had been obtained—the rate of reduction depending, however, on the quality of fuel oil used. Regarding chromium plating of cylinder liners this was referred to in his reply to Mr. Yamaguchi.

Mr. Andresen well recalled the difficulties of about twenty-five years ago to which Mr. Arnold had referred, with regard to cylinder covers of 13 per cent Cr steel. It had then been very difficult to both cast and to apply correct heat treatment to these cylinder covers. These were now of a more simple design and made of Cr Mo steel which was easier to cast and heat treat. It was very seldom that cracks occurred in these covers which were entirely satisfactory.

The exhaust valves did not stick and therefore the exhaust cams did not receive any wear or undue impact neither was wear of fuel pump cams any problem.

An hydraulic system for operating the exhaust valves, as suggested by Mr. Arnold, had been under consideration, but the advantage of such a system compared to the simple existing system was not certain enough for its introduction as yet since in any case a shaft was needed for the indicator gear, the cylinder lubricating pumps, etc.

In reply to Professor Chamber's question regarding the building up of the engines in the shop, it should be mentioned that it was necessary to erect the larger parts from the bedplate and crankshaft to the cylinder cover, in the engine builder's shop, as it would be more expensive to do this in the various yards to which the engines were delivered. To complete the engine entirely and have it tested in the shop was, of course, not a necessity, but it was thought to be cheaper for the owner and also for the shipyard which normally did not have the same facilities to do any work which might be necessary during the first testing of the engine.

Mr. Archer had suggested that crankcase explosions seemed to occur more often today than in the earlier days of water cooling. However, according to information from a classification society the difference between the number of explosions in engines with oil cooling and those with water cooling was so small that the relative number of explosions in water cooled engines was higher when the small number of these engines was taken into account. Mr. Archer had mentioned a case of stalling of the centre blower in a nine-cylinder engine with three blowers. The reason for this had probably been that the loading of the three centre cylinders had been lower than that on the others, whereby the blower was working to near the stalling limit which might eventually give rise to stalling in connexion with an uneven load between the three cylinders.

The information given by Mr. Storm regarding the noise measurements on board 15 ships was most interesting and the author agreed that a further reduction of the noise level on board should be aimed at for new ships and especially for the manœuvring platforms and the workshops where the personnel remained for longer periods.

The main purpose of the shaft-driven lubricating oil pump and the hydraulically-driven cooling water pumps to which Commander Hall had referred, was that the ship in normal service at sea only needed to have the turbo-generator running on steam from the exhaust gas boiler. The capacity of the pump was about 500 sq. m./hr., which was normal for the lubricating oil pump for the main engine in question. The efficiency of the drive was of minor importance and was estimated to be about the same as that of electrically-driven pumps for which the efficiency depended on that of the generator and the electric motors.

In answer to Mr. Parodi's question regarding the exhaust gas boiler, Mr. Andresen stated that the total back pressure for the main engine should preferably be below 250 mm. maximum 300 mm. water column. For the watertube boilers used, sufficient evaporating surface for the steam production mentioned was obtained without the back pressure being too high and thus corresponded very well with the calculation, the thermal conductivity was about 85-90 kcal/sq. m. hr. deg. C.