# The Use of Medium-speed Geared Diesel Engines for Ocean-going Merchant Ship Propulsion

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Geared medium-speed Diesel engines under development have been examined in the context of propulsion plant of 16 000 bhp for a bulk carrier and a fast cargo liner. Economic and technical comparisons have been made primarily with the slow-speed direct-drive Diesel engine, but some comparisons with existing medium-speed Diesel engines are also included.

The study shows that a ship fitted with the medium-speed geared Diesel plant offers the shipowner the possibility of an economic advantage over a ship driven by a slow-speed direct-drive Diesel engine.

The economic advantage can be achieved if the following conditions are observed:

a) the cost of the medium-speed Diesel machinery remains significantly below that of the slow-speed Diesel engine;
b) the projected engines operate reliably on heavy fuel without imposing an undue maintenance load on the shipowner;
c) advantage is taken of the existence of the gearing to ensure that the optimum propeller size and propeller rev/min are selected;
d) the machinery installation design and the design of the ship itself, are carefully tailored to take the maximum advantage of the smaller size and weight of the medium-speed geared Diesel machinery to give a smaller and cheaper ship.

## INTRODUCTION

Diesel engines now predominate as prime movers for the propulsion of ocean-going merchant ships. Currently these engines are usually of the slow-speed direct-drive type, and development has been gradually resulting in larger outputs per engine and in reduced specific weight and size. This trend of gradual improvement is likely to continue, but there are no indications of any projected major breakthrough in the near future with this type of engine.

An alternative to the use of a slow-speed direct-drive Diesel engine for main propulsion is to adopt medium-speed Diesel engines, one or more of which can be geared to the propeller shaft. In recent years, an increasing number of merchant ships have been fitted with medium-speed geared Diesel engine propulsion and a sizable body of service experience has been accumulating.

In an endeavour to assess the potential economic and technical advantages of medium-speed Diesel engines for the propulsion of ocean-going merchant ships, as compared with the use of slow-speed direct-drive Diesel engines, the Ministry of Technology recently placed a study contract with the Yarrow-Admiralty Research Department, and this paper is based on the results of that study.

The paper has been related in particular to two British two-stroke medium-speed engines—the Ruston and Hornsby AO engine and the Mirrlees National opposed-piston engine—which are still being developed. However, comparisons are provided also with other medium-speed engines available now, including the four-stroke Mirrlees National K Major and the S.E.M.T. Pielstick PC2 engines, as shown in Table I.

With the engine no longer connected directly to the propeller, considerable freedom exists in choosing the engine speed. However:

i) if the speed is too low, it is unlikely that savings in weight, bulk and cost of the engine will outweigh the margin required for the gearing;
ii) if the speed is too high, burning of heavy fuel is likely to present difficult problems.

As can be seen from Table I, the engine makers have generally picked speeds in the 400 to 600 rev/min bracket.

The existence of the gearing also permits the freedom of choice of the propeller rev/min, which can be based on economic considerations as applicable to the particular ship in question.

## BASIS

It is not enough to examine the various technical and economic aspects of the medium-speed engines alone. Nor is it enough to compare such engines with the slow-speed Diesel. To permit meaningful conclusions to be drawn, it is necessary to consider and compare main propulsion machinery installations using these two types of prime mover in the context of several carefully selected ship types.

The power level at which to make the main comparison has been taken as 16 000 bhp, which is the maximum that can be provided by two Ruston and Hornsby AO engines, at the present stage of their development, geared to a single propeller. It should be noted, however, that the Mirrlees National opposed-piston engine will be capable of providing much greater powers than this, with either a single or a multi-engined installation.

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TABLE I—COMPARISON OF MEDIUM-SPEED DIESEL ENGINES WITH POWERS OF 8000 BHP AND ABOVE

| Item | | Two-stroke | | | | | Four-stroke | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| | | Valve in head | | | Opposed-piston | | | | | |
| Maker | | Ruston and Hornsby | Sulzer | Mitsubishi | Mirrlees National | Fairbanks Morse | SEMT-Pielstick | Mirrlees Nat. | M.A.N. | Werkspoor |
| Designation | | AO | Z40/48 | UET52/65 | OP | 38A20 | PC2 | K Major | V 40/54 | TM 410 |
| Bore | in/mm | 14·25/362 | 15·748/400 | 20·47/520 | 15/381 | 20 & 10/508 & 254 | 15·748/400 | 15/381 | 15·748/400 | 16·142/410 |
| Stroke | in/mm | 18·5/470 | 18·898/480 | 25·59/650 | 2×15/2×381 | 21·5 & 10·75/546 & 273 | 18·11/460 | 18/457 | 21·26/540 | 18·50/470 |
| Stroke/bore ratio | | 1·3 | 1·2 | 1·25 | 1 + 1 | 1·075+1·075 | 1·15 | 1·2 | 1·35 | 1·12 |
| Swept volume per cylinder | in$^3$ | 2950 | 3680 | 8420 | 5300 | 7599 | 3530 | 3180 | 4140 | 3785 |
| Date first engine available for marine service | | 1967 in line<br>1968 vee | 1965 in line | 1961 | 1969 | 1965 | 1962 | 1965 | 1964 | 1968 |
| Maximum continuous output/cylinder(1) | Bhp | 500* | 591 | 740 | 1250* | 1000 | 448* | 445* | 530 | 493 |
| Corresponding speed | Rev/min | 450 | 445 | 300 | 600 | 400 | 465–500 | 525 | 400 | 500 |
| B.m.e.p. | lb/in$^2$ | 150 | 143 | 116 | 156 | 130 | 216 | 211 | 254 | 206 |
| Mean piston speed | ft/min | 1390 | 1400 | 1280 | 1500 | 1433/717 | 1400–1510 | 1576 | 1420 | 1540 |
| Maximum cylinder pressure | lb/in$^2$ | 1500 | — | 920 | 1500 | 1050 | 1280–1350 | 1400 | 1635 | 1705 |
| No. of cylinders available | In line | 6, 8, 9 | 6, 8, 9, 10, 12 | 6, 7, 8, 9, 12 | 8, 12, 16 Double bank | 6, 9 | 6, 8, 9 | 6, 7, 8, 9, | 6, 7, 8, 9, | 6, 8. 9 |
| | Vee | 12, 16 | 8, 12, 16, 18 | — | — | 12, 18 | 8, 10, 12, 14, 16, 18 | 12, 14, 16, 18 | 10, 12, 14, 16 | 12, 16, 18 |
| Maximum continuous power output | Bhp | 8000 | 10 640 | 8870 | 20 000 | 18 000 | 8060 | 8000 | 8480 | 9000 |
| Engine weight | Tons | 58 | 91 | 122 | 150 | | 74·5 | 95 | 93·5 | 110 |
| Specific weight | lb/bhp | 16·25 | 19·2 | 30·8 | 16·8 | | 20·7 | 26·6 | 24·7 | 27 |
| For 8000 bhp: | | | | | | | | | | |
| No. of cylinders | | 16 | 16 | 12 | 8 | 9 | 18 | 18 | 16 | 16 |
| Power output/cylinder | Bhp | 500 | 500 | 667 | 1000 | 890 | 445 | 445 | 500 | 500 |
| Engine speed | Rev/min | 450 | 430 | 300 | 500 | 400 | 465–500 | 525 | 400 | 500 |
| B.m.e.p. | lb/in$^2$ | 150 | 125 | 105 | 150 | 116 | 214–199 | 211 | 240 | 206 |
| Mean piston speed | ft/min | 1390 | 1355 | 1280 | 1250 | 1433/717 | 1400–1510 | 1576 | 1420 | 1540 |
| Engine weight | Tons | 58 | 82 | 122 | 90 | 93·8 | 74·5 | 95 | 93·5 | 100 |
| Overall length | | 18 ft 1 in | 25 ft 9in | 39 ft 2½ in | 18 ft 4 in | 36 ft 6 in | 31 ft 7 in | 32 ft 7 in | 30 ft 0 in | 25 ft 11 in |
| Width | | 8 ft 11 in | 10 ft 4½in | 5 ft 10½ in | 15 ft 6 in | 8 ft 6¾ in | 10 ft 9 in | 12 ft 10 in | 11 ft 10 in | 12 ft 0 in |
| Overall height | | 16 ft 0 in | 13 ft 6½ in | 12 ft 10 in | 15 ft 8¾ in | 16 ft 4 in | 10 ft 7 in | 12 ft 9 in | 13 ft 9 in | 12 ft 6 in |
| Crane hook height | | within engine height | 14 ft 0 in | 17 ft 3½ in | within engine height | 18 ft 11 in | 12 ft 3 in | 14 ft 9 in | 15 ft 3 in | — |
| Specific fuel consumption(2) | lb/bhp-h | 0·338 | 0·356 | 0·349 | — | 0·352 | 0·342 | 0·335 | 0·336 | 0·34 |

*Notes:*

(1) Makers' quoted continuous outputs for merchant ship propulsion under temperate conditions.
*These engines rated in accordance with B.S. 649:1958, i.e. with air intake temperature of 85°F, sea-water temperature to charge air coolers of 75°F. However, for some engines the output is also available under tropical conditions.

(2) Specific fuel consumptions are typical figures based on operation on Diesel fuel.

It is also of interest that some already-developed medium-speed Diesel engines could provide the 16 000 bhp, for example two Mirrlees National K Major 18-cylinder engines.

The ships selected to serve as vehicles for the comparison have been a 16-knot bulk carrier and a 20-knot cargo liner. In the first instance each of these ships has been assumed to be powered by a slow-speed direct-drive Diesel engine having a service power of 14 400 bhp, i.e. 90 per cent, of 16 000 bhp. On this basis the particulars of these ships are as follows:

| | Bulk carrier | Cargo liner |
|---|---|---|
| Length, bp, ft | 750 | 500 |
| Breadth moulded, ft | 105 | 70 |
| Depth, moulded, ft | 56 | 43 |
| Deep draught, ft | 38·5 | 31 |
| Deep displacement, tons | 68 700 | 18 450 |
| Deadweight, tons | 55 000 | 12 000 |
| Cargo capacity, $ft^3$ | 2 700 000 | 665 000 |
| Nominal speed, knots | 16 | 20 |
| Service power, bhp | 14 400 | 14 400 |

It has been assumed that the above are the service speeds and cargo capacities required by the shipowner. For the purposes of economic comparison, therefore, account has been taken of differences in the following:

a) machinery space;
b) machinery weight;
c) bunker weight;

in estimating the dimensions of a reduced size of ship to give the same cargo capacity for the medium-speed engined ships as for the slow-speed engined ships. Only the saving in space below the main deck has been taken into account in reducing the ship sizes.

It has been assumed that electrical power is provided by Diesel generators, on the basis that one is under maintenance, one is standby and the remaining set or sets are running.

The estimated electric loads are as follows:

| Electrical load | Bulk carrier | Cargo liner |
|---|---|---|
| Slow-speed engine, kw | 810 | 575 |
| Medium-speed engine, kw | 660 | 425 |

The medium-speed engines are assumed to have engine-driven lubricating oil pumps, which accounts in part for the reduction in generator capacity for the medium-speed engined ships.

The following figures have been used in the study:

| | Bulk carrier | Cargo liner |
|---|---|---|
| Days at sea per annum | 300 | 160 |
| Fuel stowage (days) | 40 | 30 |
| L.O. stowage (days) plus one oil change | 40 | 30 |
| *Fuel burned*: | | |
| *Main Engine*: | | |
| Marine fuel oil | 3500 sec Redwood 1 at 100°F | |
| Net calorific value | 17 000 Btu/lb | |
| Price | £5 per ton | |
| *Diesel Generators*: | | |
| Diesel fuel | | |
| Net calorific value | 18 000 Btu/lb | |
| Price | £9.10.0 per ton | |

A total interest rate of ten per cent has been assumed to allow for depreciation, interest and insurance. This is a nominal figure, and the economic comparison introduced later has been generalized to permit easy adjustment of this rate to other figures, as might be appropriate in any particular case.

## ENGINES

Table I gives the principal details of the Ruston and Hornsby AO and of the Mirrlees National OP engines in comparison with existing medium-speed engines, including the Mirrlees National K Major. The tables includes only engines which are capable of an output of 8000 bhp or more when burning heavy fuel, and which could therefore be used for propulsion of the two ships considered in detail in this study.

To serve as a yardstick for comparison, a seven-cylinder Burmeister and Wain 84-VT2BF-180 has been chosen for the slow-speed engine installation.

Fig. 1 shows overall arrangements of the projected engines and of existing medium-speed engines complete with gearing for a total power of 16 000 bhp. It can be seen that the projected and existing medium-speed engines offer a significant reduction in length. An outline of the seven-cylinder Burmeister and Wain 84-VT2BF-180 engine has also been included in this figure.

Table II provides a tabular comparison of the projected engines with existing medium-speed engines, complete with gearing and related to an installed power of 16 000 bhp per shaft, and with selected slow-speed engines giving the same bhp, viz the B. and W. engine selected as basis of comparison, and the Doxford 76J7 engine.

### *Ruston and Hornsby AO Engine*

The AO engine[1] is a through-scavenged, valve in head, two-stroke cycle engine with cylinders arranged in vertical in-line or 50° vee-form. For marine propulsion it may be supplied in direct reversing form.

The engine frame is of lattice-work form, approaching a space frame in concept and in which stresses and deflexions can be accurately predicted. It comprises cast steel members welded together in such a way that butt welds are used throughout and the maximum to mean stress ratio across the section of the weld approaches unity. All welds are designed for a definite fatigue life in excess of 25 years.

The cast-iron cylinder head has a replaceable steel flame plate containing coolant passages which give controlled flow around the injector and the exhaust valve seats. There are four direct-seating exhaust valves which are operated by twin cams and push-rods.

The piston has a steel crown, carrying three compression rings, and a cast-iron skirt carrying a further compression ring and two scraper rings. The crown is cooled by lubricating oil which is forced to follow a positive path and the supply and return is by means of telescopic tubes.

The single-piece cylinder liner is porous-chrome plated and has a timed system of lubrication, situated above the port belt in order to reduce wear rates, particularly when burning heavy fuel.

In the vee-engine, the connecting-rod large ends are of fork and blade construction, enabling a 21-in cylinder centre distance to be obtained.

For the particular installation considered, two 16-cylinder vee-type engines will be required. Each engine will have four turbochargers mounted on the top of the engine, between the banks, and these will supply the air required by the engine under all operating conditions.

The lubricating oil and the fresh water pumps may be engine-driven if desired.

The AO engine has a competitive fuel consumption, but initially may have a slightly higher lubricating-oil consumption than its four-stroke cycle competitors.

While this paper is orientated towards the engining of merchant ships, it is relevant to consider the activities of the Ministry of Defence (Navy) in respect of ship propulsion by medium-speed geared Diesel engines. The slow-speed cathedral-type engine is too big and too heavy for use in warships, but the Navy has ten years operational experience with a class of frigates powered by eight ASRI engines geared to two propeller shafts. Today the Navy is actively interested in the prospective use of the current British designs of geared medium-speed engines for warship propulsion, and has ordered a 12-cylinder vee Ruston and Hornsby AO engine for testing to establish:

1) the rating of the engine for naval propulsion purposes;
2) satisfactory endurance running using Diesel fuel;
3) appropriate installation arrangements, including simulations of manoeuvring procedures.

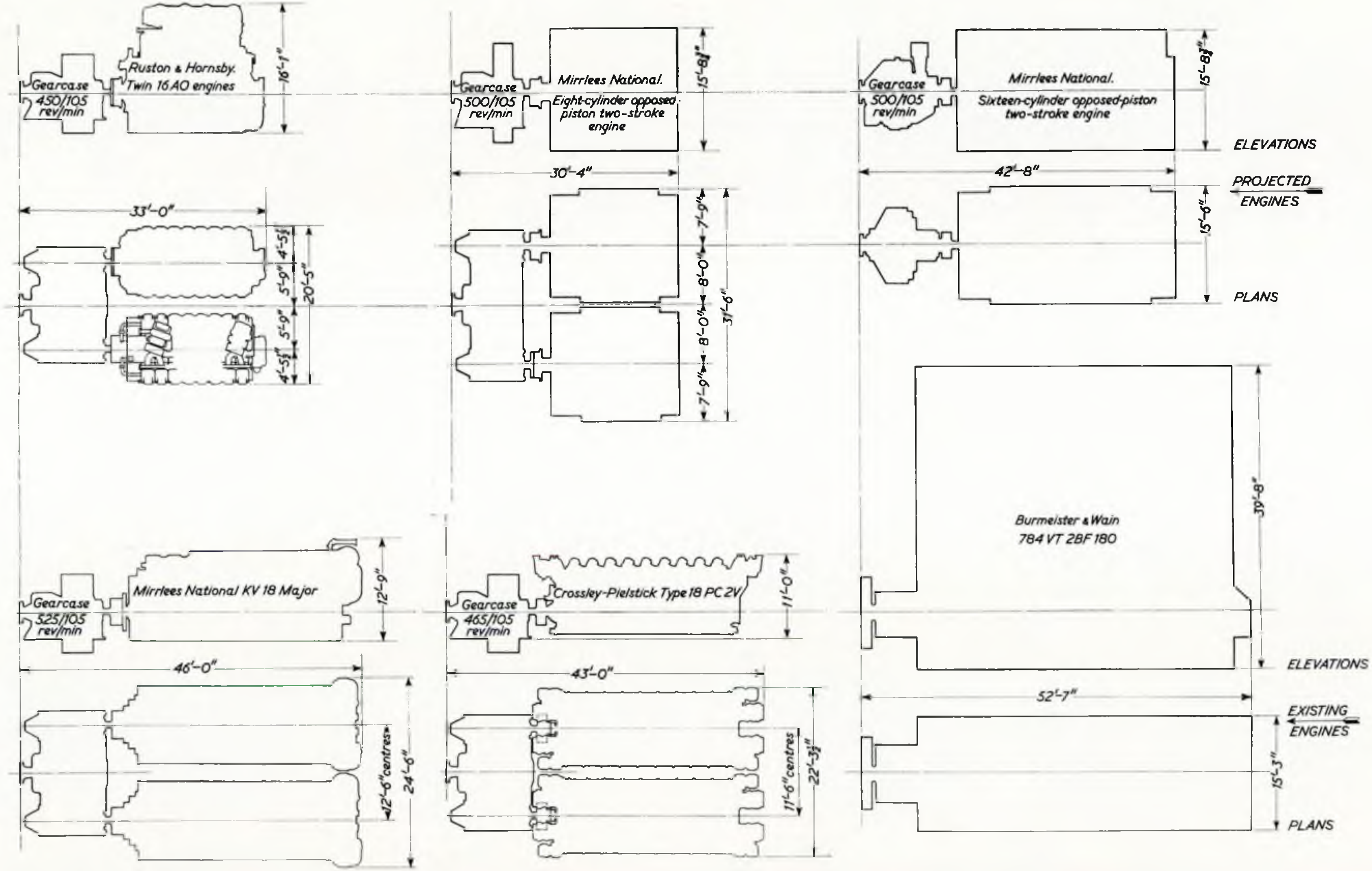



FIG. 1—*Overall arrangements of projected engines and existing medium-speed engines*

TABLE II—COMPARISON OF GEARED MEDIUM-SPEED AND SLOW-SPEED DIESEL ENGINES FOR 16 000 BHP

| | Medium-speed geared | | | | Slow-speed | |
|---|---|---|---|---|---|---|
| Maker | R. and H. | Mirrlees | Mirrlees | Mirrlees | B. and W. | Wm. Doxford |
| Designation | 16AO | OP8 | KV Major 18 | OP16 | 784VT2BF180 | 76J7 |
| No. of engines for 16 000 bhp | 2 | 2 | 2 | 1 | 1 | 1 |
| Engine type | Valve in head Two-stroke | Opposed-piston Two-stroke | Four-stroke | Opposed-piston Two-stroke | Valve in head Two-stroke | Opposed-piston Two-stroke |
| Cylinder bore, in | 14·25 | 15 | 15 | 15 | 33·07 | 29·92 |
| Stroke, in | 18·5 | 15/15 | 18 | 15/15 | 70·86 | 20·47/65·35 |
| Maximum continuous output/cylinder, bhp | 500 | 1000 | 445 | 1000 | 2270 | 2500 |
| Corresponding engine speed, rev/min | 450 | 500 | 525 | 500 | 114 | 119 |
| Corresponding b.m.e.p., $lb/in^2$ | 150 | 150 | 211 | 150 | 142 | 138 |
| Mean piston speed, ft/min | 1390 | 1500 | 1576 | 1500 | 1196 | 406/1295 |
| No. of cylinders for 16 000 bhp | 32 | 16 | 36 | 16 | 7 | 7 |
| Cylinder arrangement | Vee | Double bank | Vee | Double bank | In line | In line |
| Engine weight (wet), tons | 2×63 | 2×94·5 | 2×97·5 | 155 | 615 | — |
| Gearbox weight (including couplings) wet, tons | 67* | 71* | 67* | 28* | — | — |
| Total weight including oil and water, tons | 193* | 260* | 262* | 183* | 621 | 530 |
| Length of installation | 33 ft 0 in* | 30 ft 4 in* | 46 ft 0 in* | 42 ft 8 in* | 52 ft 7 in | 48 ft 6 in |
| Overall width | 20 ft 5 in | 31 ft 6 in | 24 ft 0 in | 15 ft 6 in | 15 ft 3 in | 13 ft 0 in† |
| Overall height | 16 ft 1 in | 15 ft 8$\frac{3}{4}$ in | 12 ft 0 in | 15 ft 8$\frac{3}{4}$ in | 39 ft 8 in | 33 ft 9 in |
| Crane hook height | Within overall height | Within overall height | 14 ft 9 in | Within overall height | 42 ft 0 in | 38 ft 2 in |
| Gearbox centres | 11 ft 6 in | 16 ft 0 in | 12 ft 6 in | — | — | — |

*Based on propeller speed of 105 rev/min. †Width over bedplate.

This order has had the effect of accelerating the development of the vee-engine and initial running has already been carried out on the engine in a six-cylinder form. Plans were made to start the tests of the complete 12-cylinder engine at Lincoln in January 1967 and to continue them for a period of about 12 months. Fig. 2 shows the 12-cylinder engine under construction and Fig. 3 shows it on the test bed.

Of particular interest and relevance will be the "installation" tests, which are planned to include testing of a Wiseman friction clutch, and possibly also of the Fawick clutch.

The tests will also examine various ways of silencing the engine and its behaviour when mounted on a vibration-isolating system of "constant position mountings" developed by the Yarrow-Admiralty Research Department.

*Mirrlees National OP Engine*

The OP engine is a through-scavenged opposed-piston two-stroke cycle engine with the cylinders arranged in twin parallel vertical banks. For marine propulsion use the engine will normally be direct reversing.

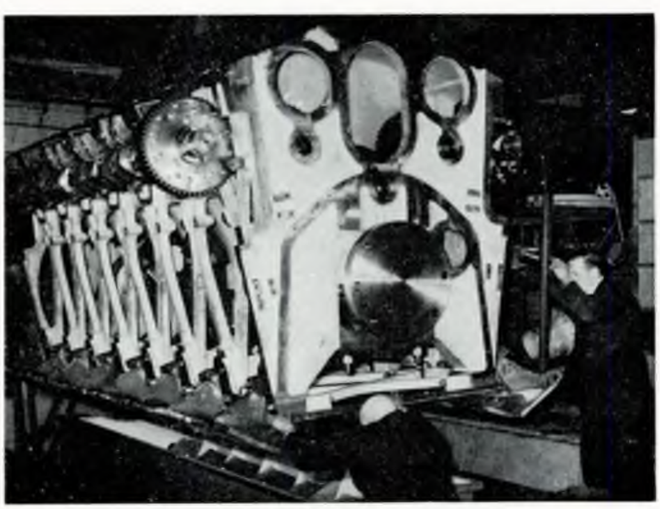


FIG. 2—*12 AO engine under construction*

The upper pistons of each bank will control flow through the exhaust ports and the lower ones the flow through the inlet ports. The engine will have four crankshafts, rotating in the same direction, which may be coupled to the output shaft either by gearing or possibly a drive-plate arrangement. With the former method the output shaft will normally rotate at a lower speed than the crankshafts in a marine propulsion installation.

Details of the design have now been settled and a single-cylinder downward flow engine has been manufactured and is now being used to test the thermally-loaded components such as pistons and exhaust port bars. Other test rigs have also been manufactured to assess performance of fuel pumps, small-end bearings, air flow through ports etc. A half-scale back-to-back rig has also been manufactured to investigate the unique method of coupling the four cranks with a common drive plate to a central output shaft. The design of a twin-cylinder test engine is now well advanced and this should be running by the end of 1967. The object is to produce an engine in which the maintenance load is comparatively light, due to long intervals between overhauls and designed accessibility. In particular, it will be possible to remove a complete cylinder liner sideways from the engine without disturbing the crankshafts and it will be possible to remove any of the four crankshafts without disturbing the engine frame or its alignment with the gearbox.

The engine frame will be of cast-iron construction with hydraulically-tensioned through bolts carrying the firing loads.

For the installations considered in this study, either twin eight-cylinder engines or a single 16-cylinder engine may be used, and these alternatives are compared in Fig. 1.

## OPERATION ON HEAVY FUEL

Because of the large difference in cost between marine Diesel fuel and heavy fuel grades it is mandatory that the propulsion machinery for large ocean-going ships should be capable of operating on heavy fuels.

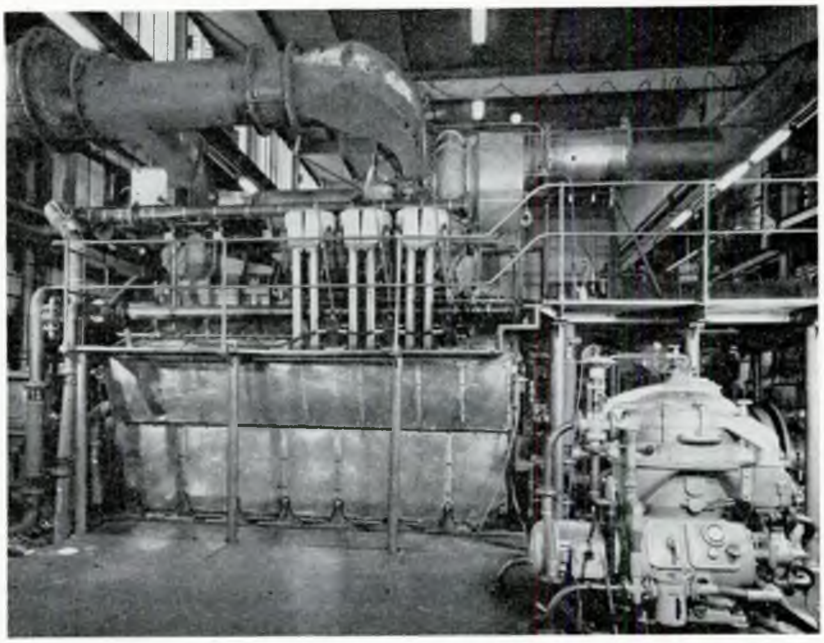


FIG. 3—*12 AO engine on test bed*

Fig. 4 indicates the variation in fuel cost with viscosity. It will be seen that compared with Diesel fuel the rate of price reduction is very steep up to about 200 sec Redwood 1 and that at about 600 sec Redwood 1 the majority of the price reduction has been obtained. Despite this however the projected engines have been designed to operate with fuel having a viscosity of 3500 sec Redwood 1 at 100°F.

A good deal of research and development has been carried out by the manufacturers of medium-speed Diesel engines into the problems associated with the operation of their engines on heavy fuels and, over the last twelve years, increasing numbers of medium-speed engines have been sold for operation on this fuel.

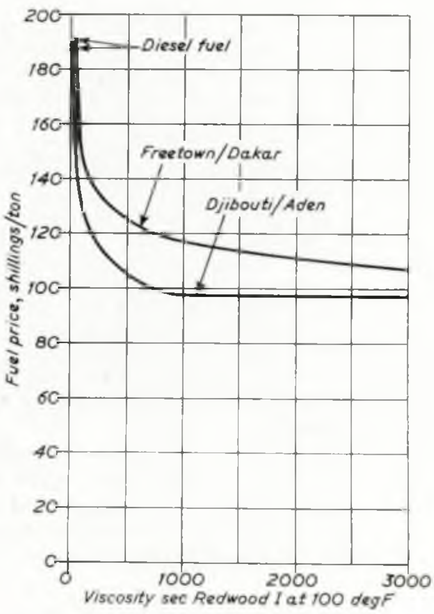



FIG. 4—*Variation of marine fuel oil prices with viscosity—Based on current oil fuel bunker contract prices for 16th August 1965*

Discussion of the difficulties experienced in burning heavy fuels has been covered fully in a number of papers[2, 3, 4, 5]. The main problems have been the build-up of carbon on the injector nozzle tips in the form of trumpets, leading to the eventual deterioration of combustion, and the rapid deterioration of exhaust valve seats, particularly when burning fuels with a high sodium and vanadium content. The most effective cure for both of these problems has been found to be improved cooling of the components concerned and this has been incorporated in the more recent engines which have become available with the result that times between overhaul should approach those required for operation on distillate fuels. In some cases, where cooling is not so effective, exhaust-valve life has been extended by positive rotation of the valves.

### LUBRICATING OIL

The successful operation of a medium-speed trunk-piston engine burning heavy oil depends very much on the correct selection and treatment of lubricating oil.

Two of the principal difficulties experienced when slow-speed marine engines were first operated on heavy fuel were corrosive wear of the cylinder liners and crankcase corrosion. The cure for the former has been the development of highly-alkaline cylinder lubricants to neutralize the acidic products of combustion, and for the latter, the provision of glands around the piston rods to prevent these products from entering the crankcase.

In the medium-speed trunk-piston engine, corrosive liner wear has not proved to be such a problem, since the liner surface temperature is generally higher than that in the slow-speed engine and not subject to such large variations. Due to the lack of a means of separation of the cylinders from the crankcase, it is essential that the respective lubricants be compatible and, in practice, the same lubricant is normally used for both cylinder and bearing lubrication. Since any blow-by past the piston rings will result in contamination of this oil, a high-duty detergent oil is required with sufficient alkalinity to prevent it becoming acidic and causing corrosion of the crankcase components, particularly when the engine is shut down.

In practice, this alkalinity is somewhat less than that of the cylinder oil used in slow-speed engines, and its cost lies approximately midway between that of the cylinder oil and a straight mineral oil, as is frequently used in the slow-speed engine's crankcase.

The additives in the oil used in the crankcase of the medium-speed engine are expended eventually and it is therefore essential that the oil should be regularly tested. Due to the inevitable loss of crankcase oil to the cylinder walls, the lubricating oil consumption of the medium-speed trunk-piston engines is higher than that of the slow-speed engine. The process of topping up the engine sump regularly assists in maintaining the oil in a satisfactory condition and frequent oil changes are not required.

For the purposes of making an economic comparison the total lubricating-oil consumption, i.e. crankcase and cylinder oil has been taken to be as follows:

slow-speed engine—32 gallons per day;
medium-speed engine—96 gallons per day.

A separate lubricating-oil system using a straight mineral oil is provided for the gearing.

#### NOISE

The results of a comparative study of noise levels between slow and medium-speed Diesel engined ships carried out by the British Ship Research Association indicate that, at the cylinder tops, the noise level is slightly higher with the medium-speed engines. At the manoeuvring platform, the lower-speed engines gave the higher noise levels. This latter finding is presumably due to the fact that the higher-speed engines produce noise at the higher-frequency end of the spectrum where attenuation is easier to achieve.

It is apparent that both types of engine produce noise levels, at certain positions in the engine room, to which personnel should not be exposed continuously.

The siting of the manoeuvring platform or the provision of a machinery control room appear to be more important considerations in the design of the machinery installation from the aspect of noise than the type of main engine employed.

#### MAINTENANCE

At the present stage of development of the projected engines, maintenance effort in service and associated costs can only be based on estimates. By contrast, service maintenance data are available for slow-speed engines, although unhappily the data from various sources are not always consistent.

Within these limitations, an endeavour has been made to make the fairest possible comparison of maintenance efforts, based on:

i) for the AO engine, on detailed estimates of man-hours required for maintenance aboard ship, provided by the engine builders and resting upon their experience with maintaining existing engines and building and maintaining the test AO engines;
ii) for the B. and W. engine, on information received from the engine builders supplemented by discussion with various shipowners.

Details of the maintenance schedules are given in Appendix I which indicates that the maintenance requirements are likely to compare as follows:

| | Man-hours required for maintenance at regular intervals of 1000 hours or more, over an engine-running time of 60 000 hours. | |
|---|---|---|
| | Hours | Per cent |
| One seven-cylinder B. and W. engine | 12 271 | 100 |
| Two 16-cylinder R. and H. AO engines and main gearing | 17 384 | 142 |

The difference of 5113 is spread over a period of 60 000 hours and in the context of the two ships under consideration this represents additional maintenance man-hour requirements as follows for the medium-speed engined ships:

bulk carriers — 615 man-hours per year;
cargo liner — 326 man-hours per year.

For the purposes of the economics comparison the whole of the cost of the above maintenance man-hours has been charged against the medium-speed engines, although in fact it would be fair to assume that some of the work would be undertaken by ship staff at no additional cost to the shipowner.

#### RELIABILITY

According to a number of authorities the reliability of the present-day slow-speed Diesel engine is extremely high.

In the power range under consideration it has not been possible to get a direct comparison of the reliability of slow-speed and medium-speed engines. Lloyd's Register of Shipping have, however, provided a comparison of reported defects between geared and direct-drive Diesel engines of comparable powers and ages. This information is reproduced in Appendix II.

According to their age and power the installations have been divided into groups A, B and C, each group comprising the same number of shaft sets driven by geared and by direct-drive Diesel machinery. The defects reported are only those which have figured in reports by Lloyd's Register surveyors, and do not therefore include all the defects that have occurred.

In endeavouring to draw some conclusions from the statistical data, it must be borne in mind that the available sample is limited, and no attempt has been made to establish to what extent it can be considered to be representative.

However, the figures do show a well-defined trend and to highlight this the following procedure has been adopted:

a) a combined list has been drawn up to include all the parts mentioned in Appendices IIA, IIB and IIC;
b) from Appendices IID and IIE, the total number of the parts included in the list a) above was estimated for each group of ships;
c) this number was defined as "The number of parts at risk" and abbreviated to No. P.R.;
d) the percentage of the defects reported as a function of No. P.R. has been evaluated;
e) the only parts contained in the Lloyd's Register data, but excluded from the analysis are crankpin bearing bolts indicated in Appendix IIC; these parts are mentioned only in that Appendix, and there is only one defect from 140 parts for direct-drive and 528 parts for geared engines; it was felt that inclusion of this portion of the data would have an inappropriate swamping effect.

The analysis of the Lloyd's Register data, carried out on the above basis, is summarized in the table overleaf:

From this table it is evident that:

1) in group A, the lower power group, the ratio of defects to number of components is similar for direct-drive and geared; the total number of defects in the geared machinery is about three times those in the direct-drive machinery;
2) in group B, the higher power group, the ratio of defects to number of components for direct-drive is about three times the ratio for geared drive; the number of defects in the geared machinery is about 30 per cent more than in the direct-drive machinery;
3) in group C, of similar mean power to group B, the ratio of defects to number of components for direct-drive is about six times that for geared drive; the number of defects in the geared machinery is about half the number in the direct-drive machinery;
4) based on the average percentage of defects, the number of defects predicted in the slow-speed engine installation would be five and in the medium-speed engined installation, nine.

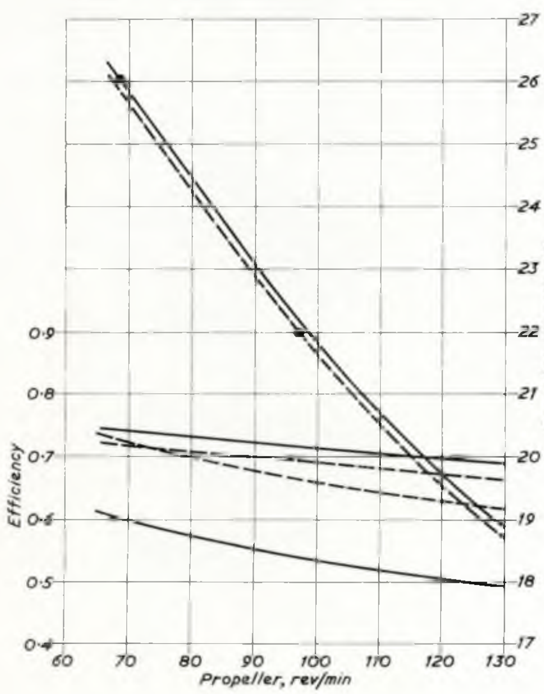
| Lloyd's Register Group and shp shaft | Drive | No. of parts at risk | No. of defects | Ratio: Defects No. P.R. per cent |
|---|---|---|---|---|
| A 2000–4500 | Direct<br>Geared | 774<br>2015 | 35<br>117 or 63* | 4·5<br>5·8 |
| B 5750–8250 | Direct<br>Geared | 405<br>1514 | 41<br>55 or 29* | 10·1<br>3·6 |
| C 6000–6400 | Direct<br>Geared | 632<br>2040 | 60<br>31 or 20* | 9·5<br>1·5 |
| C′ | Direct<br>Geared | 632<br>2040 | 85<br>44 | 13·4<br>2·2 |
| Slow-speed engine<br>Medium-speed engine | Direct<br>Geared | 55<br>230 | 5<br>9 | 9·3†<br>3·9† |

*A single incident appears to have caused a number of defects and although the higher figures have been used it might be argued that the lower figures would be appropriate.
†Mean of groups A, B and C′. These percentages were used to predict the number of defects in slow-speed and medium-speed engines.
*NOTE:* Groups A and B were built in 1956/7.
Group C was built in 1958/59/60.
Group C′ is group C extrapolated *pro rata* to 1956/7 to permit the ratio No. P.R.: Defects to be averaged.
In all cases the defects are the totals reported from going into service to June 1965.

The foregoing information suggests that the number of defects to be expected in a medium-speed geared Diesel installation is not increased *pro rata* to the number of parts at risk as compared with direct-drive machinery.

### PROPELLER SPEED

The freedom of choice of propeller speed in a geared installation permits the optimum speed to be selected. It is not enough to fit the most efficient propeller which can be accommodated, however, as the cost of the transmission system varies with propeller speed and must be taken into account.




———— Bulk carrier
- - - - - - - - Cargo liner

FIG. 5—*Variation of propulsive efficiency and propeller open efficiency with propeller rev/min—Based on information provided by the National Physical Laboratory*

Curves showing the variation in propeller diameter, propeller open water efficiency and propulsive efficiency with propeller rev/min provided by the National Physical Laboratory (Ship Division) for the ships in question are shown in Fig. 5, and show the extent of the increase, with reduced rev/min, of:

i) the propulsive efficiency;
ii) the propeller diameter for maximum propulsive efficiency at that rev/min.

The maximum propeller diameter which can be accommodated in the two ships has been taken to be as follows:

bulk carrier 26 ft;
cargo liner 22 ft.

On this basis the propeller rev/min for optimum propulsive efficiency in the two ships is as undernoted:

bulk carrier 68 rev/min;
cargo liner 97 rev/min.

The gains in propulsive efficiency associated with the larger slower-turning propellers are as follows, with the rev/min of the slow-speed engine determined by engine speed:

| | Bulk carrier | | Cargo liner | |
|---|---|---|---|---|
| | Direct-drive | Geared | Direct-drive | Geared |
| Rev/min | 114 | 68 | 114 | 97 |
| Propulsive efficiency | 0·702 | 0·741 | 0·678 | 0·694 |
| Gain in propulsive efficiency, per cent | Basis | 5·5 | Basis | 2·3 |

With ships having higher block coefficients, greater improvements in efficiency can be expected with the slower-turning propellers.

Based on the e.h.p. requirement derived for each ship, which takes advantage of the reduced displacement of the medium-speed engined ships, as discussed later, the comparison of power requirements becomes as follows:

| | Bulk carrier | | Cargo liner | |
|---|---|---|---|---|
| | Direct-drive | Geared | Direct-drive | Geared |
| Rev/min (max. power) | 114 | 68 | 114 | 97 |
| Ehp (service power) | 9920 | 9825 | 9570 | 9360 |
| Propulsive efficiency | 0·702 | 0·741 | 0·678 | 0·694 |
| Dhp | 14 110 | 13 250 | 14 110 | 13 500 |
| Shafting efficiency | 0·98 | 0·98 | 0·98 | 0·98 |
| Shp | — | 13 520 | — | 13 780 |
| Gearing efficiency | — | 0·98 | — | 0·98 |
| Bhp | 14 400 | 13 800 | 14 400 | 14 040 |

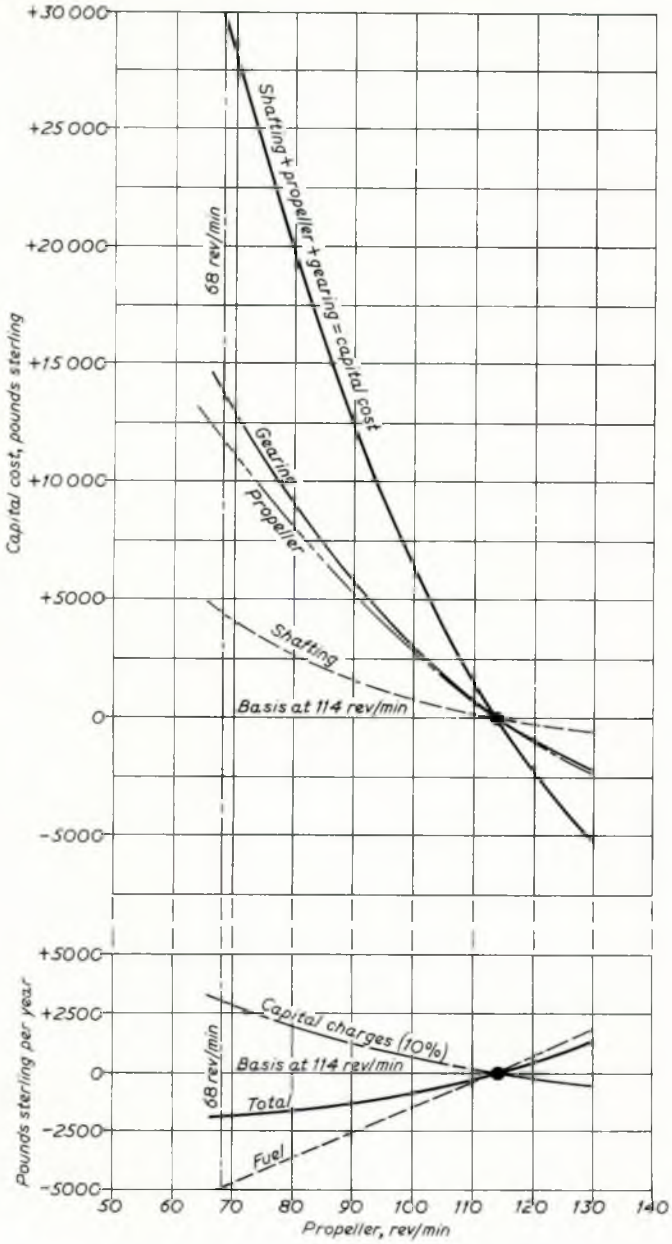



FIG. 6—*Cost variation, propeller rev/min, bulk carrier—Basis: fuel cost £5 per ton, capital charges 10 per cent per annum, propeller curve includes half the cost of a spare propeller, shaft curve includes cost of a spare tailshaft, gearing curve includes cost of Lloyd's spares*

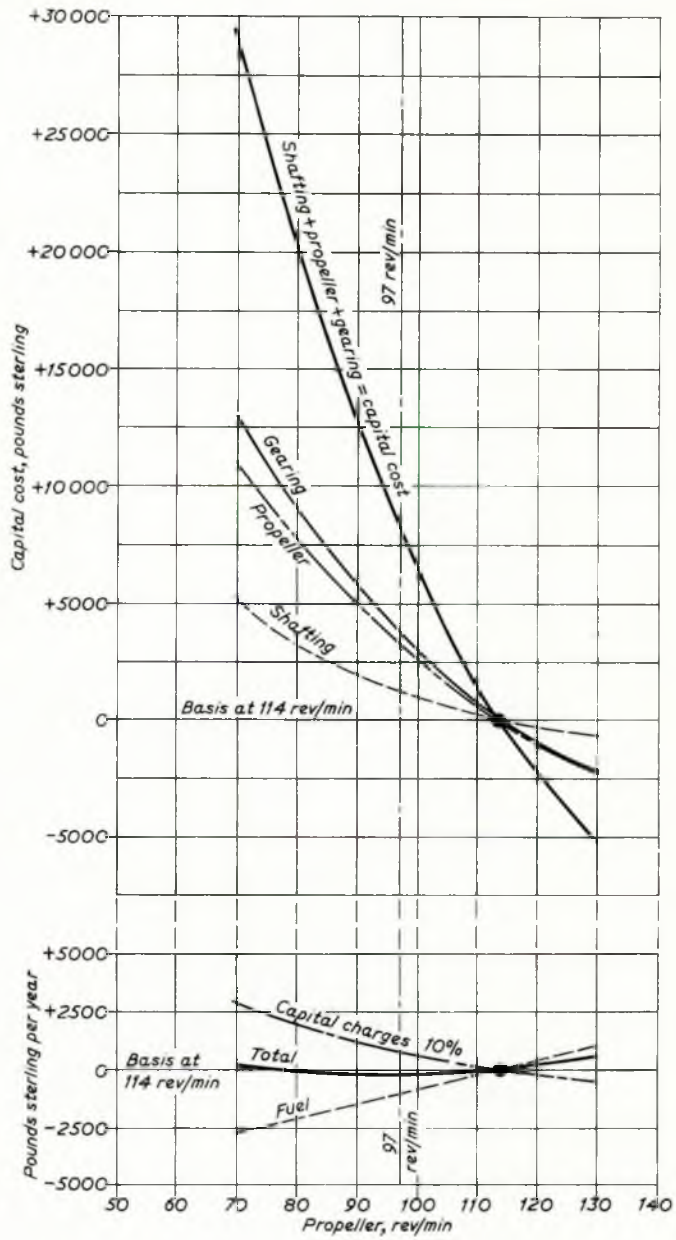



FIG. 7—*Cost variation, propeller rev/min, cargo liner—Basis: fuel cost £5 per ton, capital charges 10 per cent per annum, propeller curve includes half the cost of a spare propeller, shaft curve includes cost of a spare tailshaft, gearing curve includes cost of Lloyd's spares*

As propeller rev/min is reduced the transmission torque increases and hence gearing and shafting, as well as the propeller, become heavier and more expensive.

The variations in these prices have been plotted in Figs. 6 and 7 for the bulk carrier and cargo liner respectively. The individual prices have been added to show the variation in total capital costs. The shafting prices include the cost of a spare tailshaft. The practice in some companies is to provide one spare propeller for a class of ships, while others provide a spare propeller for each ship. For the purposes of this study it has been assumed that one spare propeller is shared by two ships. The curves of propeller costs therefore include the working propeller plus half the cost of a spare propeller.

Based on a total annual charge on capital of ten per cent, the capital charges per annum have also been plotted. The variation in required b.h.p. to maintain a constant ship speed has been determined from the curves in Fig. 5 and the variation in annual fuel costs with propeller rev/min estimated assuming a gearing efficiency of 98 per cent and a shafting efficiency of 98 per cent. The curves of capital charges and fuel costs have been added to give the variation in annual outlay.

It will be seen that, for the bulk carrier, the running cost is falling as propeller rev/min is reduced and therefore the slowest running propeller is the most economic.

The cargo liner has longer shafting than the bulk carrier and the cost of the shafting for the cargo liner increases at a slightly greater rate as the propeller rev/min is reduced compared with the bulk carrier. Also, the cargo liner spends much less time at sea, and as can be seen in Fig. 7, the saving in fuel costs is less significant than in the case of the bulk carrier.

The slowest running propeller is the optimum for the cargo liner, although the advantage is quite small.

The fuel price used (£5 per ton) is about the cheapest which can be expected and with higher fuel prices the adoption of slow propeller speeds would increase the advantage of geared engines.

### CLUTCHES AND COUPLINGS

When two engines are geared to a single propeller shaft, couplings may be required between the engines and gearbox, to provide the following functions:

i) ready means of connecting or disconnecting either engine;
ii) torsional flexibility;
iii) transverse flexibility.

The equipment available and the functions it can fulfill are summarized below:

| Connexion and disconnexion | Torsional flexibility | Transverse flexibility |
|---|---|---|
| Hydraulic couplings<br>Electromagnetic slip couplings | | Elastic couplings |
| Friction clutches<br>Dog clutches | Elastic couplings | |

Transverse flexibility is essential only if the engines are flexibly mounted, since otherwise a rigid combined seating is provided under the engines and gearbox and the relative movement will be extremely limited. Flexible mounting of the engines is not being considered in the present context.

In a geared twin-engined installation the provision of a ready means of connecting or disconnecting either engine from the gearbox is desirable to enable:

a) the ship to operate on one engine when ship operating conditions permit;
b) the ship to proceed at reduced speed after breakdown of one engine;
c) testing of the engines to be carried out in harbour without turning the main shaft.

The selection of the means of connecting to, and disconnecting the engines from, the gearbox must take into account the method by which manoeuvring will be carried out, and the following methods are possible:

| Engine | Normal manœuvring | Crash astern |
|---|---|---|
| Direct-reversing engines | i) Run one engine ahead and the other astern and engage as necessary the appropriate clutch/coupling.<br>ii) Use only one engine and direct reverse it as appropriate.<br>iii) Use both engines and direct reverse as appropriate. | Direct reverse both engines |
| Uni-directional engines | a) Reversing gearbox.<br>b) Controllable-pitch propeller. | |

The reversing gearbox or c.p. propeller may show to advantage in certain cases, even when reversing engines are available, but they have not been considered further in the general study as they would be more expensive than direct-reversing engines.

Of the remaining methods, i) is the most desirable from engine aspect, since it limits to a minimum the number of occasions on which cold starting air is admitted to the engine cylinders. Some form of slipping clutch is essential for this duty, which precludes the use of a simple dog clutch.

The hydraulic coupling is the most popular form of coupling in use today due to its ability to absorb torsional vibrations and shock loads and also to take up the drive gradually on engagement. Electromagnetic slip couplings have advantages similar to the hydraulic coupling. However both the hydraulic and the electromagnetic couplings involve a continuous loss of power and are a little more expensive than the friction clutch, as discussed later.

Particular attention has been paid to friction clutches, the use of which improves the economic advantage of the geared engines. Friction clutches of various types have been commonly used in reverse/reduction gearboxes of medium and low power, where the maximum power transmitted by an individual clutch is about 1500 bhp.

However, the application of friction clutches for manoeuvring at the higher powers and with the large ships considered in this study imposes problems with regard to the dissipation of the heat produced when slipping occurs. A number of clutch designs, capable of transmitting 8000 bhp at 450 rev/min and potentially capable of coping with manoeuvring conditions without seizure or undue wear, is available and includes:

Wiseman oil-operated multi-plate clutch.
Hindmarch/MWD oil-operated serrated-disc clutch.
Lohmann and Stoltertfort Pneumaflex air-operated double-cone clutch.
Wichita air-tube disc clutch.
Fawick Airflex air-tube clutch.

The Wiseman clutch is a new design which is to be tried out in conjunction with the 12-cylinder AO engine as part of the type tests which the Ministry of Defence (Navy) are conducting on the engine. The test rig is being designed to simulate a marine propulsion installation in some respects, so that the capabilities of the clutch under manoeuvring conditions can be assessed. The Wiseman clutch is intended for flange mounting at the aft end of the gearbox, being driven by a quillshaft passing through the hollow gearbox input shaft. A gear tooth coupling is used to couple the quillshaft to the clutch. Oil for cooling and lubricating is taken from the gearbox system. The use of this clutch has been assumed in the comparison.

There is little doubt that any of the clutches mentioned can transmit the required torque and act as isolating clutches, but it is desirable that claims made with regard to their manoeuvring capacity should be backed up by both calculations and tests.

In most installations of two engines coupled to a single propeller by a gearbox, torsional flexibility is required between the engines and gearbox in order to limit the torque variation at the gearing due to torsional vibration.

Elastic couplings will not normally isolate the engines from the gearbox to the extent that hydraulic or electromagnetic slip couplings do. This is due to the limited elasticity which can be obtained without overstressing the flexible elements. The type of coupling which will normally be required, in order to obtain a sufficiently low stiffness, will have rubber in shear elements and three such couplings are known to be available for the duty required:

Metalastik BB20 Duolastik.
Twiflex-Vulkan EZ 320.
Pneumaflex KAH. 380M.

The Twiflex-Vulkan EZ and Pneumaflex couplings have a torsional stiffness which is almost independent of torque with the Pneumaflex being slightly stiffer than the Twiflex-Vulkan coupling.

The Metalastik coupling has an extremely low stiffness up to a certain change-over torque, thereafter torque is transmitted mainly through rubber buffers in compression, so that the resulting stiffness is comparatively high. The coupling depends for its operation on the fact that the torque required from the engine varies with the square of the speed. Thus although the change-over point may occur at 25 per cent full load torque, this corresponds with 50 per cent of full speed. At low speeds when operating on the low stiffness part of the coupling characteristic curve, all serious criticals due to the first two modes of vibration should be situated well below the engine idling speed. At higher speeds when operating on the high stiffness portion of the characteristic, these critical speeds may enter the running range, but should still be situated below the change-over speed.

Calculations have been carried out by Ruston and Hornsby Ltd. into the torsional vibration situation of both the bulk carrier and the cargo liner, when propelled by twin 16 AO engines. In these calculations, both a Metalastik Duolastik coupling and a quillshaft have been used to obtain the required flexibility between engines and gearing and a satisfactory disposition of critical speeds has been obtained to give an operating speed range from 125 to 450 rev/min in any of the possible modes of operation, i.e. two engines driving the propeller, one engine driving the propeller and the engine running disconnected from the propeller.

A Metalastik BB20 Duolastik coupling has been assumed in the economic comparison of medium-speed and slow-speed engines.

### GEARING

No difficulty is foreseen in obtaining suitable gearing for the installations under consideration.

The gearing designs and cost data used have been provided by a number of the leading gearing designers in this country. To assist in determining the most economic propeller rev/min, the gearing designs covered a wide range of output rev/min. The approximate variation in costs with propeller rev/min is shown in Figs. 6 and 7.

It is interesting to note that above a propeller speed of about 90 rev/min the pinion centres can be arranged at the minimum which the engines will permit, i.e. 11 ft 6 in for the AO engine. Below this rev/min, however, to maintain an acceptable face width: diameter ratio, the pinion centres and hence the engine centre distance has to be increased.

Where a single medium-speed engine is large enough to provide all the power, and a 16-cylinder Mirrlees OP engine for example would give the 16 000 bhp for the installations under consideration, a significant reduction in the cost of the gearing is possible. Such an engine and gearbox are shown in Fig. 1.

### MACHINERY ARRANGEMENTS

To compare space requirements for the slow-speed and medium-speed engines, machinery arrangement drawings have been prepared based on the Burmeister and Wain engine and the Ruston and Hornsby AO engine.

Fig. 8 summarizes the machinery arrangements for the cargo liner.

Arrangement I shows the space required for the slow-speed engined installation and as can be seen the full height of the ship is used.

The medium-speed engines allow greater freedom in the arrangement of the machinery space and to give some indication of this, two machinery layouts have been prepared based on the medium-speed engines.

The main engine would allow a low deckhead if required and therefore Arrangement II has been drawn on this basis. The total length of the engine room is 5 ft 0 in longer than for the slow-speed engine, but, apart from the engine-room casing, a complete deck is made available for cargo.

Arrangement III uses the full height of the ship and the total length of the engine room is 10 ft 0 in shorter than that for the slow-speed engine. The space saved can therefore be added to the cargo space, or the ship made smaller.

The overall sizes and volumes of the machinery space for the three arrangements are as follows:

| | I Slow-speed | II Medium-speed | III Medium-speed |
|---|---|---|---|
| Height to deckhead of engine room above keel, ft | 44 | 33 | 44 |
| Length, ft | 77½ | 82½ | 67½ |
| per cent | 100 | 106 | 87 |
| Volume: | | | |
| i) Below main deck, $ft^3$ | 180 000 | 151 000 | 155 000 |
| per cent | 100 | 84 | 86 |
| ii) Above main deck, $ft^3$ | 45 000 | 29 000 | 29 000 |
| iii) Total, $ft^3$ | 225 000 | 180 000 | 184 000 |
| per cent | 100 | 80 | 82 |

It will be seen that the total volume occupied by the geared medium-speed installation is similar in each case and is approximately 18 per cent less than the space required for the slow-speed machinery installation.

### BULK CARRIER

Fig. 9 shows two machinery arrangements for the bulk carrier.

The depth of this ship is such that in way of the machinery space it could be divided into four different levels. Even with the slow-speed engine the top flat is not necessarily required for machinery and could be used say for crew accommodation or ship stores. It is hardly likely that a particularly low machinery space would be required in this class of ship and therefore only one machinery arrangement has been prepared for the slow-speed installation and one for the medium-speed installation.

In each case, therefore, it has been assumed that the deck immediately below the main deck has been used for crew accommodation.

As mentioned earlier, due to the low propeller rev/min the gearing centre distance has had to be increased for the bulk carrier with the result that the main engine unit occupies slightly more width in the bulk carrier than in the cargo liner. However the ship has plenty of beam to accommodate this.

The overall size, and volume, of the machinery space for the two arrangements are as follows:

| | I Slow-speed | II Medium-speed |
|---|---|---|
| Height of deckhead of engine room above keel, ft | 47 | 47 |
| Length, ft | 102 | 93 |
| per cent | 100 | 91 |
| Volume: | | |
| i) Below main deck, $ft^3$ | 217 000 | 187 000 |
| per cent | 100 | 86 |
| ii) Above main deck, $ft^3$ | 45 000 | 29 000 |
| iii) Total, $ft^3$ | 262 000 | 216 000 |
| per cent | 100 | 82·5 |

Again the space occupied by the geared medium-speed engined installation is approximately 18 per cent less than the space required for the slow-speed machinery installation.

### ALTERNATIVE MEDIUM-SPEED ENGINES

The Mirrlees OP engine is not provided with arrangements to drive pumps and therefore independent motor-driven lubricating-oil pumps are required. The Diesel generators would be larger than those required for the AO-engined installation, but smaller than those in the slow-speed engined installation.

If two engines are used per shaft, the Mirrlees OP engines are at a slight disadvantage compared with the Ruston and Hornsby engines at the power under consideration due to their greater width, which would prevent the siting of auxiliaries at the sides of the engines and hence demand an engine room slightly longer than that for the Ruston and Hornsby AO engines.

A machinery installation based on a single Mirrlees OP engine developing 16 000 bhp would, however, have space requirements similar to the AO engines. Although the single engine and gearbox would be longer than the twin engines, the saving in width compensates for the increased length.

As will be seen from Fig. 1, if existing engines were used in place of the projected engine the space requirements would be increased a little.

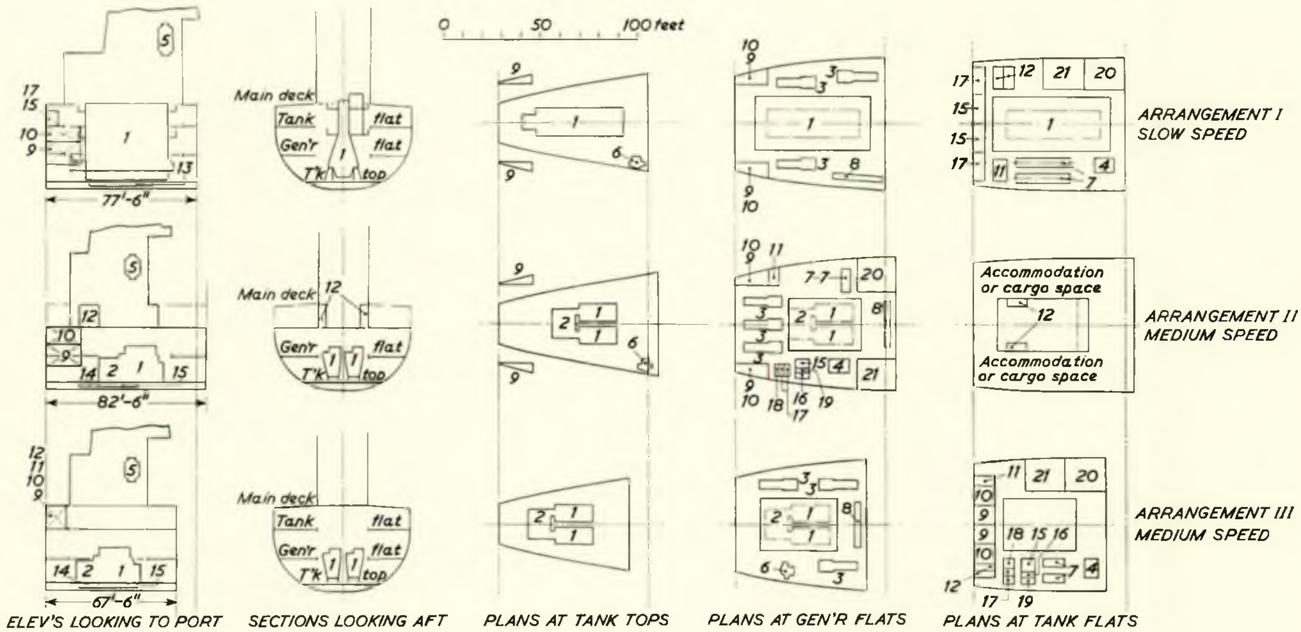



FIG. 8—*Comparison of machinery arrangements for cargo liner*

Key to Figs. 8 and 9:
1) Main engines
2) Gearing (medium-speed only)
3) Diesel alternators
4) Oil-fired boiler
5) Waste-heat boiler
6) Evaporator
7) Main air reservoirs
8) Main switchboard
9) Fuel-oil settling tanks
10) Fuel-oil service tanks
11) Diesel-fuel settling tank
12) Diesel-fuel service tanks
13) Main-engine lubricating-oil drain tank (slow-speed only)
14) Gearing lubricating-oil drain tank (medium-speed only)
15) Main-engine lubricating-oil renovating tanks
16) Gearing lubricating-oil storage tank (medium-speed only)
17) Main-engine lubricating-oil renovating tanks
18) Gearing lubricating-oil renovating tank (medium-speed only)
19) Diesel alternators lubricating-oil storage tank
20) Engineers' workshop
21) Engineers' stores

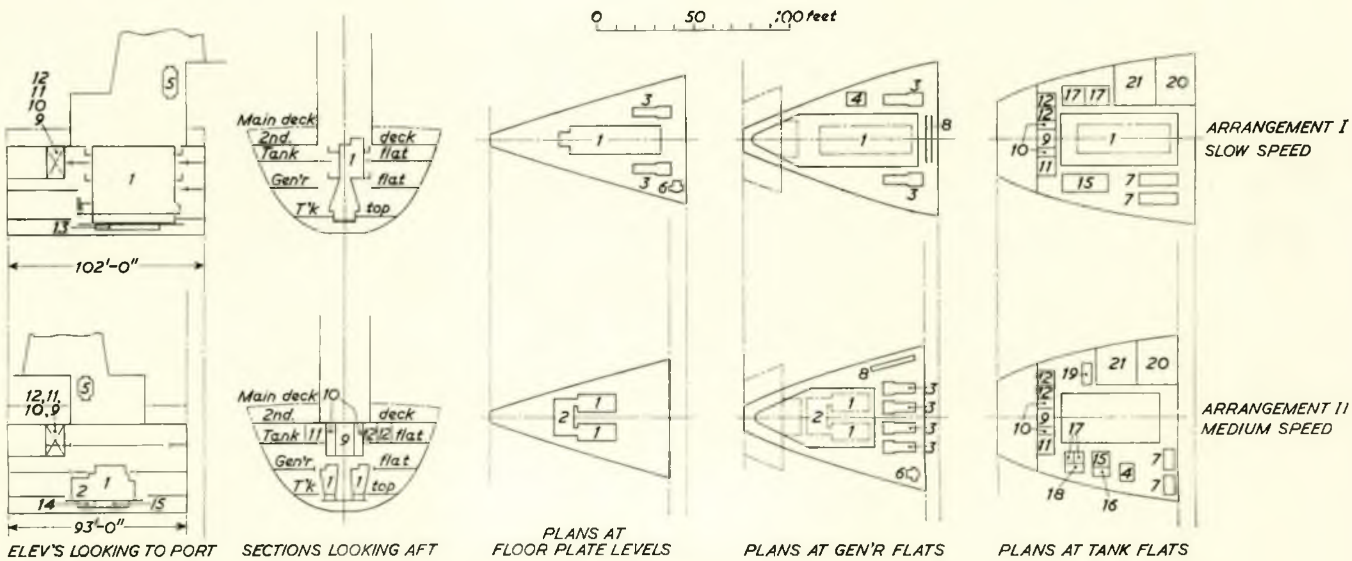



FIG. 9—*Comparison of machinery arrangements for bulk carrier*

WEIGHT

Estimates of the weights for the medium-speed geared and the slow-speed engined installations are given in Table III for both the bulk carrier and cargo liner.

The estimates for the geared installation are based on Ruston and Hornsby AO engines and include the slow-speed propeller and associated shafting referred to earlier. The gearing weight is also based on the low output rev/min.

TABLE III—MACHINERY WEIGHTS

| Machinery items | Bulk carrier | | Cargo liner | |
|---|---|---|---|---|
| | B. and W. | R. and H. | B. and W. | R. and H. |
| | Weight, tons | Weight, tons | Weight, tons | Weight, tons |
| Main engine (wet) | 621 | 126 | 621 | 126 |
| Gearing including clutches and couplings (wet) | — | 79 | — | 67 |
| Shafting and propeller | 66 | 91 | 87 | 92 |
| L.O. pumps, starters, coolers, filters, pipes and valves (wet) | 19 | 25 | 19 | 25 |
| F.W. cooling pumps, starters, coolers, pipes and valves (wet) | 10 | 11 | 10 | 11 |
| S.W. cooling pumps, starters, pipes and valves (wet) | 9 | 9 | 9 | 9 |
| Fuel supply system pipes and valves (wet) | 1 | 1 | 1 | 1 |
| Fuel valve cooling pumps, starters, cooler, pipes and valves (wet) | 2 | — | 2 | — |
| Starting air compressors, starters, air reservoirs, pipes and valves | 22 | 10 | 22 | 10 |
| Diesel alternators | 68 | 52 | 63 | 51 |
| Exhaust uptakes | 2 | 3 | 2 | 3 |
| Waste-heat boiler (wet) | 18 | 18 | 18 | 18 |
| Lubricating oil in engine-room tanks | 82 | 43 | 80 | 36 |
| Total | 920 | 468 | 934 | 449 |
| Difference | Basis | –452 | Basis | –485 |

The weights given in Table II, for various medium-speed engines including gearing, give an indication of the variation which can be expected in machinery weights with other engine types.

FUEL CONSUMPTION

The installed b.h.p. of each ship is the same. However, due to different ship sizes, propeller speeds and transmission efficiencies the service b.h.p. in each case varies as follows:

| | Bulk carrier bhp | Cargo liner bhp |
|---|---|---|
| Medium-speed geared engines | 13 800 | 14 040 |
| Slow-speed Diesel-drive engines | 14 400 | 14 400 |

Based on fuel oil of 3500 sec Redwood 1 at 100 deg F with a net calorific value of 17 000 Btu/lb the main engine fuel consumptions have been derived from the above powers and the undernoted specific fuel consumptions:

| | |
|---|---|
| Medium-speed engines (including engine-driven L.O. pumps) | 0·385 lb/bhp-h; |
| Slow-speed engines | 0·378 lb/bhp-h. |

It has been assumed that the steam for bunker heating and fuel oil heating is supplied from the waste-heat boiler and that it is not necessary to burn oil in the oil-fired boiler at sea.

Due mainly to the provision of engine-driven lubricating-oil pumps on the medium-speed engines, the difference in electrical load of the slow-speed and medium-speed installation is about 150 kW and therefore different sizes of Diesel generators have been provided in each installation. The difference in electrical loads has been taken into account in estimating fuel consumptions which have been based on Diesel fuel with a net calorific value of 18 000 Btu/lb and the following electric loads:

| | Bulk carrier | Cargo liner |
|---|---|---|
| Medium-speed engines | 660 kW | 425 kW |
| Slow-speed engines | 810 kW | 575 kW |

The total fuel required for each installation has been estimated to be as follows:

| | Bulk carrier (Fuel for 40 days) | | Cargo liner (Fuel for 30 days) | |
|---|---|---|---|---|
| | Slow-speed | Medium-speed | Slow-speed | Medium-speed |
| Main engine—heavy fuel, tons | 2333 | 2278 | 1750 | 1738 |
| Diesel generator—Diesel fuel, tons | 176 | 145 | 93 | 70 |
| Total, tons | 2509 | 2423 | 1843 | 1808 |

The estimated annual fuel costs are as follows:

| | Bulk carrier | | Cargo liner | |
|---|---|---|---|---|
| | Slow-speed | Medium-speed | Slow-speed | Medium-speed |
| Heavy fuel (£5 per ton) | £87 500 | £85 390 | £46 600 | £46 340 |
| Diesel fuel (£9 10s. per ton) | 12 500 | 10 300 | 4 710 | 3 540 |
| Total | 100 000 | 95 690 | 51 310 | 49 880 |
| Difference £/year | Basis | –4310 | Basis | –1430 |

CONTROL AND MANNING

Various forms of control philosophies can be applied to the medium-speed and slow-speed engines and consideration has been given to the following:

1) local control on engines, including alarms and auto-start devices and auto-shut-down equipment;
2) control console on platform in engine room, if required with acoustic enclosure, and otherwise as 1);
3) manned remote machinery control room with facility for bridge control;
4) unmanned remote machinery control room with bridge control and automatic recording of important parameters;
5) as 4) plus comprehensive data-logging monitoring 155 and 252 points in the slow-speed and medium-speed installations respectively.

The more sophisticated forms of control are, at first sight, difficult to justify on economic grounds, but mainly in view of the difficulty in obtaining suitable engine-room staff a trend towards such controls and unmanned engine rooms must be expected.

Due to the two engines and additional number of parameters to monitor, any of these control systems is more expensive for the medium-speed engines than the slow-speed engine.

For the purposes of the economic assessment the costs of a control console within the engine room have been used, i.e. 2) above, which is estimated to involve an additional cost of £3000 in the medium-speed engined installation.

Based on estimated operational and maintenance requirements, it is considered that the manning of the engine room would be similar for the slow-speed engines and medium-speed engines and that this aspect will therefore not affect the economic comparison of the installations.

#### EFFECT ON SHIP DESIGN, CONSTRUCTION AND COST

It has been shown that the saving in machinery space, weight and fuel bunkers in favour of the medium-speed engines is as follows:

| | Bulk carrier | Cargo liner |
|---|---|---|
| Space, ft$^3$ (below main deck) | 30 000 | 25 000 |
| Machinery weight, tons | 452 | 485 |
| Fuel bunkers, tons | 86 | 35 |

To estimate the effect of these savings, on the ship as a whole, the basic ship particulars have been revised as necessary when using medium-speed geared Diesel engines to restore the cargo deadweight in the case of the bulk carrier and the cubic capacity in the case of the cargo liner to the same values as with direct-drive engines. The resulting ship particulars are shown below:

| | Bulk carrier | | Cargo liner | |
|---|---|---|---|---|
| | Basis ship | Modified ship for medium-speed engine | Basis ship | Modified ship for medium-speed engine |
| Length bp, ft | 750 | 743·75 | 500 | 489 |
| Breadth moulded, ft | 105 | 105 | 70 | 70 |
| Depth moulded, ft | 56 | 56 | 43 | 43 |
| Draught, deep, ft | 38·5 | 38·5 | 31 | 31 |
| Displacement, deep, tons | 68 700 | 67 930 | 18 450 | 17 860 |
| Block coefficient | 0·79 | 0·789 | 0·59 | 0·584 |
| Ehp | 9920 | 9825 | 9570 | 9360 |
| Cargo capacity, ft$^3$ | 2 700 000 | 2 699 000 | 665 000 | 665 000 |
| Deadweight, tons | 55 000 | 54 907 | 12 000 | 12 072 |
| Fuel oil weight, tons | 2509 | 2423 | 1843 | 1808 |
| Deadweight less fuel, tons | 52 491 | 52 484 | 10 157 | 10 264 |
| Steel weight, tons | Basis | −215 | Basis | −145 |
| Outfit weight, tons | Basis | −10 | Basis | −22 |
| Service speed, knots | 15 | 15 | 19¼ | 19¼ |

The reduction in steel and outfit weight shown in the above table in favour of the medium-speed engines results in a reduction in the cost of the ship which is taken into account later.

It is quite clear from the work carried out that, even in ships of widely differing size from those selected as basis, the medium-speed geared Diesel installation will show a definite saving in weight and space as compared with direct-drive slow-speed Diesel engines. It has been argued in the past that more floor space is required for twin-geared than for a slow-speed engine, but the size of the projected medium-speed engines has been reduced to the extent that geared twin AO engines or a single-geared OP engine occupy only about 80 per cent of the area of a slow-speed engine.

The value of medium-speed engines, whether direct-driven or geared, is already well exploited in special purpose vessels such as tugs, coasters and cross channel ferries and the projected engines will give increased advantage in reduced weight, space and cost of these vessels as compared with existing engine designs in the appropriate power range.

#### OVERALL ECONOMIC COMPARISON

A comparison of those aspects of the bulk carrier and cargo liner which affect the economics of ship operation and which are influenced by the type of machinery selected has been made, including the effects of the following:

i) capital cost of hull, main machinery and auxiliary machinery;
ii) lubricating oil consumption;
iii) fuel consumption;
iv) maintenance and crew costs;

It is not practicable to translate this information into annual costs, which could be regarded as typical of any one company, in view of the wide difference in such factors as interest rates and fuel prices, which will exist between one ship operator and another depending upon the company policy, the trade and the part of the world in which the ship operates.

In consequence, the information which will affect the economic assessment of the ships with geared engines or direct-drive engines has been presented in a manner which will permit shipowners to substitute their own values and determine with their particular operating conditions how much advantage there will be in adopting geared machinery.

It is to be noted, however, that typical assumptions had to be made in estimating the optimum propeller speed.

### *Capital Costs*

A summary of the cost estimates for the slow-speed and medium-speed engined installations is given in Table IV for both the bulk carrier and the cargo liner.

In the case of the geared machinery the costs for gearing, shafting and propeller have been based on the most economic propeller speed as discussed earlier.

Including an allowance for the reduction in the hull cost for the medium-speed engined ship, the total variation of capital cost of machinery and hull in favour of the geared Diesel engines is as follows:

| | Bulk carrier | Cargo liner |
|---|---|---|
| Saving in machinery cost | £26 900 | £52 300 |
| Saving in hull costs | £32 000 | £27 000 |
| Total | £58 900 | £79 300 |

### *Maintenance and Crew Costs*

Assuming that the whole of the additional man-hours required for maintenance of the medium-speed engines compared with the slow-speed engines will be paid for at a rate of £1 per hour, the additional annual costs will be as follows:

| | |
|---|---|
| Bulk carrier | £615 per annum; |
| Cargo liner | £326 per annum. |

### *Economic Comparison*

Taking into account those aspects which affect the economic comparison of geared medium-speed and direct-drive slow-speed engines, the annual running costs may be expressed in terms of the following:

$I$ per cent = annual capital charge as percentage of capital cost, to cover interest, depreciation and insurance;

$L_{cyl}$ = lubricating oil price (cylinder), shillings/gallon;

$L_{cr}$ = lubricating oil price (crankcase), shillings/gallon;

$L$ = lubricating oil price for medium-speed engine, shillings/gallon;

$F$ = heavy fuel oil price, £/ton;

$D$ = Diesel fuel oil price, £/ton;
$M$ = cost of maintenance labour, £/hour;
$d$ = days at sea per year.

TABLE IV—MACHINERY INSTALLATION COSTS

| | Bulk carrier | | Cargo liner | |
|---|---|---|---|---|
| | B. and W. | R. and H. | B. and W. | R. and H. |
| Machinery items | Installed cost (£) | Installed cost (£) | Installed cost (£) | Installed cost (£) |
| Main engines including recommended spares controls, gearing, clutches and flexible couplings | 311 000 | 275 640 | 311 000 | 267 255 |
| Propeller, shafting, stern tube, spare tailshaft and half cost of spare propeller | 37 690 | 54 240 | 39 920 | 44 140 |
| Waste-heat boilers (5000 lb/h) | 4925 | 4950 | 4925 | 4950 |
| L.O. pumps, starters, coolers, filters, pipes and valves (including engine-driven L.O. pumps) | 21 260 | 32 700 | 21 260 | 32 700 |
| F.W. cooling pumps, starters, coolers, pipes and valves | 10 430 | 12 880 | 10 430 | 12 880 |
| S.W. cooling pumps, starters, pipes and valves | 12 290 | 12 990 | 12 290 | 12 990 |
| Fuel supply system pipes and valves | 1680 | 1750 | 1680 | 1750 |
| Fuel valve cooling pumps, starters, coolers, pipes and valves | 1500 | — | 1500 | — |
| Starting air compressors, starters, air reservoirs, pipes and valves | 10 250 | 4580 | 10 250 | 4580 |
| Diesel alternators | 68 840 | 55 781 | 67 016 | 51 630 |
| Exhaust uptakes | 160 | 240 | 160 | 240 |
| Tanks in engine room | 2450 | 2270 | 2400 | 2170 |
| Total costs | 482 475 | 458 021 | 482 831 | 435 285 |
| Total costs + 10 per cent profit | 530 722 | 503 823 | 531 114 | 478 813 |

The saving in total annual costs of the bulk carrier powered by medium-speed geared engines, as compared with the costs when powered by direct-drive slow-speed engines, in £ per annum is as follows:

| Generalized formulæ | | Typical figures | |
|---|---|---|---|
| | | £/year | Basis |
| Capital charges | $I$ per cent of £58 900 | 5890 | $I$ = 10 per cent |
| Lubricating oil | $0{\cdot}725d\,(L_{cr} + 1{\cdot}17L_{cyl} - 6{\cdot}8L)$ | −7600 | $d$ = 300; $L_{cr}$ = 5s. 6d.; $L_{cyl}$ = 9s. 0d.; $L$ = 7s. 6d. |
| Heavy oil | $1{\cdot}41Fd$ | 2110 | $F$ = £5/ton |
| Diesel oil | $0{\cdot}77Dd$ | 2200 | $D$ = £9 10s./ton |
| Maintenance | $615M$ | −615 | $M$ = £1/h |
| Total | | +1985 | |

A positive answer indicates a saving in favour of the ship powered by medium-speed geared engines.

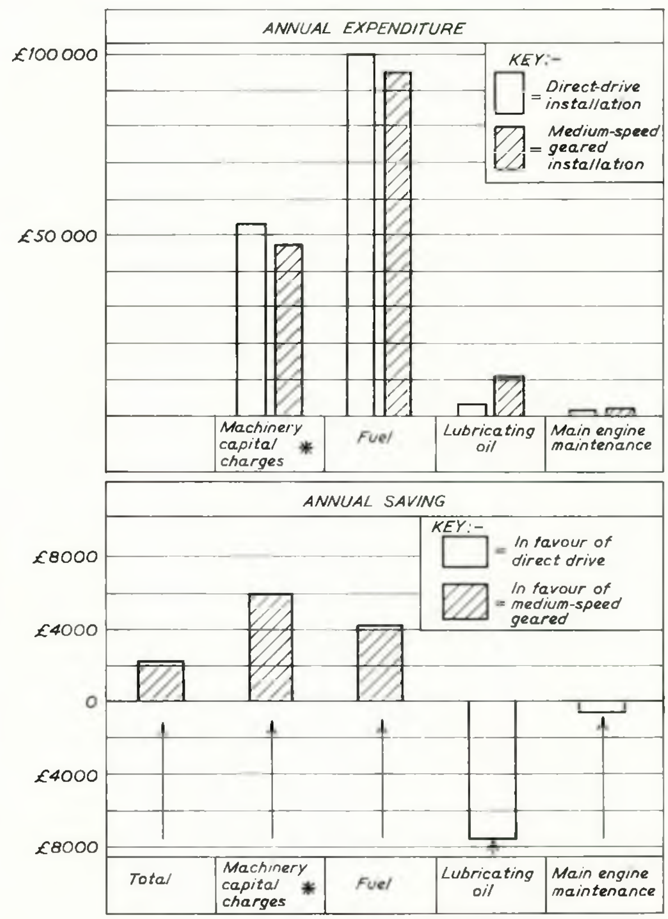
The saving in total annual costs of the cargo liner powered by medium-speed geared engines, as compared with the costs when powered by direct-drive slow-speed engines, in £ per annum is as follows:

| Generalized formulæ | | Typical figures | |
|---|---|---|---|
| | | £/year | Basis |
| Capital charges | $I$ per cent of £79 300 | 7930 | $I$ = 10 per cent |
| Lubricating oil | $0{\cdot}725d\,(L_{cr} + 1{\cdot}17L_{cyl} - 6{\cdot}97L)$ | −4200 | $d$ = 160; $L_{cr}$ = 5s. 6d.; $L_{cyl}$ = 9s. 0d.; $L$ = 7s. 6d. |
| Heavy oil | $0{\cdot}325Fd$ | 260 | $F$ = £5/ton |
| Diesel oil | $0{\cdot}77Dd$ | 1170 | $D$ = £9 10s./ton |
| Maintenance | $326M$ | −326 | $M$ = £1/h |
| Total | | +4834 | |

Again a positive answer indicates a saving in favour of the ship powered by medium-speed geared engines.

A comparison of the capital and operating costs for the bulk carrier and cargo liner is shown in graphic form in Figs. 10 and 11.

An indication is given below of the change in cost which would occur by the adoption of features other than those assumed in the geared installation.




* *Based on main engines and associated auxiliaries only and takes into account saving in hull cost in medium-speed engined ship*

FIG. 10—*Summary comparison—Bulk carrier annual expenditure*

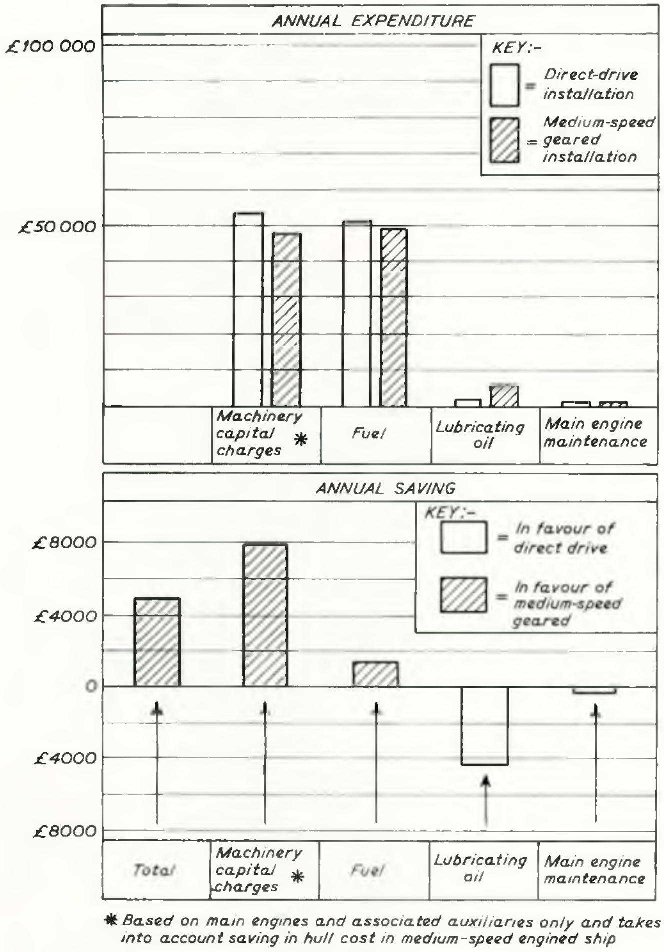



FIG. 11—*Summary comparison—Cargo liner annual expenditure*

A single engine geared to the propeller shaft gives a price reduction of £17 000 due to a simple gearing and transmission system.

If hydraulic couplings are used in place of friction clutches, the price is increased by about £3000, the annual fuel bill increased by £2140 for the bulk carrier and £1160 for the cargo liner, based on fuel at £5/ton, and the fuel stowage is increased by approximately 50 tons.

This would result in an annual loss of £455 in the bulk carrier and would reduce the annual saving in the cargo liner to £3374.

In the cargo liner, if the L.O. pumps are motor-driven instead of engine-driven, the price of generators is increased by £8000 and the annual fuel bill is increased by approximately £50.

## CONCLUSIONS

The possible widespread use of the medium-speed engines for merchant ship propulsion is conditional upon their ability to operate reliably on heavy fuels. There is evidence, from land-based engines and a smaller number of marine engines, which indicates that the engines under development and those already available will be capable of operating satisfactorily on heavy fuels, provided that the planned test programmes are carried out to determine the optimum conditions for burning such fuels and that the correct grades of lubricants are used.

As could be expected, the work reported in this paper does not show an overwhelming case either for, or against, the use of medium-speed geared Diesel engines in lieu of direct-drive Diesel engines for merchant ship propulsion. However, based on the data and assumptions used in the calculations, the geared engine installations utilizing the projected medium-speed engines do show up to economic advantage in each of the particular ships studied.

The net savings stem largely from reduced first cost of the machinery coupled with savings in expenditure on fuel oil, but offset by greatly increased expenditure on lubricating oil.

No doubt the companies concerned will endeavour to reduce the lubricating-oil consumption of the medium-speed trunked engine, but at this time there is no evidence to show that any large reduction will be forthcoming soon. Similarly, it is difficult to see how the fuel consumption of the medium-speed engine can be further reduced to any significant extent, unless of course the particular ship in question offers a greater improvement in propulsive efficiency at reduced propeller speed.

Thus, for the economic future of medium-speed geared engines for the propulsion of ocean-going merchant ships in general, it is vital to maintain the lower capital cost of these engines and associated machinery, and if possible to reduce it further as compared with direct-drive Diesel engines. The projected medium-speed Diesel engines are at the beginning of their development, and it is to be hoped that as service experience is gained it will be found possible to increase their rating, thus reducing the cost per bhp.

The gearing contributes a significant portion of the overall machinery capital cost. It would certainly be advantageous if a standard range of gearboxes could be offered to reduce design effort and streamline manufacturing processes. The curves shown in Figs. 6 and 7 suggest that the penalty for having to forego some freedom in the selection of propeller speed might not be too great.

The use of friction clutches as opposed to hydraulic couplings assists in reducing the capital and operating costs. Whilst these clutches are available in the required sizes, so far they have been in service at sea only in smaller ships and smaller powers. The projected tests of a friction clutch by M.O.D.(N) should provide valuable data within the next year or 18 months.

As an overall conclusion, therefore, it is felt that, as from now, shipowners might find it of advantage to investigate, for each projected new construction, whether they can obtain an economic advantage by fitting medium-speed geared Diesel engines. Such an investigation must, of course, pay due attention to the effect on ship design of savings in weight and space, and to the selection of the most economic propeller speed.

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Fife Forge Co. Ltd.
Harland and Wolff Ltd.
Holset Engineering Co. Ltd.
Alfred Holt and Co. Ltd.
Johnson Line Ltd.
Ab. Lindholmens Varv
Lloyd's Register of Shipping
Manchester Liners Ltd.
Metalastik Ltd.
Mirrlees-National Ltd.
M.O.D.(N)
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National Physical Laboratory (Ship Division)
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Richardson, Westgarth and Co. Ltd.
Ruston and Hornsby Ltd.
Serck Radiators Ltd.
Shell International Petroleum Co.
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## APPENDIX I

The table in Appendix IA gives a comparison of the approximate times required for planned maintenance of slow-speed and medium-speed Diesel engines. The table lists those items requiring maintenance at regular intervals of 1000 to 60 000 hours inclusive. Items occurring more frequently than 1000 hours have been regarded as routine watchkeeping operations. The relative distribution of the maintenance time for any

## APPENDIX IA

| MAINTENANCE REQUIREMENTS | 2 × 16-cylinder R. and H. AO | | 784—VT2BF—180 B. and W. | |
|---|---|---|---|---|
| | Periodicity | Man-hours | Periodicity | Man-hours |
| **Group I** | | | | |
| *Crankshaft and Main Bearings* | | | | |
| Check all main bearing nuts for tightness | 1000 | 12 | 1500 | 16 |
| Check crankshaft alignment | 4000 | 10 | 4000 | 6 |
| Examine main bearings | 8000 | 14 | 8000 | 4 |
| Change main bearings | 20 000 | 98 | 16 000 | 118 |
| Regrind crankshaft | 60 000 | 336 | — | — |
| **Group II** | | | | |
| *Crankcase, Sump* | | | | |
| Clean explosion door gauzes | 1000 | 4 | — | — |
| Clean sump and renew oil charge | 5000 | 16 | — | — |
| Clean double bottom L.O. tank | — | — | 8000 | 20 |
| Check bedplate holding down bolts | 8000 | 4 | 4000 | 4 |
| Check crankcase oil discharges | — | — | 2000 | 1 |
| **Group III** | | | | |
| *Connecting Rods, Bottom Ends, Telescopics, etc.* | | | | |
| Check crosshead clearance | — | — | 4000 | 6 |
| Check bottom end bearing clearance | — | — | 8000 | 6 |
| Dismantle crosshead bearing for inspection | — | — | 16 000 | 70 |
| Dismantle bottom end bearings | — | — | 16 000 | 70 |
| Dismantle and inspect S.E. and L.E. bearings and load reversers | 5000 | 24 | — | — |
| Piston cooling telescopic inspection | 8000 | 4 | 8000 | 4 |
| Renew S.E. and L.E. bearing shells | 20 000 | 50 | — | — |
| Examine L.E. bearings | 5000 | 4 | — | — |
| Dismantle and examine S.E. bearings | 5000 | 18 | — | — |
| Take up crosshead guide clearance | — | — | 60 000 | 120 |
| **Group IV** | | | | |
| *Camshaft Drive, Gearing and Chain Drives, etc. Manoeuvring Gear, Governor Gear.* | | | | |
| Examine overspeed trip and governor | 2000 | 6 | 8000 | 1 |
| Examine pump and camshaft drive gears or chain | 8000 | 8 | 8000 | 4 |
| Examine and maintain reversing gear | 8000 | 4 | 8000 | 20 |
| Renew drive gears, stub shaft or chain and chain gear | 60 000 | 20 | 60 000 | 42 |
| Renew worn parts of reversing gear | 60 000 | 54 | — | — |
| Fit reconditioned governor | 60 000 | 3 | — | — |
| **Group V** | | | | |
| *Camshaft and Fuel Injection System* | | | | |
| Change fuel injectors and service | 1000 | 34 | 2000 | 32 |
| Examine fuel pumps operating gear | 2000 | 1 | — | — |
| Examine camshaft bearings | 8000 | 34 | 8000 | 1 |
| Examine cams and followers | 8000 | 4 | 4000 | 2 |
| Inspect fuel pumps | 8000 | 16 | 8000 | 6 |
| Replacement of fuel pumps with shore reconditioned units as necessary | 30 000 | 40 | 30 000 | 12 |
| Dismantle and examine fuel feed pump | 8000 | 4 | 8000 | 1 |
| Remove camshaft, check profiles, etc. | 60 000 | 80 | 60 000 | 42 |
| Renew cam followers | 60 000 | 112 | 60 000 | 30 |
| **Group VI** | | | | |
| *Air Starting, Exhaust and Safety Valves* | | | | |
| Clean air start valves | 8000 | 36 | 8000 | 21 |
| Grind exhaust valves and examine springs | 5000 | 152 | 2000 | 126 |
| Renew exhaust valves, guides and springs as necessary | 20 000 | 12 | 4000 | 28 |
| Exhaust valve operating gear maintenance | — | — | 30 000 | 14 |
| Overhaul pilot valves, distributor etc., on s.a. system | 8000 | 12 | 8000 | 25 |
| Overhaul relief valves | 8000 | 36 | 8000 | 21 |
| Check clearance absorbers | — | — | 8000 | 4 |
| Renew valve gear, rocker bearings, etc. | 60 000 | 80 | 60 000 | 20 |

APPENDIX IA—(*continued*)

| MAINTENANCE REQUIREMENTS | 2×16-cylinder R. and H. AO | | 784—VT2BF—180 B. and W. | |
|---|---|---|---|---|
| | Periodicity | Man-hours | Periodicity | Man-hours |
| Group VII<br>*Turbochargers* | | | | |
| Clean air inlet filters | 1000 | 24 | 1000 | 8 |
| Strip and clean blowers (renew bearings) | 5000 | 84 | 8000 | 72 |
| Group VIII<br>*Air Intercoolers, Check Valves, Scavenge Spaces, etc., Heat Exchangers* | | | | |
| Inspect air chest and clean | 4000 | 23 | 1000 | 6 |
| Clean intercoolers air side | 8000 | 24 | 8000 | 16 |
| Inspect and clean heat exchangers | 8000 | 42 | 8000 | 30 |
| Descale heat exchangers, intercoolers, and pressure test | 60 000 | 16 | 60 000 | 12 |
| Group IX<br>*Pistons, Cylinders, Liners* | | | | |
| Check piston rings, liners through ports | — | — | 1000 | 1 |
| Withdraw pistons (remove crowns) clean, check rings | 5000 | 288 | 8000 | 210 |
| Measure liner wear | 5000 | 16 | 8000 | 4 |
| Clean cylinder lubricators | 5000 | 24 | 8000 | 14 |
| Examine cylinder lubricator pump and distributor | 5000 | 2 | | |
| Fit reconditioned flame plates | 20 000 | 28 | — | — |
| Fit reconditioned liners | 20 000 | 40 | 40 000 | 140 |
| Renew or replace piston bodies | 20 000 | — | 16 000 | 140 |
| Cylinder head inspection and cleaning including cylinder head water spaces | 5000 | 40 | 8000 | 14 |
| Group X<br>*Miscellaneous* | | | | |
| Renew or clean fuel oil filter element | 1000 | 2 | 1500 | 1 |
| Renew or clean L.O. filter element | 2000 | 3 | 2000 | 1 |
| Inspect exhaust duct | 8000 | 2 | 8000 | 1 |
| Check and tighten stay bolts | — | — | 8000 | 8 |
| Group XI<br>*Main Gearing* | | | | |
| Examine teeth | 4000 | 4 | — | — |
| Examine pinion bearings | 8000 | 32 | — | — |
| Examine main bearings | 8000 | 24 | — | — |
| Examine clutches | 4000 | 24 | — | — |
| *Total (Man-hours)*<br>For 60 000 hour operating period (including 60 000 hour service) | 17 384 | | 12 271 | |

specific task between shipboard and shipyard labour will be a function of the ship operator's policy, the vessel's ports of call and the type of ship. The required time for any specific item of maintenance in the table, therefore, includes the hours for both shipboard and shipyard labour. The costing of the labour required for the total maintenance of either installation is further complicated by the sub-division of both the shipyard and shipboard labour into skilled and unskilled components, each with different hourly rates of pay. The costing of this labour presents a problem in that, in the case of the shipboard labour, payment is not normally made on an hourly basis and, therefore, provided the maintenance requirements are within the capabilities of the number of crew on board, small variations in the hours to be worked by shipboard labour will not affect the cost of maintenance. The totals of man-hours required for maintenance give a rough indication only of the time required.

An arbitrary period of 60 000 running hours has been chosen as the approximate half life of the ship, and the total numbers of man-hours required for the listed maintenance items during this period have been computed. This gives the following totals:

i) 2 × 16-cylinder R. and H.AO 17 384 man-hours;
ii) 784-VT2BF-180 B. and W. 12 271 man-hours.

The difference between these two figures is 5113 man-hours, spread over an operating period of 60 000 hours, and is equivalent to:

615 man-hours per year for the bulk carrier;
326 man-hours per year for the cargo liner.

The same philosophy has been adopted for both engines in the table regarding the use of factory reconditioned spares in that, with the exception of fuel pumps and fuel injectors, all reconditioning of items of equipment removed from either engine is assumed to be performed by either shipboard or shipyard personnel. In the case of the fuel equipment, time has been allowed for the removal, cleaning, testing and replacement of fuel injectors or pumps (Group V) but it is assumed that reconditioning of these units is performed at the factory. If this trend towards using ship's personnel to remove and replace items that are maintained by the manufacturer develops, the medium-speed engine will be at a definite advantage, due not only to the smaller size of individual units, making transport to and from spares depots easier, but also to the quantity production of the medium-speed engine builder which should help to reduce the cost per item of the spares produced. Such a system is of course dependant on the development of an efficient and world-wide spares replacement and reconditioning system.

## APPENDIX II

*Report by Lloyd's Register of Shipping*

COMPARISONS BETWEEN GEARED AND DIRECT-DRIVE DIESELS

1) *Object*

To compare the numbers of reported defects (of certain categories) in geared and direct-drive main-propulsion Diesel engines of comparable ages and horsepower per shaft.

2) *Engines Studied*

The investigation has been confined to main propulsion installations at present classed with Lloyd's Register of Shipping.

Three groups of geared Diesel installations have been selected, i.e.:

| Group | A.1 | B.1 | C.1 |
|---|---|---|---|
| Built | 56-57 | 56-57 | 58-60 incl. |
| Hp per shaft range from | 2000 | 5750 | 6000 |
| to under | 4500 | 8250 | 6400 |
| No. of ships | 17 | 6 | 8 |
| No. of shafts | 17 | 8 | 12 |
| No. of engines | 31 | 18 | 24 |
| No. of cylinders | 252 | 196 | 264 |

Further particulars of these geared installations are given in Appendix IID.

The foregoing groups of geared installations have been matched as closely as possible (in relation to date of build, horsepower per shaft and position of machinery) by three corresponding groups of direct-drive Diesel installations, as follow:

| Group | A.2 | B.2 | C.2 |
|---|---|---|---|
| Built | 56-57 | 56-57 | 58-60 incl. |
| Hp per shaft range from | 2000 | 5750 | 6000 |
| to under | 4500 | 8250 | 6400 |
| No. of ships | 17 | 7 | 12 |
| No. of shafts | 17 | 8 | 12 |
| No. of engines | 17 | 8 | 12 |
| No. of cylinders | 96 | 51 | 70 |

Further particulars of these direct-drive installations are given in Appendix IIE.

It will be seen in Appendix IIE that it was not found possible to match some of the twin-screw geared installations closely with direct-drive twin-screw installations. In these cases, two direct-drive installations were chosen, and these are indicated by (a) and (b) in Appendix IIE. Also, it will be noted that some of the selected matching installations are slightly outside the age and horsepower ranges indicated above.

3) *Defects Considered*

The categories of defects considered are listed in Appendices IIA, IIB and IIC in which defects which could occur in either geared or direct-drive installations are distinguished from those peculiar to geared installations only.

The numbers of defects given include only those reported by the surveyors in the categories cited. The numbers therefore do not comprise all defects or troubles experienced by the installations since some of these may not have been within the purview of the surveyors, or have been in other categories than those listed.

Damages and defects due to external causes such as collision, grounding, fire or explosion have not been included, nor renewals due to normal wear and tear.

No attempt has been made to assign relative severities or importance to the reported defects.

4) *Period Considered*

The period considered, for each installation, has been the time between the date on installation to 31st June 1965 and the aggregate times of service given are calculated on the same basis.

5) *Results*

The numbers of reported defects are compared for groups A.1 and A.2 in Appendix IIA, for groups B.1 and B.2 in Appendix IIB and for groups C.1 and C.2 in Appendix IIC for each of the defect categories listed.

In each appendix, the aggregate months of service at risk are given separately for shafts, engines and cylinders in each group. These figures can be applied in calculating the incidences for the various categories of defects, depending on whether they are considerd to be governed by the service period at risk for shafts, engines or cylinders. Further comparisons must depend upon the severities or importances eventually assigned to each category of defect.

## APPENDIX IIA

*Group A* Geared Diesels built 1956-57 inclusive in the horsepower per shaft range 2000 to under 4500.

Defects are listed below for Group A geared Diesels and their direct-drive counterparts.

| | | Direct-drive Group A.2 | Geared Group A.1 |
|---|---|---|---|
| **a) In parts common to both types:** | | | |
| Crankcase | explosions | — | 2 |
| Crankshafts | cracked or broken | 1 | 1 |
| | journals corroded scored | — | 1 |
| Main bearings | white metal cracked, | — | 6 |
| | wiped, overheated, scored | 1 | 18 |
| | worn and corroded | — | 18† |
| | white metal bonding breakdown | — | 8‡ |
| Crankpin bearings | white metal cracked, | 2 | 7 |
| | wiped, overheated, scored | — | 2 |
| Crosshead bearing | white metal cracked, | 14 | — |
| | wiped | 1 | — |
| Gudgeon pin | cracked | — | 2 |
| | wasted | — | 1 |
| | locking device failure | — | 2 |
| Cylinder liners | cracked | 1 | — |
| | scored | — | 9* |
| Cylinder jackets | cracked | 1 | — |
| Cylinder covers | cracked | 10 | 19§ |
| | leaking | — | 1 |
| Piston | cracked | 1 | 1 |
| | leaking | 1 | — |
| | seized | — | 3 |
| | scored | — | 8* |
| | securing nuts slack | 1 | — |
| Bedplates | cracked | 1 | — |
| Thrust bearings | scored | — | 1 |
| **b) In parts associated with gearing only:** | | | |
| Gearing | teeth pitted | — | 3¶ |
| | gearbox/engine misalignment | — | 1 |
| Hydraulic coupling | shaft oil seal white metal hammered out | — | 1 |
| | coupling holed and leaking | — | 1 |
| | coupling bearing scored | — | 1 |
| **c)** Aggregate months service for shafts | | 1734 | 1734 |
| Aggregate months service for engines | | 1734 | 3162 |
| Aggregate months service for cylinders | | 9792 | 25 704 |

* Eight scored pistons and cylinder liners are included in one report on one engine.

† The 18 bearings affected were on the two engines of one geared installation.

‡ The eight main bearings with white-metal bonding breakdown were all in one engine.

§ 17 of these cracked cylinder covers were on two engines of one ship.

¶ One set of gears renewed due to pitting and the replacement set also suffered pitting.

## APPENDIX IIB

*Group B* Geared Diesels built 1956-57 inclusive in the horsepower per shaft range 5750 to under 8250.

Defects are listed below for Group B geared Diesels and their direct-drive counterparts.

| | | Direct-drive Group B.2 | Geared Group B.1 |
|---|---|---|---|
| **a) In parts common to both types:** | | | |
| Crankcase | explosion | — | 1 |
| Crankshafts | cracked or broken | — | 2 |
| | corrosion | 1 | — |
| Main bearings | wiped, overheated | 2 | 1 |
| Crankpin bearings | white metal cracked | 4 | — |
| | metal porous and missing | — | 27* |
| | wiped, overheated | — | 1 |
| | white metal cracked | 3 | 4 |
| Crosshead bearings | white metal cracked, | 7 | — |
| | wiped | 3 | — |
| Cylinder liners | cracked | 1 | 1 |
| | scored | 1 | 3 |
| Cylinder jackets | cracked | 2 | — |
| Cylinder covers | cracked | 4 | 2 |
| Pistons | leaking | — | 2 |
| | cracked | 9 | 1 |
| | seized | 1 | — |
| | burnt | — | 2 |
| | skirt scored | 1 | — |
| Bedplates and entablature | cracked | 1 | 4 |
| Thrust | overheated | 1 | — |
| **b) In parts associated with gearing only:** | | | |
| Gearing | teeth pitted | — | 2 |
| | pinion thrust bearing overheated | — | 1 |
| Hydraulic coupling | thrust bearings damaged | — | 1 |
| **c)** Aggregate months service for shafts | | 816 | 816 |
| Aggregate months service for engines | | 816 | 1836 |
| Aggregate months service for cylinders | | 5202 | 19 992 |

* The 27 main bearings affected by porosity and missing metal were spread over the four engines of one twin-screw geared installation.

## Appendix IIc

*Group C* Geared Diesels built 1958-60 inclusive in the horsepower per shaft range 6000 to under 6400. Defects are listed below for Group C geared Diesels and their direct-drive counterparts.

| | | Direct-drive Group C.2 | Geared Group C.1 |
|---|---|---|---|
| a) In parts common to both types: | | | |
| Crankshaft | broken | — | 1 |
| Main bearings | wiped | 4 | — |
| | white metal cracked | — | 9 |
| Crankpin bearings | wiped | 6 | — |
| | white metal cracked | 3 | — |
| Crankpin bearing bolts | cracked or broken | 1 | 1 |
| Crosshead bearings | wiped | 10 | — |
| | white metal cracked | 11 | 2 |
| Crosshead pin | corroded | 1 | — |
| Gudgeon pin | fractured | — | 2 |
| Cylinder liners | cracked | 4 | 2 |
| | porous | 1 | — |
| | leaking | 1 | — |
| Cylinder jacket | fractured | 1 | — |
| Piston | cracked | 16 | 1 |
| | burnt | 1 | — |
| | wasted | — | 12* |
| Cylinder entablature | fractured | 1 | 1 |
| b) In parts associated with gearing only: | | | |
| Gearing bearings | wiped | — | 1 |
| c) Aggregate months service for shafts | | 927 | 824 |
| Aggregate months service for engines | | 927 | 1648 |
| Aggregate months service for cylinders | | 5426 | 17 880 |

* The 12 wasted pistons represent all the pistons of one engine.

## Appendix IId

Group A—geared diesels built 1956–57 inclusive, 2000 to under 4500 hp/shaft

| Installation number | Date of installation | Hp/shaft | Ship type | Normal shaft rev/min | No. of engines/ shaft | No. of shafts | No. of cylinders/ engine | Position of machinery |
|---|---|---|---|---|---|---|---|---|
| A | B | C | D | E | F | G | H | I |
| 1 | 1956–4 | 2000 | cargo | 125 | 2 | 1 | 8 | aft |
| 2 | 1956–5 | 2000 | cargo | 125 | 2 | 1 | 8 | aft |
| 3 | 1956–12 | 2000 | cargo | 150 | 2 | 1 | 8 | aft |
| 4 | 1957–3 | 2000 | cargo | 150 | 2 | 1 | 8 | aft |
| 5 | 1957–11 | 2000 | cargo | 125 | 1 | 1 | 8 | amid |
| 6 | 1956–7 | 2240 | cargo | 150 | 2 | 1 | 6 | aft |
| 7 | 1956–7 | 2410 | cargo | 219 | 1 | 1 | 9 | amid |
| 8 | 1957–1 | 2410 | cargo | 219 | 1 | 1 | 9 | amid |
| 9 | 1957–12 | 2560 | tanker | 116 | 2 | 1 | 8 | aft |
| 10 | 1956–12 | 2600 | cargo | 110 | 2 | 1 | 6 | amid |
| 11 | 1956–1 | 3040 | cargo | 146 | 2 | 1 | 8 | aft |
| 12 | 1956–9 | 3060 | cargo | 120 | 2 | 1 | 8 | aft |
| 13 | 1956–11 | 3060 | cargo | 120 | 2 | 1 | 8 | aft |
| 14 | 1957–4 | 3360 | cargo | 100 | 2 | 1 | 8 | aft |
| 15 | 1957–4 | 3800 | cargo | 125 | 2 | 1 | 9 | amid |
| 16 | 1957–5 | 3800 | cargo | 135 | 2 | 1 | 10 | amid |
| 17 | 1957–11 | 4180 | cargo | 139 | 2 | 1 | 10 | amid |

Group B—geared diesels built 1956–57 inclusive, 5750 to under 8250 hp/shaft

| A | B | C | D | E | F | G | H | I |
|---|---|---|---|---|---|---|---|---|
| 18 | 1956–3 | 5760 | bulk carrier | 110 | 2 | 1 | 8 | aft |
| 19 | 1956–6 | 5760 | bulk carrier | 110 | 2 | 1 | 8 | aft |
| 20 | 1957–7 | 6000 | cargo | 125 | 2 | 2 | 12 | amid |
| 21 | 1957–12 | 6000 | cargo | 125 | 2 | 2 | 12 | amid |
| 22 | 1956–7 | 7360 | ore carrier | 95 | 2 | 1 | 10 | aft |
| 23 | 1957–1 | 8160 | icebreaker | 194 | 4 | 1 | 12 | amid |

Group C—geared diesels built 1958–60 inclusive, 6000 to under 6400 hp/shaft

| A | B | C | D | E | F | G | H | I |
|---|---|---|---|---|---|---|---|---|
| 24 | 1958–10 | 6000 | cargo | 125 | 2 | 2 | 12 | amid |
| 25 | 1959–6 | 6000 | cargo | 125 | 2 | 2 | 12 | amid |
| 26 | 1959–11 | 6000 | cargo | 152 | 2 | 1 | 12 | amid |
| 27 | 1960–3 | 6000 | cargo | 152 | 2 | 1 | 12 | amid |
| 28 | 1960–4 | 6000 | cargo | 125 | 2 | 2 | 12 | amid |
| 29 | 1960–12 | 6000 | cargo | 125 | 2 | 2 | 12 | amid |
| 30 | 1959–7 | 6360 | cargo | 115 | 2 | 1 | 6 | amid |
| 31 | 1959–7 | 6360 | cargo | 115 | 2 | 1 | 6 | amid |

## Appendix IIe

Group A—direct-drive diesels built 1956-57 inclusive, 2000 to 4500 hp/shaft

| Installation number | Date of installation | Hp/shaft | Ship type | Normal shaft rev/min | No. of engines/ shaft | No. of shafts | No. of cylinders/ engine | Position of machinery |
|---|---|---|---|---|---|---|---|---|
| A | B | C | D | E | F | G | H | I |
| 1 | 1956–9 | 2300 | cargo | 120 | 1 | 1 | 3 | aft |
| 2 | 1956–5 | 1970 | cargo | 300 | 1 | 1 | 7 | aft |
| 3 | 1957–10 | 1850 | cargo | 160 | 1 | 1 | 5 | aft |
| 4 | 1957–2 | 2000 | cargo | 155 | 1 | 1 | 5 | aft |
| 5 | 1957–12 | 2000 | cargo | 155 | 1 | 1 | 5 | amid |
| 6 | 1956–3 | 2850 | tanker | 160 | 1 | 1 | 8 | aft |
| 7 | 1956–7 | 2410 | cargo | 128 | 1 | 1 | 6 | amid |
| 8 | 1957–9 | 2410 | cargo | 128 | 1 | 1 | 6 | amid |
| 9 | 1957–9 | 2600 | ore carrier | 115 | 1 | 1 | 3 | aft |
| 10 | 1956–8 | 2770 | cargo | 160 | 1 | 1 | 6 | amid |
| 11 | 1956–7 | 2750 | collier | 165 | 1 | 1 | 5 | aft |
| 12 | 1956–11 | 3250 | cargo | 160 | 1 | 1 | 6 | aft |
| 13 | 1956–11 | 3120 | cargo | 150 | 1 | 1 | 6 | aft |
| 14 | 1957–6 | 3400 | cargo | 108 | 1 | 1 | 6 | aft |
| 15 | 1957–4 | 3850 | cargo | 155 | 1 | 1 | 7 | amid |
| 16 | 1957–7 | 3500 | cargo | 150 | 1 | 1 | 7 | amid |
| 17 | 1957–7 | 4100 | cargo | 150 | 1 | 1 | 5 | amid |

Group B—direct-drive diesels built 1956–57 inclusive, 5750 to under 8250 hp/shaft

| A | B | C | D | E | F | G | H | I |
|---|---|---|---|---|---|---|---|---|
| 18 | 1956–4 | 5500 | cargo | 125 | 1 | 1 | 7 | aft |
| 19 | 1956–11 | 5560 | cargo | 127 | 1 | 1 | 8 | aft |
| 20 | 1957–11 | 5800 | cargo | 118 | 1 | 2 | 5 | amid |
| 21(a) | 1957–6 | 6000 | cargo | 112 | 1 | 1 | 8 | amid |
| 21(b) | 1957–8 | 6000 | cargo | 125 | 1 | 1 | 9 | amid |
| 22 | 1956–9 | 7200 | cargo | 115 | 1 | 1 | 8 | aft |
| 23 | 1957–2 | 8200 | cargo | 170 | 1 | 1 | 6 | amid |

Group C—direct-drive diesels built 1958–60 inclusive, 6000 to under 6400 hp/shaft

| A | B | C | D | E | F | G | H | I |
|---|---|---|---|---|---|---|---|---|
| 24(a) | 1957–6 | 6180 | cargo | 108 | 1 | 1 | 6 | amid |
| 24(b) | 1957–12 | 5850 | cargo | 114 | 1 | 1 | 4 | amid |
| 25(a) | 1958–1 | 6000 | cargo | 112 | 1 | 1 | 8 | amid |
| 25(b) | 1958–10 | 6000 | cargo | 112 | 1 | 1 | 8 | amid |
| 26 | 1959–1 | 6000 | cargo | 130 | 1 | 1 | 6 | amid |
| 27 | 1959–2 | 6000 | cargo | 119 | 1 | 1 | 6 | amid |
| 28(a) | 1960–1 | 6000 | cargo | 112 | 1 | 1 | 5 | amid |
| 28(b) | 1960–3 | 6000 | cargo | 112 | 1 | 1 | 5 | amid |
| 29(a) | 1960–5 | 6000 | cargo | 112 | 1 | 1 | 5 | amid |
| 29(b) | 1960–10 | 5850 | cargo | 114 | 1 | 1 | 4 | amid |
| 30 | 1958–7 | 6400 | cargo | 113 | 1 | 1 | 6 | amid |
| 31 | 1957–11 | 6300 | cargo | 125 | 1 | 1 | 7 | amid |

*Note:* When matching twin-screw geared and direct-drive Diesels there was a shortage of twin-screw direct-drive installations. Two single-screw installations (marked a and b) were used to represent twin-screw installations.

# Discussion

MR. J. MCAFEE (Vice-Chairman of Council) said that the large audience present was a compliment to the authors, and an indication of the general interest in the subject. He had looked at the bibliography and on reading Kilchenmann's paper, to which the authors referred, found that he had investigated the basic technical arguments of the case for either direct or geared drive medium-speed engines. The authors of the paper under discussion, however, had gone into the matter much more deeply and had explored very fully the economic aspects.

Taking two specific sizes and types of ship, they had calculated the capital expenditure and annual running costs for direct-drive and medium-speed geared installations. For the purpose of this comparison, they had chosen two engines, one of which had a long history, whilst the other was still, to some extent, in the development stage. Some might think this a strange choice but it did not necessarily upset the validity of their calculations. In the end, however, as they pointed out, the figures did not produce any overwhelming argument and, in one instance, the balance was even dependent on the type of clutch assumed to be fitted.

What was the prospective shipowner to make of all this? In an effort to find out, the speaker had approached a company which had been successfully operating, for a number of years, various types of medium-speed geared Diesel engine, having previously had experience only with direct drive. On inquiring why they were now favouring the medium-speed engine, he was surprised to find that none of the usual pros and cons were advanced, instead, mention was made of the ease of manœuvring in restricted waterways, the advantage of being able to run at reduced speed on one engine only for short legs of a voyage, and the confidence felt in being able to return some thousands of miles to the home port in the event of a mishap putting one main engine out of action—a necessity which had arisen on more than one occasion. With a fuel consumption of about 12 tons per day, great importance was not attached to thermal efficiency and, indeed, none of the ships was adapted for running on heavy oil. In this attitude they might well be right, for, with many services, fuel costs were only a moderate part of annual total running expenses.

The views of these particular owners, coupled with the lack of any pronounced economic advantage between the two systems—as the authors showed—suggested that, for many ships, the case would be decided on factors which could not be arithmetically assessed in a balance sheet, such as advantage of smaller working parts, manœuvrability, flexibility of operation and ease of overhaul. An owner recently told him that, for two new ships now on order, engines of a particular make had been specified mainly because of the company's excellent experience of the makers' after sales service. Mr. McAfee compared this with the experience of another owner who suffered a failure of a coupling in the main transmission system of a comparatively new ship. The builders could not offer a replacement in under nine months and appeared indifferent to the owner's plight, faced as he was with the ship being out of service for this length of time. No doubt the authors had this kind of thing in mind when they said that reduction of maintenance costs would be dependent on a world-wide spares and conditioning system. His own feeling of certainty that they were right in this was balanced by an equal conviction that his generation was unlikely to see it come to pass.

Whatever the basis of argument, there was no doubt, from the record of ships recently completed, that medium-speed geared installations were on the increase, both in numbers and maximum power. The most powerful set classed with Lloyd's Register of Shipping was fitted in the recently completed *Tor Anglia*, which had four Pielstick engines totalling 22 320 bhp. Here, as in similar large powered installations, manœuvring was carried out by controllable-pitch propellers, though the engines could be declutched. Friction type clutches for manœuvring had not been approved for powers exceeding 3000 bhp and the authors' proposal to fit such clutches for manœuvring purposes, with powers considerably in excess of this, could not yet be supported by sufficient operational experience.

Gearing was not a problem. The modern hardened and ground oil engine sets had a very good operational record.

The authors had quoted in full Lloyd's Register service defect list. Although the period covered was nearly six years previously, a random check of the period since then showed that the position was unchanged. He was not sure, however, how relevant to the general argument were all the defects recorded and analysed. What really mattered was the kind of defect which could cause delay—a broken shaft, damaged gears, cracked liners, covers, and/or bedplates. A wiped bearing or a scored piston might be of no significance—the record did not say. The authors, incidentally, might be questioned on the strict logic of making comparisons based on percentage failures of the number of parts at risk. It could always be argued that, for any given engine, the greater the number of parts contained in it, then the greater the probability that some failure would take place.

In the appendix, a brave attempt had been made—no doubt with the full support of the engine builders—to state the hours of service after which various engine parts would need to be overhauled. This might be fair enough for comparative purposes, but it seemed to imply that all would be well until the appointed hour. He found it hard to share the authors' optimism, since, in life, things seldom worked out so simply.

Having said these things the question posed at the beginning—what should the shipowner do?—still remained. Perhaps the answer was simple—he should be present at this meeting to listen to the debate which this interesting and provocative paper would arouse.

MR. R. C. THOMPSON, C.B.E., M.A. (Member)* said that on page 89 there was a reference to the comparison of the two types of main propulsion machinery being made in the context of carefully selected ship types. The words "from the point of view of showing medium-speed geared Diesels to the best advantage" seemed to have been omitted.

To illustrate this, he would mention that he considered that it was more likely that the Doxford six-cylinder J76 engines would be running in service, developing 14 400 bhp continuously when burning heavy fuel of 3500 sec Redwood 1 at 100°F,

*It is regretted that we have to report the death of Mr. Thompson on 9th March 1967.

than either of the two engines referred to in the paper as being under development for this duty. Certainly to date, no maker had been found willing to offer a medium-speed Diesel for delivery in 1968 for this duty when burning this heavy viscosity fuel.

Secondly, the statement that slow-speed engines required larger electrical loads from larger Diesel generators was misleading. Both types could have engine-driven pumps.

Thirdly, the comparison of first costs was misleading in that if this was true, it would equally apply to slow-speed had been chosen for comparison. For this engine, a specific price was given of £19 5s. 0d. per shp, compared with the Ruston and Hornsby AO at £17 2s. 0d. per shp. One could, however, today order direct-drive marine engines at £16 8s. 0d. per shp, equivalent to a saving in the example shown of £46 200. This engine would also be 140 tons lighter than the example used in the comparison.

Fourthly, on page 94, it was argued that, since the advent of high-duty detergent lubricating oils, it was now permissible and safe to revert to trunk-piston engines without a diaphragm and to allow the products of combustion, which blew past the pistons, to enter the crankcase. He would merely point out that if this was true, it would equally apply to slow-speed engines.

With regard to noise, he had normally found that the most objectionable noise in a slow-speed Diesel engine room now came from the medium-speed Diesel generators.

The whole paragraph on propellers was completely misleading in that it dealt only with propeller efficiencies when the vessel was fully loaded; both types of ship under discussion were likely to spend a good part of their sea time under partly loaded condition or in ballast. The best ballast speeds were attained by a vessel carrying the minimum amount of ballast necessary to prevent pounding and avoid propeller racing; with regard to the latter, within limits, the smaller the screw diameter the better. The choice of propeller diameter should take both loaded and ballast conditions into account and, in his opinion, the optimum diameters for the bulk carrier and cargo liner described would be about 22 ft and 20 ft respectively and he did not agree with the general statement that the slowest running propeller was the most economic.

With regard to statistical data in machinery failure, why did the authors cloud the issue with statistics related to the number of parts at risk? A shipowner was only interested in the number of failures per ship. Presumably the authors dared not publish available information on this basis and, of course, no information was available for new types not yet developed. He would like to point out that with 32 cylinders coupled to a single shaft, it was impossible to detect incipient seizure of a single piston which could easily have catastrophic consequences.

In his opinion, the best case for a medium-speed geared Diesel installation had not been made out. In a type of ship such as a train or car ferry, where headroom was important, a geared Diesel was the obvious answer, but in the opinion of his company's design team, the best propulsion would comprise a unidirectional engine or engines associated with a controllable-pitch propeller and having alternators driven from the gearbox. The installation would be all electric and all steam plant would be eliminated. This gave the best value for money at the present time, provided the limitation was accepted that the fuel must not have viscosity greater than Class B fuel.

Finally, he confessed to being somewhat envious of firms who received Government development contracts for military purposes but which were also of value commercially. He thought it would well be in the national interest for the Government to give a development contract for a slow-speed heavy oil engine capable of carrying high-viscosity fuel having a viscosity of, say, 8000 seconds Redwood 1 at 100°F. There was already considerable interest in the use of such fuel on the Continent.

Mr. W. Lowe, B.Sc., said that the authors had dealt very adequately with their comparison of the slow-speed direct-drive engine against the projected two-stroke medium-speed Diesel engine designs. However, it must not be forgotten that the objectives given in their synopsis, of low cost and reliability of operation on heavy fuel, could be met by a medium-speed four-stroke engine already on the market at the powers for which the design study had been made.

In considering the potential achievements of the two proposed two-stroke designs, it would be naive to assume that four-stroke development would stand still during the next few years until the projected two-stroke engines became commercially available. The comparison in Table II used a b.m.e.p. of just over 200 lb/in$^2$ for the KV Major four-stroke engine and there was no doubt that powers considerably higher than that would be available in the immediate future. It would not be long before the commercial ratings of 250 lb/in$^2$ b.m.e.p. were commonplace and then 300 lb/in$^2$ b.m.e.p., within the period envisaged in the paper for two-stroke development. There was yet no foreseeable limit beyond that figure for the future of the four-stroke engine.

In the particular ship applications considered in the paper, one should consider the effect of a four-stroke engine of the same speed and power, but of two-thirds the physical size and weight, to make a fair comparison. A point that was often overlooked was that, while the engine size for a given power was progressively becoming less and less, cooling equipment generally remained the same size and was of quite significant proportions compared with the engine size. Because of the intrinsic nature of heat-exchange equipment, which used low velocities and temperature differences, it was difficult to see how that could be avoided. There was a need for some breakthrough in the field of heat-exchanger equipment.

It could be seen from Table III that, as specific power increased, engine size and weight became progressively less significant when the total engine room machinery was considered. The significance of engine weight became even less when the weight of fuel carried was taken into account and reference to the fuel quantities (shown on the right of Table III) made that abundantly clear.

The costs of the machinery installation, shown in Table IV, illustrated that the main engine cost was a large proportion of the total, so that the continuing development to higher specific powers resulting in a cheaper engine per horsepower would have a significant effect on the total cost.

Mr. J. F. R. Ellison (Member) said that the authors were to be commended on the immense amount of data which they had collated and analysed to show that, at the present time, there were certain points in favour of medium-speed geared Diesel propulsion for ocean-going vessels, especially those which also covered a lot of river and canal passages.

No doubt the Ministry of Technology was anxious to further the development of that type of propulsion as there was a number of suitable British engines already in production or being designed for that purpose, and that was as it should be as—with due respect to Continental friends—already far too much money went out of this country in the form of royalties for 90 per cent of the Diesel engines built and installed in vessels produced in the United Kingdom. However, having said that, there was the shipowner's point of view to be considered and the features most important to him were reliability, ease and cheapness of maintenance, and economy in running costs. On that basis, there really was very little argument in favour of geared Diesel installations and the speaker did not think the authors would entirely disagree with that statement in view of their remarks in the second paragraph of their conclusions in the paper.

The speaker's company had had considerable experience with geared Diesel installations over the past fourteen years and, although they would now come under the category of slow-speed geared Diesels, being of the order of 230 engine rev/min, down to 100 rev/min, at the propeller, and, therefore, not directly comparable with those discussed in the paper from a space, weight and cost point of view, nevertheless, there was a number of features common to all such installations and worthy of mention.

In the first place, Mr. Ellison felt he should briefly explain the reasons why his company, engaged in the refrigerated meat trade from Australasia to the United Kingdom, decided to adopt that form of propulsion.

In the early 1950s they were still in the process of replacing their wartime losses and the highest powered single-screw motor vessel in their fleet developed a modest 7200 bhp at 114 rev/min, their naval architect was anxious to retain the more efficient single-screw arrangement, but they required larger and faster vessels, and there was no suitable direct-drive Diesel engine available which would give the required output of 11 500 shp, as that was before the days of the supercharging of large slow-speed engines. The solution, therefore, appeared to be two 12-cylinder 580 mm bore Sulzer reversible two-stroke crosshead engines geared to one shaft through magnetic slip couplings and quillshafts.

The advantages claimed for a geared Diesel installation at that time were, in the first place, low headroom, as used to considerable advantage in the former Rotterdamsche Lloyd vessel *Willem Ruys*, which they had studied in some detail, but that feature was not of great significance in a single-screw cargo vessel or tanker.

Next came very rapid manœuvring at half power by running one engine ahead and one astern, thus also saving starting air and the thermal stresses caused by frequent injection of cold air into a hot cylinder. The disadvantage of this method was that no Diesel engine took kindly to idling for long periods and the result was carbon build-up in ports and passages, frequent exhaust manifold fires and general fouling-up of the engine. With regard to the injection of cold starting air, the speaker doubted whether any case of cracked liners could be attributed to that cause.

Finally, there was the possibility of shifting berth or coasting, using one engine while the other was being overhauled, or for reasons of economy when speed was not essential. That facility was there, but advantage was seldom or never taken of it. Neither their friends on the bridge nor ship repairers favoured the idea. The former for reasons of safety; the latter from a continuity point of view, e.g. a new squad of men trying to find the various parts and losing much valuable time in becoming conversant with what had already been done.

The authors rightly pointed out that hydraulic and magnetic slip couplings were the most desirable type from the engine aspect, but both had disadvantages: the continuous slip increased the fuel consumption appreciably, and the consumption of electrical power for the slip couplings amounted to 80 kilowatts. The slip-rings for feeding the current to the outer members gave continuous trouble due to arcing and flats forming on them, although the speaker said that his company was assured by the makers that they would never require attention, except for normal cleaning. The couplings used by the company were still the largest of their type to be fitted to any vessel and every one of them—ten in all—had developed cracks in the main disc plate of the outer member, due, it was thought, to flexing when accelerating and decelerating the heavy masses of coils mounted on the inside of the outer barrel.

The quillshafts were connected at the aft end to the pinions by small tooth flexible couplings and trouble had been experienced on four of the vessels with the keyed connexions for the hubs of those couplings, which had partially sheared, and severe fretting had taken place between them and the shaft.

The authors appeared to favour friction clutches which would certainly show a saving in fuel, but would they say whether they considered that that type of clutch was capable of smoothing out the torque variations from the Diesel engines? Mr. Ellison said his company had considered several types of friction clutches, but abandoned them for fear of gearing damage from that source.

With regard to the engines themselves, many troubles experienced had been associated with the scavenge pumps and scavenge-pump driving arms which, of course, was not applicable to the modern medium-speed supercharged engines. Nevertheless, it was noted that the man-hours expended on maintenance, as quoted in Appendix IA, were considerably higher for the geared Diesel installation, and Mr. Ellison would certainly agree with that assessment, from the experience of his company.

From examination of Appendices IIA, B and C, there did not appear to be any set pattern to the defects experienced in either direct-drive or geared installations, but it was clear that the lower powered geared installations had many more defects than the equivalent powered direct drive, whereas in the highest powered category the reverse was the case. Could the authors say if that was due to higher engine revolutions being used in the low power range?

Most of the installations described in the paper involved fairly recent designs of engine. It remained to be seen how the maintenance requirements would develop over the years of service and, while it was evident that savings were effected in weight, space and initial cost, the speaker felt that he had said enough to show that in his opinion, the large-bore direct-drive Diesel was, at present, the best proposition taking the long-term view (over the life of the ship, that was of 20 or 25 years) for fast single-screw cargo liners and indeed for bulk carriers and tankers up to a limit of, say, 30 000 shp.

MR. J. F. BUTLER, M.A. (Member) said that the authors had made a praiseworthy and serious attempt to make a realistic comparison between the costs of direct-drive and medium-speed geared Diesel engines for marine propulsion.

Unfortunately, insufficient was known as yet of the performance of high-output medium-speed engines for propulsion purposes at sea. It was assumed that such engines could burn heavy fuel with the same facility as direct-drive engines, but the success of burning heavy oil in slow-speed engines had been due to the complete separation of cylinders and crankcase, and using highly alkaline cylinder oils which would be unsuitable for use in the crankcase. That separation was not possible in medium-speed trunk engines, so a compromise oil was necessary and, since alkaline oils could not be water washed, it would be necessary to replace the whole oil charge at intervals, so adding to the already high lubricating-oil consumption.

Mr. Butler said that one had to agree with the authors' conclusion that no large saving in lubricating-oil consumption in medium-speed engines could be expected in the near future and, in fact, it was clear that their overall lubricating-oil consumption would always be much higher than that of low-speed direct-drive engines because of the much greater wetted area swept by the pistons.

The savings suggested were so marginal in relation to total running costs that they hinged acutely on the accuracy of the assumptions made. For instance, the Doxford 76J7 engine mentioned in Table II was considerably more powerful than the others shown and could produce the power required, which worked out at 14 280 bhp at 104 rev/min. Since the specific fuel consumption of heavy oil at that power would not be higher than 0·365 lb/bhp-h, using the authors' figures for propeller efficiency and other factors, produced a calculated saving of £3700 per annum for the direct-drive engine in the bulk carrier, instead of a loss of £2000 per annum. For this comparison the auxiliary generator fuel consumption had been taken as the same in both cases. The argument that the medium-speed engine required less auxiliary power, because of main engine driven lubricating pumps, was not valid because, in a ship of the size considered, all the auxiliary power required at sea could be produced by an exhaust steam driven turbo-alternator.

In view of the marginal difference in costs between the two engine types, the choice of engine had really to depend on reliability and ease of maintenance. Since the medium-speed installation would have had 32 cylinders and pistons as against 7 and 14 respectively for the slow-speed engine, 128 exhaust valves against none, 32 fuel valves against 14, and 82 connecting-rod and journal bearings, against 50, one could not help feeling that the slow-speed direct-drive engine would always be more acceptable except for special-purpose ships where low headroom was a decisive advantage.

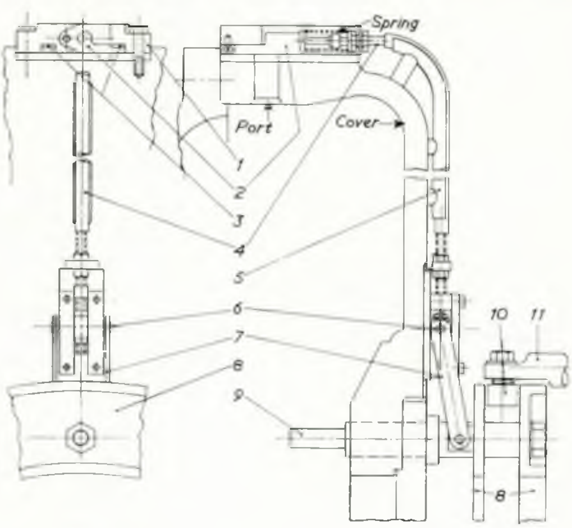
Mr. K. Maddocks, B.Sc. (Member) said, in a contribution read on his behalf by Commander F. J. Corney, O.B.E., R.N. (Member), that the weakness in the argument propounded by Mr. Neumann and Mr. Carr lay in the comparison of performance figures already achieved by the slow-speed Diesel engine with extrapolations and/or design objectives for the medium-speed Diesel engine. While it was inevitable that any forward-looking study must rely on the crystal ball to some extent, one was forced to assess that the conclusions drawn were unfairly biased towards the medium-speed Diesel.

The last paragraph of the authors' conclusions was of particular significance. His company's recent experience in similar comparative studies for various owners had shown that the deciding factor in favour of either the slow-speed or geared Diesel drive arose from the specific service conditions of the vessel.

The information available to the authors on modern slow-speed Diesel installations was apparently limited. For example, the 810 kW quoted for the electrical load of the slow-speed engined bulk carrier, compared with 400 kW to 450 kW as measured in service. The assumption that electrical power was generated by independent Diesel sets neglected the economy that was regularly achieved in using a waste-heat economizer with a turbo-alternator to carry all normal sea load. Further, the suggestion that the power requirement of an independent main-engine, lubricating-oil pump was a significant factor in the increase of electrical loading, warranted closer scrutiny. It would therefore be of interest to see a make-up of the electrical loading, particularly the normal sea load of all main engine service pumps, used by the authors.

The authors had selected an optimum propeller speed of 68 rev/min for the bulk carrier. It would be of interest to have the authors' comments on whether adequate consideration had been given to the margin for avoiding propeller-excited hull vibrations in view of this restriction of range of propeller revolutions.

In discussing various transmission systems it was surprising that, while acknowledging that the hydraulic coupling was the most popular form of coupling, the authors had elected to dismiss this well-proven unit in such a brief manner. It was axiomatic that the coupling slip (which need not be in excess of 2½ per cent and might be less) involved an equivalent increase in fuel consumption. However, from service experience of more than 20 years it could be factually stated that the maintenance costs on hydraulic couplings were virtually negligible. It was difficult to imagine that any combination of friction clutch and elastic coupling was likely to establish such a reputation and it was conceivable that maintenance costs during the life of a ship would largely offset the saving in fuel consumption.




1) Valve block
2) Slide valve
3) Needle roller tracks
4) Ball tracked operating cable
5) Protecting sheath
6) Crosshead
7) Drag links
8) Pressure plates
9) Guide rod
10) Needle roller
11) Actuating rod

Fig. 12

The well-acknowledged characteristics of the hydraulic coupling in providing:

a) complete isolation of torsionals;
b) shock-free connexion and disconnexion of the drive;
c) elimination of alignment problems;

had now been enhanced by a recently-patented system of slip control, developed by the contributor's company and illustrated in Fig. 12.

The philosophy of this design was based on locating the control valves on the periphery of the coupling—where the fluid was most effective. The full scale test rig results had shown:

1) ability to vary the slip up to 50 per cent, which permitted adjustment of propeller rev/min relative to engine speed;
2) disconnecting (coupling emptying) times of the order of a few seconds;
3) ease of remote control of the system.

These results indicated that the hydraulic coupling had the potential for retaining its lead in popularity. There was no reason to doubt the authors' statement that existing gearing technology was adequate to serve the requirements of the geared Diesel engine. Thus to complete the trio of elements required, he awaited with interest the service performance of medium-speed prime movers—using heavy fuel.

Mr. G. Victory (Member) said that some of his points had been covered by previous speakers, but he wanted to proceed, as he felt that they could not be considered to be entirely unbiased in their remarks. That was a claim which he hoped he could make with distinction.

The authors had not chosen to base their arguments on the use of medium-speed installations in smaller passenger and car ferries, or in those special category vessels in which headroom in the machine spaces was at a premium. Had they done so most people would have agreed that the medium-speed engine had some advantages. Instead the authors had chosen to argue their case on the specific examples quoted—a 16-knot bulk carrier of 55 000 dwt and a 20-knot cargo liner of 12 000 dwt—and it was in consideration of these specific categories that Mr. Victory would examine the case presented.

The authors said that the conclusions were based on the data and assumptions used in the calculations. Those few apparently innocuous words were the keystone on which they had prepared their case and the case they presented should be examined to see whether they had built on solid ground.

Mr. Victory's personal opinion was that many of the assumptions were open to dispute and that much of the data appeared to have been taken in such a way as to minimize the advantages of the slow-speed engine and to maximize those of the medium-speed engine. The disadvantages of the medium-speed engine in respect of noise, vibration and added maintenance had to a great extent been brushed aside, although no anti-vibration mountings or additional insulation had been allowed for in the paper. Mr. Butler had confirmed the view that, based on equally valid data and assumptions, the protagonists of the slow-speed engine could produce a document showing a substantial balance in favour of that engine.

The speaker felt that some of the reasons behind the assumptions used were a little difficult to understand. For example, why did the authors choose 16 000 bhp as the power required, and not the 17 500 bhp available from the Doxford engine mentioned in Table II, which would have yielded very different results? Why did they choose the Burmeister and Wain engine for comparison? The Doxford engine quoted was not only 91 tons lighter and 13 per cent less in volume, but, as mentioned, had an additional 1600 bhp in hand, and Mr. Victory imagined a distinct first cost saving. A Doxford engine of 15 000 to 16 000 bhp would show even greater advantages, including reduced volume and cost and a weight saving of about 150 tons over the engine considered and Mr. Victory understood that such an engine was available. It seemed unreasonable that the slow-speed engine should be penalized by as much as 150 kW of electrical power with all the consequent penalties, in respect of size and cost of generator and use of more expensive fuel etc., in order, it appeared, to drive the lubricating oil pumps.

How was it that the disadvantage of the medium-speed engine in respect of fuel and lubricating oil could by a piece of paper computation be shown to be no disadvantage at all? Mr. Victory, personally, thought that the difference in specific consumption had been underplayed and the figure quoted for the slow-speed engine could be improved upon.

The lubricating-oil consumption over the running periods quoted was about 10 tons higher for the medium-speed engine, yet the slow-speed engined ships were penalized by some 40 tons of additional lubricating oil storage capacity. Was the extra reserve of 50 tons really necessary?

The "crunch" finally came when, by using that 50 tons, plus the 150 tons, which would not have been available if a Doxford engine of similar power had been considered, and a good deal of saved machinery space volume, much of which was either unusable or could have been saved in any case by better room design, the authors came up with a reduction in displacement of 770 and 590 tons for the medium-speed engined ships. They then designed new ships to fit those dubious figures and found that their smaller medium-speed engined ships actually used less fuel, despite a higher specific fuel consumption, than did the medium-speed engined ships designed for the same capacity.

Mr. Victory did not think that anybody would disagree with a conclusion that there was not an overwhelming case for or against the medium-speed installation in respect of special types of ship, ferries and smaller short sea service vessels. However, there were solid grounds for disagreement over the case as presented in respect of the large bulk carrier and the high-speed cargo liner.

COMMANDER K. I. SHORT, O.B.E., D.S.C., R.N. (Member) made a verbal contribution at this point. An extended and more detailed version of his contribution now appears in the Correspondence section of the discussion.

MR. N. FLETCHER, referring to the paper, said that the need to complete a study in such depth within the time available had, he felt, resulted in the Mirrlees National opposed-piston engine being penalized rather heavily with respect to size and weight. That arose because it was necessary to use an OP16 engine, or twin OP8 engines derated from 20 000 hp to compare with the twin Ruston AO16 engines which had a maximum output of 8000 hp each at the present stage of their development.

The two ship types selected were then designed, using the slow-speed B. and W. engine with a service power of 14 400 hp. That was 90 per cent of the maximum continuous rating. As described in the paper, the vessels were then re-designed to take full advantage of the more compact medium-speed engine, driving the propeller at an optimized speed. It was not until that stage had been reached that the actual service powers of the medium-speed engine were established, these being 14 040 hp for the cargo liner and 13 800 hp for the bulk carrier.

Both those powers were below the maximum continuous rating of an OP 12-cylinder engine which produced 15 000 hp, but, unfortunately, time did not permit the vessels to be further re-designed to take advantage of that smaller and lighter unit. However, if that had been the case, it was most probable that an OP12, or twin OP6 engines would have proved suitable for the ships considered and, referring to Fig. 1 of the paper, the overall length would have been 37 ft 4 in for the single engine and 27 ft 8 in for the twin engines.

Similarly in Table II, the engine weight would have been reduced by 30 tons, giving, in the case of the single unit, a combined engine and gearbox weight of 153 tons.

The speaker felt that the foregoing had served to illustrate that tailoring the vessel to take full advantage of a medium-speed engine could result in considerable savings. Apart from the reduced power requirement of the lighter ship, a direct gain was given by the increase in propulsive efficiency provided by the slower turning propeller. Page 96 of the paper gave that figure as 5·5 per cent for the bulk carrier and 2·3 per cent for the cargo liner, but commented on the possibility of greater improvements with ships having higher block coefficients.

As the percentage reduction in fuel consumption was almost equal to the percentage increase in propulsive efficiency, it would be most interesting to hear what figures could be expected for such vessels with powers up to 40 000 hp such as could be provided by twin OP16 engines.

MR. P. JACKSON, M.Sc. (Member of Council) said that the paper was based on untried engines; the Ruston AO had only just completed its trials and had not even been tried on land before being applied to marine propulsion, while the Mirrlees engine was only on paper as yet. So many paper engines never saw service. He had himself designed a few in his day.

On the particulars given in the paper, the Mirrlees engines did not seem to have any advantage over the Ruston, certainly not for 16 000 hp, neither in dimensions, cost, weight, nor economy. So, for the moment at least, he could concentrate on the AO engine.

The AO engine was very similar to the Mitsubishi UET engine which had been available since 1961 (the paper said so), but Mr. Jackson could not find a single application of that engine to merchant service. Why was that? It was because the Japanese knew what they were doing and did not offer such an engine for such an onerous service. They had applied it for naval use for which it was suitable. Equally, the Ruston AO engine was suitable for naval service.

Geared propulsion had been in existence for at least 30 years. M.A.N. and Sulzer had both designed engines for geared propulsion. They were builders of marine engines and knew what they were doing for marine service, more so than the builders of medium-speed land engines, who thought they could get into the mercantile propulsion business with geared engines.

The medium-speed engines of M.A.N. and Sulzer had not really affected their large engine output. They had been applied to the vessels for which they were suitable, ferries, coasters and pleasure vessels. Even for such vessels, Mr. Jackson would not choose a two-cycle engine; he would compare the Mirrlees K engine or the Crossley-Pielstick, but not the type of engine on which the paper was based. Geared engines would fail on reliability. Many companies had tried them, the Chairman's company had three geared ships, *Cornwall, Middlesex,* and *Sussex,* and Mr. Jackson had heard the Chairman say "never again".

Similarly he could compare the experience of somewhat higher speed engines with even greater advantages, which had been put into two ships, the Deltic engines put into *Bahama King* and the Tasmania ferry *Bass Trader.* Mr. Jackson remembered reading in *Lloyd's List* that *Bahama King* had put into Port of Spain; "Parts of number 1 engine would be put into number 2; number 3 engine was out of action; the vessel would return to the U.K. on numbers 2 and 4 engines, when all engines would be replaced. That would be the twenty-eighth change of engine". Who would buy a car requiring so many changes of engine?

The initial development of the two engines, which was outlined in the paper, was commenced some eight years ago when the Ministry of Transport sent a party round the marine engine establishments to try and help them out of the depression from which the industry was suffering. The leader at that time was a previous Chairman of Council, the late Mr. B. P. Ingamells, C.B.E. Later the scheme was put in the hands of the Department of Scientific and Industrial Research who asked various engine companies for proposals. Mr. Jackson's company put forward the proposal that they should be given assistance towards the development of a still larger engine of over 3300 hp/cylinder. That proposal was considered by a committee under the Chairmanship of Vice-Admiral Sir Frank Mason, K.C.B., but the committee considered that the proposition did not require research, being a safe development, and D.S.I.R. could only make a loan for research. So money and help was given to Rustons and Mirrlees-National for their highly experimental and risky ventures.

Somewhat later, the National Research and Development Committee did offer a loan, but on terms far worse than could be obtained from normal commercial banking facilities at that time. What had been the result? Mr. Neumann had called the large engines "cathedrals", but what had been developed? These "pups" to compete with the Continental "cathedrals", with the Sulzer, Burmeister and Wain and M.A.N. large engines. The British marine engine had to compete with these well-known Continental engines and these developments would not do it; they were "red herrings". Mr. Jackson's personal opinion was that it was reprehensible for a Government department, the Ministry of Technology, to spend public money on backing these ventures to compete with the existing marine engine industry and divide it, whereas the Ministry of Transport's initial aim had been to aid it.

A report had been prepared by the Yarrow-Admiralty Research Department, as propaganda for these geared engines, which was sent to all shipowners and was again paid for by public money. The paper was a less biased, condensed, version of that report.

There was not time to deal with the many details in the paper. On the question of electrical power for large engines, it came back to the power which those who built large engines said was necessary for driving the pumps. Both types of engine had the same heat to deal with, but the makers of large engines quoted 30 per cent, or 50 per cent higher power than the makers of medium-speed engines, because they knew the marine trade and knew that the pumps had to develop their output even under worn conditions, or even if they were fouled.

All pumps used to be driven from the main engines, but that was dropped many years ago because, if a pump broke down, it could be necessary to shut down the main engine until the pump was either disconnected or repaired. Engine-driven pumps were dropped and, likewise, they would have to be dropped from the medium-speed engine as experience was gained.

On the question of the length of the engine room, it was true that medium-speed engines were much shorter than slow-speed, but the length of an engine room was not nowadays controlled by the main engine; it was controlled by the auxiliaries. Even with the so-called "cathedrals", the engine room was often up to ten feet longer than was necessary for the main engine, in order to accommodate the auxiliaries.

The lubricating-oil consumption of medium-speed engines was much heavier than on slow-speed engines. His company knew that from auxiliaries which were using at least three times as much oil as even the large amount that had been quoted in the paper. Heavy oil had not been used by medium-speed engines for 12 years but for more like three years. It was only about 14 years since the large engine had begun to use heavy oil.

On the question of noise, medium-speed auxiliaries were the noisiest machines in an engine room and the two-cycle engines advocated in the paper would be more noisy.

It was a good paper spoilt by the examples which it advocated.

Mr. B. E. Welbourn (Associate Member) said that clutches and couplings were important, since the reliability of the medium-speed engine would depend on their success. Over the past ten years manufacturers of flexible or elastic couplings had had a fairly easy time because propulsion systems had been fairly conventional. With the introduction of the much more sophisticated multi-engine systems, they had to tackle the problems which faced the shipowner and Diesel engine designers.

The authors had made a misleading statement in suggesting, on page 99, that the coupling and quillshaft had been used to obtain the required flexibility between engines and gearing. That put the flexible coupling manufacturers in a rather peculiar light. Elastic couplings did not depend on incorporation of the quillshaft in the system. Perhaps the authors would comment on that.

While he realized that the paper had essentially been projected towards propulsion systems for ocean-going vessels, would the authors comment on aspects of propulsion for vessels on coasting duties, where manœuvring requirements became of prime importance in the transmission specification?

Although the authors had commented briefly on controllable-pitch propellers, he would appreciate their comments towards an economic comparison of such a system with the more conventional systems already described in detail by them.

While being aware of certain of the design advantages of controllable-pitch propeller systems, it would be of assistance to appreciate the balance between economic and technical merit when considering power transmission requirements satisfied by prime movers specifically mentioned in the paper.

It was important that transmission coupling manufacturers were able to understand fully the trend in modes of marine propulsion and associated implications, since coupling development depended essentially on the accuracy of such forecasts.

Dr. H. Watson, B.Sc. said that from his personal dealings, having provided some of the medium-speed engine information reproduced in the paper, he knew that the authors had taken nothing on trust and any figure associated with the maintenance and operation of medium-speed engines had only been included after discussion with users of both medium and slow-speed engines. He felt that the overall policy of the authors had been deliberately to give any benefit of doubt to the established slow-speed engine and this bias away from the medium-speed engine should eliminate any doubt as to the validity of the main conclusions, namely that the medium-speed engine showed an overall cost saving.

This bias, and a tendency towards conservatism, was further illustrated by the comparison made of two engine-room installations, one utilizing main machinery 52 ft long and the other utilizing machinery 26 ft long. The engine-room length chosen to accommodate the shorter unit was only 10 ft shorter than that necessary for the longer unit.

He also considered that savings in ship construction credited to the medium-speed engine were conservative and that, in future, more imaginative design could give additional credit. Similarly, it would seem logical that the cost of ancillary pumps debited to the medium-speed engine could be reduced considerably, since, under certain circumstances, standby units could be eliminated as a standby engine virtually existed and, in other cases, a single standby unit could be made to serve both engines.

The comparison of lubricating oil costs between slow and medium-speed installations was somewhat unfortunate and had certainly given the protagonists of the slow-speed engine a talking point. Of the slow-speed engines available, the unit and design chosen, namely the Burmeister and Wain engine, had one of the lowest consumptions on record. The authors' comparison was absolutely fair in this particular case, but, had a slow-speed engine of different design been chosen, for instance the Doxford engine, the cost differential associated with lubricating oil could have been much less.

The value of lubricating-oil consumption used in the paper for the medium-speed engine was at present achieved. Intensive development was, and would continue to be, applied to reduce this value and it was probable that over the next five years a reduction by as much as 30 per cent would be achieved. It was unlikely that a comparable reduction would be achieved by the slow-speed engine.

In the conclusion to the paper, the authors stated that it was vital to maintain the lowest capital cost of medium-speed machinery. He felt it was true to say that inflation of some degree would apply for some time to come and, since the medium-speed engine had development recovery as an appreciable portion of total cost, a charge which was fixed and which was of much bigger proportion than in the case of the slow-speed engine, future inflation was likely to widen the gap in first cost between the slow and medium-speed engine, irrespective of future development towards higher ratings.

Finally, he would like to express disappointment to the authors for what he considered to be an important omission from the comparison of operating costs, namely the cost of spare parts. He was convinced that the medium-speed engine would, by having a lower spares replacement cost, eliminate any debit due to the higher maintenance man-hours. During the compiling of the paper, medium-speed engine builders were prepared to give spare parts prices. No such values were available from the slow-speed engine manufacturers and he could only conclude here that there was something to hide.

He would welcome the authors' views on the foregoing and in particular on the last point.

Many points had been raised by other contributors to the discussion on which he would like to comment but he did not feel it was in his place to do so. He could not, however, leave without saying that over the last ten years Ruston and Hornsby had provided more engine installations spending up to 30 days away from port on each voyage than had Doxfords in the same period.

MR. F. A. SNEAD, referring to clutches, said that it was difficult to say that clutch manufacturers could do exactly what was asked of them when, very often, they were not even told what it was they had to do.

In 1942, a reduction gearbox was required for a tank landing craft. Mr. Snead's company manufactured about 4500. They were relatively light in horsepower and loading requirements. The reversing clutches used in those days also acted as couplers. It was obvious that, with the larger engines, a different type of clutch was required and, consequently, it was necessary to rely upon a good, flexible coupling to handle the problems thrust upon them by engine vibrations. However, it had to be assumed that it could be taken care of by the coupling or quill arrangement. Reference had been made to the use of the quill and the coupling. In many instances that was brought about because of the quill-mounted type of clutch arrangement, in other words it was mounted on the back end of the gearbox, which allowed for ease of maintenance. With the air type clutch it was mounted outboard of the gearbox and there was not the problem of removing portions of the gearbox for maintenance.

Mr. Snead's company was able to supply quill-mounted clutches, for 400 rev/min, up to 20 000 hp. The capacity of these to handle manœuvring was a point about which very few facts were known—as was the crash-reverse requirement. Facts were not available. These clutches had accommodated all crash-reverse requirements in the small-powered vessels, including also multi-engined vessels of up to 730 ft. The characteristics of the larger vessels, one was told, were different from the smaller ones in that, when the engine was disconnected from the propeller shaft on the smaller vessels, the ship lost way swiftly, so when the propeller shaft was re-engaged with the engine in the reverse direction, the loading would be relatively small, in comparison to a vessel that had not lost way and was trailing its propeller. His company had an installation for 500 rev/min at full speed; the engine was brought back to idling speed at 200 rev/min. On a small vessel, if disconnexion was made at that point and the engine put into reverse, or a reversing clutch picked up, the specific load would be based upon idling speed. However, where the propeller was drawn into a higher than idling speed, there was a slightly heavier energy load placed on the clutch. Allowance had been made for this. It was a period of time that was involved, not only to let the engine speed fall to zero and then climb back up to reverse speed, but also to protect the engine from overload. The engine must not be stalled and, therefore, the amount of torque that the clutch would transmit had to be limited. This could be done by limiting the air pressure.

The normal engagement procedure was to pre-fill the clutch. This would mean, say, that 35 lb of air pressure was admitted at engine idling speed, which would protect the engine from stalling and take care of all the thrust loads through the gearing to prevent hammering, etc. One would then increase speed through the governor, which automatically increased the air pressure to the clutch, ensuring that there were two and a half times the engine torque capacity in the clutch.

The crash reversal, although it required that operation be as swift as possible, had to follow the same path, otherwise the engines would be blown and heavy loading placed on the box and other parts of the transmission. There was a strong fear of using big clutches on the engines. Mr. Snead's company had increased factors. They had so many of the clutch installations of all types that they were fairly confident that they could handle any job. This still had to be proved to the marine world and tests were being carried out in Japan. The capacity of this particular unit was about 5000 hp at about 500 rev/min. The figures for these tests would be some time reaching this country. There were other installations which were similar to those for marine applications. The speaker said that he found it very difficult to talk to marine people because they lived in a world on their own, isolated from all others. Mr. Snead sold clutches; he said that it was a nice job and he was well paid. He called on all types of industry. Many of his company's clutches, which they were proposing to use for marine operations, had been used for many years in industrial applications which were even more demanding on the clutches than marine applications.

To accommodate manœuvring and engagement in torque at relatively low speeds, and also the long periods of time when clutches were left fully engaged or disengaged was, to his company, a simple task, because they still had to use those same clutches for industrial applications, where cycling was perhaps five or ten a minute, and were expected to accommodate that type of installation with the same units. Naturally, they used different selection procedures but, basically, the units were the same.

A difficult point for the speaker to make concerned the types used. He said that he sold a particular type of clutch—a drum type. Everyone was free to have his own choice, of course, but he felt that this type had a particular advantage over the disc type clutch, i.e. the drum clutch, when it disengaged, was completely disengaged. It would give a completely disengaged unit when one engine was being used. That was the prime factor of its use. Furthermore, hammering of the keys, etc., due to vibrations in the shafting, affected discs.

Mr. Snead said that information from testing would always be made available to the Institute, in order that its members could be kept fully informed of developments.

## Correspondence

MR. R. YATES, O.B.E. (Member) wrote that in Fig. 1, page 92, the engine centres on the medium-speed twin installation would appear to make the engines almost inaccessible on the inboard side. This must make any overhauls more difficult, particularly work in the crankcase.

In Fig. 4, page 94, the savings to be gained by burning 3500 seconds fuel, compared with a 600 seconds fuel, seemed to be hardly worth while, even if there was exhaust-gas heat available. This would be especially so if the use of heavier grades meant added expense, such as water-cooled exhaust valves or other complications.

On page 94 there was no mention of the difference in cost for the initial charge of lubricating oil applicable to the two types of engines. The slow-speed engine would require 6000-10 000 gallons in circulation in addition to a further 10 000-15 000 gallons for daily use and possible recharge, compared with 700 gallons plus 1000 gallons for the medium-speed engine—a ratio of 10 : 1. After a period operating medium-speed engines on intermediate fuels he would suggest that a more comparative figure for lubricating oil consumption might be in the ratio of 40 to 85 gallons per day in favour of the slow-speed engine.

With regard to operation on heavy fuel (page 93), he wrote that exhaust valve burning appeared to be high on the list. There must surely be some simple means of controlling exhaust-valve life without resort to the complicated gear often associated with water-cooled exhaust valves.

Referring to page 99, under the heading of "Machinery Arrangements", he suggested that the medium-speed Diesel might, with advantage, be adapted for forward power take-off for the supply of electrical power. In this way heavy fuel only would be used when the vessel was at sea.

He would agree with the authors' conclusions that the medium-speed trunk-piston engine would burn intermediate fuels without any operational disadvantages in terms of fuel consumption or wear. The economic advantage of the geared Diesels might be largely influenced by the ability of the engine builder to keep down the initial cost of the engine.

MR. A. E. FOTHERGILL, B.Sc. (Member) wrote that the paper indicated many of the numerous factors which should be taken into account and the lines of approach, when considering the relative merits, in ships, of medium-speed geared Diesel installations and those with slow-speed direct-drive machinery, but it would be unwise to accept the conclusions arrived at as generally applicable to all prospective motor ships. In the two examples chosen for study, the authors had worked their own specification and different results could be obtained when conforming to specifications put out by owners for tendering.

The savings on fuel costs deduced for the medium-speed type were mainly due to assuming much lower speeds for the shafting than were usual with direct-drive and, in the two ships concerned, this resulted in the propeller-blade tip immersion factors being below the values normally considered desirable. There were many ships with direct-drive in service, in which the propeller diameter was at or near optimum and the immersion normal. In these there would have been little advantage in designing for lower speeds, with any increase in efficiency partly offset by the effects of insufficient immersion and proportionately smaller aperture clearances. In this connexion it was significant that, referring to Appendices IID and IIE, the average shaft speeds for the thirty-one geared Diesel-engined ships was 134 rev/min and for the thirty-three direct-drive ships 136 rev/min—a difference of only two rev/min.

As regards other items affecting fuel costs, the consumption rate quoted for slow-speed main engines could be bettered by at least two per cent and, as large engine-driven pumps were not favoured in marine practice, the fuel consumed by the generator engines should be the same for both kinds of drive.

From the foregoing remarks it seemed that the possibility of saving on fuel costs, by adopting medium-speed geared Diesel drive was doubtful. The space taken up by the present-day complicated auxiliary machinery was now the major factor which determined the size of engine room required and gains claimed due to reduction of engine dimensions were inclined to be exaggerated, only fully worked-out machinery arrangement plans could prove that.

Reliability of the main engines was of paramount importance and could take precedence over economic and other considerations, particularly so with regard to immunity from breakdown at sea. Appendices II, IIA, IIB and IIC gave some guidance on this, but were based on a limited number of ships. A scrutiny of the daily defects lists published over the last nine and a half years showed that, for vessels registered at Lloyd's, main engine defects in motor ships had been reported on about 3600 occasions for direct-drive slow-speed machinery, compared with about 1100 for medium-speed engines, a ratio of 3·3. It was estimated that there were approximately twice as many medium-speed engined ships as there were slow-speed, so that the trouble risk with the slow-speed was apparently 6·6 times that with the medium-speed, but it had to be noted that a large proportion of the medium-speed installations were in the smaller size ships which traded in more sheltered waters on short runs and only a few operated on heavy fuel. The reliability of the very highly rated versions envisaged in the paper remained to be proved.

The speeds given for the two ships studied required explanation, as, in the early part of the paper, 16 and 20 knots were quoted, whereas in a later section 15 and 19¼ knots were mentioned. Also, it would be of interest to know whether the e.h.p. shown in the section on propeller speed were for smooth water or included an allowance for average weather at sea. The estimated fuel costs tabled in the section on fuel consumption were based on full service power being developed all the time during the days at sea assumed, irrespective of conditions of draught and weather; this seemed rather unlikely.

MR. B. GRZYBOWSKI, M.Sc. (Member) commented, in a written contribution, that the paper undoubtedly gave a lot of information, but it seemed that it could be supplemented by a few more facts in order to make it more valuable.

Firstly, besides the maintenance requirements of the projected Ruston and Hornsby 16-cylinder AO engine which were the estimated figures only, it would be useful to tabulate the figures for the Mirrlees National K Major engine taken from shipowners' experience. It was understood that it was not the same type of engine, but the comparison would be interesting.

It was regrettable that the authors had not given their estimate for the projected Mirrlees National OP engine.

Having such a table comprising one slow-speed engine and three medium-speed engines side by side (the figures in one case taken from shipowners' experience) the picture would be clearer and perhaps more convincing.

Secondly, it would have been very useful if the authors had given even the approximate cost of spares required for low and medium-speed engines during the lifetime of the ships. Everyone concerned with maintenance of ships knew that spares represented a considerable operating expense, especially when the ship was more than twelve years old.

He had some doubts regarding the figures given for maintenance requirements and in many cases considered them much too low for medium-speed engines.

For example, according to the authors it was enough to spend 152 hours only to grind exhaust valves and examine springs. If one considered that there were 128 valves which were to be disconnected, withdrawn, brought to the ship's workshop, taken to pieces, ground, checked for tightness, ground again, reassembled and refitted in the cylinder head, it seemed doubtful whether this could be done in 72 minutes for each valve.

Only 40 hours were allowed for fitting 32 reconditioned liners. It was very difficult to believe in such efficiency on the part of the ship or shipyard crew.

To withdraw the pistons (remove the crowns) and clean and check the rings was estimated to take 288 hours only. There were 32 pistons, 192 piston rings and three cylinder heads with valves which were to be disconnected. The cylinder heads were to be removed, then again everything was to be refitted. It seemed that it could not be done under normal conditions. It meant that two men would do this work in 4½ hours—a very optimistic estimate.

There were more doubtful figures in this table, but, in order to contradict them firmly, one should at least see this engine.

It was also very doubtful whether any shipowner having installed two 16-cylinder engines would use the same number of engine crew as in an engine room with seven-cyclinder engines.

In his opinion the great majority would increase the crew by at least three men.

All that he had said so far did not mean that he was in opposition to medium-speed engines—not at all.

If the output per cylinder and even rev/min could be increased and the number of cylinders could be reduced this would mean an ideal solution for shipowners for the maintenance costs of an engine could be described as:

$$M.C. = K + f(N)$$

where $K$ = constant, nearly the same for certain types of engine.
$N$ = number of cylinders.

Mr. T. W. Major (Member) wrote that the section on maintenance showed the geared Diesel in an unfavourable light, but there were possibilities for considerable savings in both maintenance and survey costs which would more than make up for this disadvantage.

For example, he had seen a piston of a Ruston and Hornsby AO engine, complete with connecting-rod and bottom end, put out on the platform inside an hour, using men who were not familiar with the job.

If reconditioned parts were used, a piston could be refitted in a further hour. The units removed could be overhauled without haste during the next passage.

If with the piston removed, it was necessary to change a cylinder liner then this could be done in well under two hours.

A turbocharger weighed only about twelve hundredweight and was reasonable in cost. By carrying and fitting a spare, the used unit could be cleaned, overhauled and tested with compressed air on the next passage, ready for the next exchange.

The very short overhaul times made it possible to work on the geared Diesel while anchored, waiting for a tide, or during a half-day stop in port, when it would be quite impossible to do similar work on the larger engine.

Main engine maintenance and survey work was very costly when it had to be carried out by shore labour, but, with the geared Diesel, there did now seem to be a real chance of making substantial savings in working costs.

Another matter which was assuming considerable importance as vessels were built with powerful engines placed aft or three-quarters aft, was the reliable and accurate functions of the very large amount of shipboard instrumentation. With the exception of the Doxford engine, all the large-bore, slow-running engines had substantial out-of-balance forces, the exact magnitude depending on the cylinder combinations.

These out-of-balance forces could have a disastrous effect on instruments and superintendents' carefully worked out schemes for reduced manning based on monitoring systems could become useless, in addition to the cost of servicing or renewing. In geared Diesels these out-of-balance forces were usually negligible.

Mr. E. A. Bridle, B.Sc. (Associate Member) commented that the maintenance costs estimated by the authors seemed to be very low, amounting, in the case of the direct-drive engine, to less than £1500 per annum for the bulk carrier and less than £800 per annum for the cargo liner.

An estimate of operating costs for the 84-VT2BF-180 type engine had been published*. This gave the annual maintenance cost as $1.53 per rated brake horsepower for 349 days operation. Assuming the maintenance costs to be proportional to the number of days operation, the authors' figures, on this basis, would have to be multiplied by a factor of five.

If the relative costs of maintaining the geared and direct-drive engines remained in the same ratio as that estimated by the authors, the totals in the tables on page 103 of the paper would be changed from +1985 and + 4834 to −1090 and +3204.

This result was still inconclusive, although somewhat less to the advantage of the geared Diesel machinery and one wondered how the authors felt able to recommend the expense of further studies in the doubtful hope of obtaining marginal improvements over established types of prime mover of proven performance and continually increasing reliability.

Mr. R. M. Dunshea (Member) wrote that when one was was in agreement, a contribution to the discussion on a paper presented difficulties. The authors were clearly pursuing the trend which the builders of large-bore engines had themselves been actively following over many years, namely, reducing the weight and space required for main propulsion machinery. They had convincingly shown that worthwhile savings could be achieved in the first cost of a ship propelled by medium-speed machinery.

As was usual, conservative opposition to medium-speed machinery by superintendents was to be expected. A parallel case was the opposition to high-speed generating plant, which had now shown itself to be completely reliable. Mr. Dunshea considered that so long as the number of cylinders did not exceed 36 ( this number could be adequately handled for fuel-valve and exhaust-valve routine replacement by a small engine-room staff in the short periods that the ship would be in port) the case for medium-speed machinery would be hard to refute logically.

With slow-running engines now up to 1000 mm bore and highly rated, thermal stresses arising from heavy sections were bound to cause problems, to which solutions would be doubtful. When large poppet valves were employed, cooling could not be as efficient as with the small valves of medium-speed engines.

Dimensional accuracy was more readily achieved with smaller components, thus components would fit without being made to fit, as was frequently required in large-bore engines. It was also apparent that alignment problems which could be so time-consuming after an engine had been in service for some years, would be more readily solved in medium-speed engines.

In this connexion, consideration should be given to one type of opposed-piston engine now in service, but which was however infrequently built, where on the 750-mm bore type the weight of the exhaust piston and associated running gear totalled 27·7 tons.

The problem of lifting components and checking alignment which had to be done on occasions was formidable and it was considered that even six pistons with connecting-rods and bearings on a medium-speed engine would require far less effort and time. Engineers with any experience of large-bore Diesel engines must agree that such alignment investigations as described had to be carried out from time to time, often in foreign ports under difficult climatic conditions and with very little shore assistance.

Since one of the medium-speed engines under consideration was an opposed-piston engine, of which few details were known, a comment concerning experience with tripartite liners would not be out of place.

After some years in service on the engine to which reference was previously made, great difficulty was experienced in maintaining a gas-tight joint between the three components. The original liners which were produced for the 620-mm version of the engine suffered severely from fractures.

In discussing these troubles with a well-known Continental designer of large-bore engines, the latter stated that during the war he had carried out extensive investigations into the design of an opposed-piston engine and found that it appeared almost impossible to produce a satisfactory cylinder liner. This would appear to have been confirmed by Mr. Dunshea's experience and should be borne in mind by future designers.

Efficient combustion would be more difficult to achieve in medium-speed engines. In this connexion, in 1966, he had been able to inspect a Pielstick 12PC2V engine at St. Nazaire after a test run of 8000 hours on fuel up to 3500 seconds viscosity. The general condition of components was good and he had asked why the turbochargers were not opened up. He had been told that their performance at the end of the test was no different to that at the beginning and that there had been no good reason why they should be dismantled.

*Sato S., 1964. "The Design of 100 000 ton d.w. Diesel-engined Tankers" *The Motor Ship*, B. and W.—Hitachi Supplement, Vol. 45. No. 533, p. 39.

The impression gained after inspection of the engine was that periods between overhauls of components would be longer than indicated for medium-speed engines by the authors in their paper and that they should exceed those for larger-bore engines. Against the medium-speed engine, the lubricating oil consumption was high. To make the engine really attractive more research was necessary to reduce this to not more than half the present average consumption rate.

All the medium-speed engines considered had thin-wall bearings with a flash of white metal. To avoid scoring, the standards of cleanliness of staff carrying out overhauls must be much higher than presently obtained in ship repairing yards.

Since crankshaft alignment could not be altered by varying the thickness of bearing crowns, particular attention would have to be paid to the stiffeners of engine seating and chocking.

Given a progressive attitude now, after many years of conservatism apparent in the shipping industry, the future of the medium-speed propulsion installation appeared to be assured. The geared-Diesel installation was by no means new. During the last war, many German surface raiders were propelled by such installations and some of us knew how well they operated over long periods.

Mr. D. Honour (Associate Member), in a written contribution, remarked that the paper had been presented at a time when several major shipowning companies were examining the possibilities of using geared medium-speed Diesel engines in place of the conventional direct-drive type. Design studies, of the type carried out by the authors and their organization, were essential when trying to arrive at the correct decision, but each shipowner must tailor the calculations to suit his own vessels and the trading pattern involved.

For instance, the lubricating oil storage seemed rather too low at 40 days' supply. Six months' storage was not unusual and some owners even carried a year's supply. It would be interesting to learn what effect six months' supply would have on the weight and space conclusions drawn.

Could the authors elaborate on the noise levels to be expected with medium-speed engines? It was noted that a sound-insulated control station was recommended, but the authors thoughts on acoustic protection for personnel engaged on maintenance would be appreciated. This could be a serious problem, while working on one engine at sea, and it was suggested that some form of portable acoustic screen between the two engines might be justified. The employment of Diesel generators only added to this noise problem, not only because of the noise they generated, but also for the number of occasions when personnel would be working on them. Many motor ship operators had already come to the conclusion that a turbo-alternator, operating on low-grade steam from a waste-heat recovery system, was much more suitable and fully justified on economic and technical grounds.

It was surprising that only a simple exhaust-gas boiler was considered in this study, as the energy potential in the exhaust gases must surely be adequate to meet the electric load, at least in the cargo liner. The electrical load for the bulk carrier seemed very high at 810 kW, when compared to some oil tankers developing 14 000-18 000 bhp and which could handle the full sea load at all times on a 600 kW turbo-alternator. These vessels had all-electric auxiliaries and full air-conditioning.

The implementation of the routine maintenance schedule must be given very careful consideration. With the bulk carrier being at sea for 300 days per annum, shore labour would be required every eight or nine months to handle the amount of work due at 5000-hour intervals. This might not be popular with some owners, particularly those trading in areas where first-class labour was not readily available. It was to be hoped that future experience and development would raise the period between piston overhauls to something nearer the 8000 hours of the slow-speed engines.

The authors had pointed out the desirability of fitting clutches or hydraulic couplings between each engine and the gearbox. Most owners would require such devices and it seemed a little unfair that the inefficiency of these had been neglected in the calculations on power requirements and fuel costs. On the other side of the scale, 0·378 lb/bhp-h seemed rather pessimistic for the specific fuel rate of the slow-speed engine. A more realistic figure would have been something like 0·365 lb/bhp-h.

Commander K. I. Short, O.B.E., D.S.C., R.N. (Member) said that he considered that the paper provided a most useful service by bringing into focus a series of excellent papers and articles on the subject of medium-speed Diesel engines.

The authors were preaching to the converted as far as he was concerned.

His faith in the ability of the medium-speed Diesel engine to show up to advantage in certain propulsion applications was currently being supported by his company which was constructing two 12 000-dwt cargo ships, each propelled by two medium-speed Diesel engines operating on heavy fuel, geared to a single shaft and controllable-pitch propeller through elastic couplings and plate clutches.

He mentioned this new building to record, at the outset, the practical nature of the partisanship behind his contribution.

Medium-speed Diesel engines were not necessarily the panacea for all shipowners. Whether or not they were suitable depended upon the particular trade, ports visited, pattern of operations and so forth.

In the case of his company, cargo ship dimensions were limited as follows:

1) beam and draught by the Manchester Ship Canal;
2) length by the need to turn in the Kiddepore Dock at Calcutta;
3) block coefficient as usual to obtain reasonable compromise between power and cargo deadweight.

The Hooghly River imposed a draught limitation on ships and as a result:

a) His company's were deadweight ships at low draught. So important was this that the shipbuilders were under "penalty" to provide a specified cargo deadweight at 21-ft draught. Any saving in machinery weight in this case within the limited ship dimensions over the equivalent large engine installation was obviously of special importance as it normally represented additional freight and not just potential additional freight.
b) The amount aft by which they could afford to place the machinery, without risking having to ballast at low draught and shut out cargo, was limited. The lighter the machinery, the less this problem became.

Thus, with these limitations, the weight saving of 280 tons offered by the medium-speed engine installation, together with the associated space saving, was very attractive to his company. The medium-speed engine, taking consideration of space, weight and price enabled the use of a twin-engine arrangement and this:

i) would enable maintenance and continuous surveys to be progressed systematically on one main engine whilst still having the other available to move the ship;
ii) would enable ship's personnel to take advantage of delays in ports to maintain one main engine whilst always having one of them available to move the ship.

In planning this installation his company had, from the very outset, discussed and reached agreement on, with their marine superintendents, the importance of accepting, without reservation, that the ship was "seaworthy" on one engine. It was essential to take the deck department along with one if the maximum benefit was to be reaped from this type of engine arrangement. They had heard of some earlier attempts to capitalize on twin-geared Diesel installations, in the way he had described, which had foundered as a result of over-cautious operators insisting that two engines were necessary for seaworthiness.

He would now like to touch upon the use of twin-engined geared medium-speed Diesel engines for smaller tankers, particularly for independent operators. He did not infer that the same advantages were not also available for oil companies, but the latter had more flexibility in selection of maintenance periods and, on the average, had more time available for repairs and maintenance than did the independents. As a result, potential savings in maintenance time were particularly attractive to the independent operators.

Big engines needed thorough unit maintenance at intervals of approximately 12 months. As this could not be done during loading and unloading, independents had to aim to do this work during the low freight period, which in Europe corresponded with the summer months. To be worth while any improvement in big engines must increase maintenance periods by at least 12 months and, whilst it had been reported that some tankers had operated successfully for two years with big engines without stripping units at least once, such reports were not the general rule.

With a twin-engine geared medium-speed installation:

1) it would be possible to maintain/prepare for survey one main engine whilst proceeding on the other during the ballast period with little loss of speed;
2) it would be possible, he believed, to plan confidently for two-yearly drydocking periods and save the time currently taken out for drydocking every other year;
3) the saving in weight and space would be useful—the latter particularly for low specific gravity products;
4) liability would be limited in event of engine breakdown.

Crew costs had been rising so rapidly that, whether anyone liked it or not, they were being forced into the unmanned engine room, or the reduced-manning alarmed engine room, concept. In his view the techniques were available now to make these steps—the nuclear engineers would not be in business if they were not. With such systems emergency watchkeepers would still be needed and normally would be available for day work maintenance. The geared twin-engine installation would enable such maintenance effort to be deployed on the main engines at sea.

For large single-shaft tankers and bulk carriers he visualized that eventually the Phase III medium-speed Diesel engines might take over from turbines—by Phase III he referred to engines in the power range of the Fairbanks Morse and Mirrlees opposed-piston engines. He would prefer to limit the number of engines on a shaft to two, but if sufficient power was not available from two engines, he would not discount the use of additional engines. As could be seen, however, the majority of advantages had been acquired with two engines and he questioned whether there were practical advantages to be gained from adding further engines.

At least one company had recently reported large losses as a result of breakdowns of the single main engine in several of their ships. The larger the ship, the larger such losses and the greater the attraction of any system such as the multi-engined installation which promised to limit them.

He thought that the authors had dismissed the controllable-pitch propeller too lightly on the score of cost. His company's experience had been that the controllable-pitch propeller installation was not, in practice, as expensive as it appeared at first sight.

The following actual tender differences for two 12 000-ton cargo ship designs which differed only in the respects listed might prove of interest:

| *Design A* | *Design B* |
|---|---|
| Controllable-pitch propeller | Fixed propeller |
| 3 × 320 kW Diesel generators | 4 × 440 kW Diesel generators |
| Non-reversing engine | Reversing engine |
| 500 kW shaft generator | — |
| Reduced size air reservoir | Large air reservoir |
| Two spare propeller blades | Half cost spare manganese bronze propeller |
| Half cost of spare tailshaft | Half cost of spare tailshaft |
| Bridge control of propeller pitch | Bridge control of engines |

| Tender | Design | | Notes |
|---|---|---|---|
| | A (c.p.) | B (reversing) | |
| Foreign | 0 | + £1170 | a) Subsidiary manufactures propellers.<br>b) U.K. auxiliary Diesel engines. |
| U K. I | 0 | – £8500 | a) Foreign licensed engine.<br>b) Foreign licensed propeller. |
| U.K. II | 0 | – £4500 | a) U.K. design engine.<br>b) Foreign licensed propeller. |

All of these costs were within the estimating margin and the reduced maintenance and saving of Diesel fuel by the use of the shaft generator would soon pay off the highest of the differentials quoted.

Incidentally, he was doubtful whether one should claim any advantage for controllable-pitch propellers from cutting down the number of occasions upon which cold air was admitted to cylinders when manœuvring. Certain vee-engines only had starting air on one bank and he understood that no difference had been found in unit wear rates as between banks.

Taking a foreign-licensed, U.K.-built, large reversing engine without a controllable-pitch propeller as the basis, U.K. tender differential prices as follows were obtained for a 12 000-ton cargo ship in respect of twin medium-speed geared reversing installations:

+ £2400 for a prototype British medium-speed Diesel engine;
+ £24 000 for a reasonably well proven foreign licensed, British-built medium-speed Diesel engine.

These costs took into account weight savings and changes in ship dimensions to obtain optimum ship design. The gap between the two installations could, it was believed, have been narrowed as the shipbuilder quoting for the engine room manufactured the large engine and was over-cautious in costing the rival arrangement.

These figures did, he believed, allowing for estimating margins, confirm the authors' contention in respect of cargo ships that, from the point of view of cost, there shouldn't be much to choose even at this stage, between a large and medium-speed engined ship for the same duty.

It was, as the authors contended, essential for wide deep-sea usage of medium-speed engines in sizable ships that they operate satisfactorily on heavy fuel. He only wished that he was as confident as the authors and engine builders that the problems associated with heavy fuel burning had been solved. He was not in the engine design business, but as far as he could see the exhaust valves were still very much the Achilles' heel of medium-speed engines, in fact he gathered that even the large engines treated this area with respect. There was still comparatively little service experience at sea available on different heavy fuels in medium-speed engines. Different lines of attack on the problem were confidently advocated by different engine builders and this led him to suggest that the fundamental problem was still not fully understood. Why was it that water-cooled valves showed to advantage in one engine and not in another? Why, similarly, did valve rotators "work" in one engine, but not in another? Some builders insisted on caged exhaust valves, but others claimed that this was unnecessary. Fairbanks Morse advocated water washing of the fuel and its chemical treatment before injection. Surely the differences of opinion between designers should be narrowed down.

Statistics would indicate the periodicity at which one should maintain exhaust valves of stationary Diesel engines operating on heavy fuel of the same analysis to prevent burning to a stage that reclamation was impossible or unduly expensive. Ships, however, operated on fuels of widely varying analysis. He considered that more information and guidance was needed as to how to detect incipient valve burning to ensure that nugatory work was not being carried out or the risk of gross burning of valves and seats being run. It was his company's intention in their new ships to chart record all exhaust valve temperatures and to alarm these readings. He hoped thereby that it would prove possible to detect a trend before the alarm operated and thence indicate a more sophisticated and satisfactory alarming system, but he did not feel too optimistic about their chances.

The cost of lubricating oil, was, as the authors pointed out, one of the disadvantages of the medium-speed installation, but was this cost differential as much as was often quoted? He had a feeling that large engines got through more lubricating oil on the average than was sometimes claimed. Again, could economies not be effected by separate cylinder lubricating oil injection with a cheaper crankcase oil examined regularly and spiked as necessary with additives? Here perhaps Fairbanks Morse thinking might be in advance of that prevailing in Europe.

He believed that the authors had dismissed the noise problem too lightly. There was a problem here, but he was confident that it could be solved. For medium-speed engines a sound-proofed machinery control room for watchkeeping was a *sine qua non,* whereas one could get away with watchkeeping in the open for large engines, provided one chose the location of the watchkeeping position intelligently. To enable maintenance to be carried out at sea, he thought it would be possible to rig padded curtains between the main engines and, despite some erudite predictions, enough attenuation to permit comfortable working was expected.

Amongst other advantages of the medium-speed Diesel engine, as opposed to large engines, which had not been mentioned were, in his view:

a) Ease of personnel familiarization: These engines were simply "blown up" auxiliary Diesel generator engines. Most junior engineers were familiar with motor car engines and he believed that they would find it easy to familiarize themselves with medium-speed Diesel engines if properly directed by the senior engineers.

b) Automatic and remote controls: The essential simplicity of these engines made it a fairly easy matter to automate them or arrange for remote control.

c) Assembly of engines: Medium-speed engines could and should be assembled under more hygienic conditions ashore. However, more could be done in this direction and he recommended that engine builders take a leaf out of the nuclear engineers' book and see what they could do to improve these conditions.

"Fire and flood" had always to him been the two major risks in unmanned operation of engine rooms. The probability of the former occurring was increased if oil leaks were prevalent. Far too many medium-speed engines were dirty in that fuel and lubricating oil leaks were accepted as inevitable. Engine builders, in conjunction with packing manufacturers, should do more to eliminate such leaks. The motor car manufacturers had been forced to do this by adverse criticism and as a result he understood special jointing materials had been developed.

Whilst congratulating the Ministry of Technology in supporting the medium-speed Diesel engine for ship propulsion, he could not help but wonder whether such help was not a bit too late.

Inevitably information on this subject had a tendency to be biased, as it was generally supplied and published by those having a vested interest in medium-speed or large engines. He thought that a debt of gratitude was owed to the editors of technical magazines, etc., for keeping people informed factually as to what companies were doing and how they were doing it. One could not value too highly these balancing contributions.

In conclusion, he believed that the engine builders had provided a reliable modern tool in the shape of the medium-speed Diesel engine. It was now for shipowners and shipyards to exploit this tool.

Mr. H. Sinclair wrote that on page 98 under the heading "Clutches and Couplings", the authors made it clear that direct-reversing engines geared to drive a fixed-pitch propeller were preferred for ocean-going merchant ships and hence the subject of the study, being lower in capital cost than unidirectional engines for use with either a reversing gearbox or controllable-pitch propeller.

As regards manœuvring, the authors advocated the practice of running one engine ahead and the other astern and engaging the appropriate clutch/coupling as required, the "crash astern" operation being carried out by direct reversal of both engines.

The hydraulic coupling was praised as the most popular medium for the connexion between the engine and pinion shaft, for the reasons given in the paper. On the other hand, the somewhat higher capital cost compared with friction clutches was noted and the more serious disadvantage of the continuous "slip loss" and the resulting additional cost of fuel, as shown on page 104 by the figures immediately beneath Fig. 11.

As to the selection of a good design of friction clutch, the authors mentioned five types of clutch offered for transmitting up to 8000 bhp at 450 rev/min. It might be mentioned that the Renk multi-plate friction clutch, compactly arranged on the pinion shaft in the Renk reduction gear, was widely used on the Continent. It should be noted that ratings up to 8500 hp per clutch were quoted in a paper read by Brauer, as recently as 17th November 1966, before the Schiffbautechnische Gesellschaft in Berlin.

As regards the choice of torsionally flexible coupling, the authors mentioned three types, with rubber elements. It would be interesting to know whether the Geislinger design, with laminated springs and oil damping action, compared favourably in overall terms—bearing in mind the fatigue aspects.

Reverting to the preferred method of manœuvring with one engine ahead and the other astern, an element of doubt was present about the durability of one or other type of friction clutch of 8000 hp, when so used for manœuvring.

In the light of this consideration, the authors might consider an alternative machinery arrangement, viz to combine the acknowledged superiority of the hydraulic coupling for manœuvring with the non-slip feature of an engaged friction clutch to transmit the power during normal ahead propulsion.

With this arrangement, the hydraulic coupling would be of reduced size, weight and cost and would work with about 7½ per cent slip, as compared with an hydraulic coupling of conventional size, with a slip of about 2½ per cent, permanently in the transmission line. A lock-out clutch would be provided in parallel with the smaller hydraulic coupling to eliminate the slip loss during all normal ahead propulsion, this clutch being disengaged immediately prior to manœuvring. One of the pair of engines would then be reversed and manœuvring effected by filling/emptying the required hydraulic coupling smoothly and without wear.

For the "crash astern" operation, the lock-out clutches would remain in engagement, both engines being reversed together as advocated in the paper.

As an alternative to a friction clutch for locking the hydraulic coupling, it would be a simple matter to use a clutch made by his company of the synchro self-shifting, multiple-tooth type, with a control lock which would be engaged for the normal working condition. Such a clutch, when unlocked, gave complete disengagement in both the driving and the overrunning sense, as was necessary for the clutch to be free when manœuvring by filling/emptying the appropriate hydraulic coupling.

The reduction gear would have the impeller and rotating casing of the hydraulic coupling conveniently mounted on the primary shaft, on the engine side of the pinion, and the runner would be mounted directly on the hollow pinion shaft. A quill drive would be provided with his company's clutch on the other end of the pinion shaft.

A torsionally resilient coupling, having the most suitable characteristics for the engine and geared to the propeller, would be arranged as a separate unit for direct connexion to the engine crankshaft.

In this application, the hydraulic coupling could be of the most rudimentary type, having only two rotating elements and conventional leak-off nozzles and being provided with a simple control valve on the fixed manifold, having two settings, thus:

1) to fill the working circuit from the pump or gravity tank; or
2) to empty the working circuit through the valve directly to the sump.

There would be no need to have quick-emptying valve gear of conventional type at the periphery of the coupling—since, when manœuvring with one engine ahead and the other astern, the vortex pressure condition within the working circuit of the respective hydraulic coupling caused rapid emptying via the control valve when it was opened to discharge to the sump.

As regards his company's clutch with its locking control, this would be similar in superficial appearance, although different in purpose and construction, to that in Fig. 34 of the paper by Weaving and Sampson.*

In his company's clutch, the two sets of pawls would be arranged with their noses pointing in opposite directions to meet the requirement that, when unlocked, the helically-splined clutch member had to shift axially into mesh and pass right through engagement with the internally toothed clutch ring from either the driving or the overrunning condition, while the manœuvring operation was being carried out with the hydraulic couplings. After completion of manœuvring and when again going ahead on both engines, the clutch locking control would be operated, and the fuel control of each engine temporarily closed and then re-opened, to engage and lock the respective clutch for normal ahead propulsion.

The size of hydraulic coupling to transmit 8000 hp at 450 rev/min with a slip of $7\frac{1}{2}$ per cent would be 70-in profile diameter, with an outside diameter of 6ft 6in and width of 24in, over the rotating elements.

In contrast, a conventional hydraulic coupling without lock-out clutch, to transmit 8000 hp with $2\frac{1}{2}$ per cent slip, would have a profile diameter of 86in with an outside diameter of about 8 ft and width of 30in.

The cost of the smaller hydraulic coupling would be little more than half that of the conventional size of coupling.

As regards the size of his company's clutch, which would be operating in parellel with the hydraulic coupling, the overall diameter would be about 25in and length 22in.

The foregoing machinery would be of somewhat lower capital cost than reduction gearing with conventional hydraulic couplings and the slip loss, with attendant extra fuel cost, would be eliminated during normal propulsion.

The lock-out clutch, whether friction-type or his company's type, would be virtually free from wear.

As regards availability, this presented no problem, it being noted that the hydraulic coupling and his company's clutch would be separate items mounted at each end of the pinion shaft. There was no limitation on the power-transmitting capacity for the vessels under consideration in the paper; clutches made by his company of 100 000 hp per clutch being in service on medium-speed as well as high-speed application on turbine-driven industrial plant.

*Weaving, P. D. V. and Sampson, W. H., 1963 "Progress and Development in Naval Propulsion Gears 1946—1962" *Trans. I. Mar.E.* Vol. 75, p. 73.

# Authors' Reply

The authors wished to express their thanks to those who took part in the discussion and those who corresponded, for the valuable contributions and critical interest shown in the paper. They particularly appreciated Mr. McAfee's opening remarks. With regard to provision of after sales service, sales of medium-speed engines generally had a large outlet in electrical power generation fields. Since these engines could not be returned home to pick up spares, overseas spares supply and service facilities had in many cases already been provided. The medium-speed engine had an advantage in that parts manufactured by the various licensees should be completely interchangeable.

Mr. McAfee's comments on the current position regarding approval, by Lloyd's Register of Shipping, of friction clutches for manœuvring purposes were appreciated and confirmed the need for further information and testing as suggested in the paper.

With regard to the question of reliability, the authors regretted that the way in which the data were made available to them prevented a meaningful assessment of the relative importance of the defects listed. The reason for using the ratio, number of failures to number of parts at risk, was to investigate the veracity of the statement that was commonly made that a 16-cylinder medium-speed engine would be likely to have twice as many failures as an eight-cylinder slow-speed engine. From the figures available it would appear that this was not true.

The authors did not pretend that the maintenance comparison in Appendix I was anything but an idealized one. There was insufficient statistical evidence to permit the inclusion of contingency allowances for all the things which could go wrong, nevertheless, as Appendix II showed, it was appreciated that both types of engine did suffer unscheduled failures.

The authors were heartened by Mr. Thompson's contribution, which indicated a determination by the manufacturers of the British designed and built Doxford engine to capture an increasing share of the marine Diesel engine market. Certainly if this engine could offer reliable operation at lower weight, space and cost as compared with other slow-speed engines, it would be in an excellent position to compete with them.

The Burmeister and Wain engine was selected in the paper to serve as a basis for comparison for two reasons: firstly, it was one of the most popular slow-speed engines being produced; secondly, the output of the seven-cylinder Burmeister and Wain engine matched the power requirements almost exactly. At this stage, it was of interest to record an apparent paradox. As Mr. Thompson said, the sizes of the two ships were selected to fit the power available from two Ruston 16 AO engines, but far from gaining an advantage from this, the method of analysis adopted had turned this to a disadvantage of the medium engine as shown in the following table.

| | Bulk carrier | Cargo liner |
|---|---|---|
| Power from two Ruston 16 AO engines:<br>Continuous<br>90 per cent rating | <br>16 000 bhp<br>14 400 bhp | <br>16 000 bhp<br>14 400 bhp |
| Power required for basis ship fitted with slow-speed engine | 14 400 bhp | 14 400 bhp |
| Power required for the reduced-displacement ship powered by medium-speed geared engines (see section headed "Propeller Speed" in paper) | 13 800 bhp | 14 040 bhp |
| Percentage rating at which medium-speed engine is working | 86 per cent | 88 per cent |

It was evident, therefore, that the medium-speed engine had been utilized at a rating which could be perhaps criticized as being a little on the low side.

It was true that engine-driven pumps could be used with slow-speed engines, but provision was not normally made to drive these pumps from the engines and it was questionable whether a simple and reliable drive could readily be obtained.

As Mr. Thompson had said, slow-speed engines could employ alkaline oils in their crankcases to obviate the need for diaphragms, but this would necessitate more frequent replacement of the crankcase oil and the oil charge, which was roughly four times that of the two medium-speed engines, would be rather expensive. The temperature inside the crankcase of the slow-speed engine was considerably lower than that of its medium-speed counterpart and it was likely that a more alkaline oil would be required if corrosion was to be avoided.

The question of noise was one on which it was difficult to generalize since the noise level and signature differed from engine to engine. In the opinion of the authors, the provision of insulated machinery control rooms was likely to become more common. Since some form of noise reduction treatment was desirable with both types of engine, it was unlikely that the noise question would influence the final selection of machinery.

To generalize, as one sometimes must do to reach a conclusion, implied that the examples chosen would apply in only a limited number of cases. The facts relating to the particular project in hand must be inserted in place of any assumed values or conditions, and this applied to costs and to operating conditions alike. The authors agreed with Mr. Thompson that if a ship spent a good part of its time in ballast then it would be necessary to take this into account in determining the optimum propeller diameter. It would be strange and peculiar if the optimum diameter always resulted in a propeller speed which exactly suited the slow-speed engine. The message which the authors wished to get across was that the geared engines offered virtually complete freedom of choice of propeller speed and therefore the optimum diameter and speed had to be determined on the basis applying to the ship in question.

For example, Fig. 7 indicated that the choice of propeller speed for the cargo liner was not critical since the curve of "total" cost was fairly flat, and hence a propeller running at 97 rev/min gave little "economic" advantage over one running at 114 rev/min. However, if, say, the capital cost should vary at a slightly steeper rate, or should a higher interest rate be more appropriate, it could well be that the most economic propeller speed was greater than 114 rev/min. The geared engines gave the necessary flexibility to choose the most appropriate rev/min.

The reason for the method of presentation of the statistics on failures had already been given in reply to Mr. McAfee. However reference to Appendix II of the paper would give anyone all the information available to the authors on this question, including the total number of failures. It was not true that it was impossible to detect incipient piston seizure on a ship powered by 32 cylinders. Such a condition would surely result in a temperature rise of the cooling oil from the piston concerned and, if required, the oil returns from each piston could be monitored and an alarm sounded should the temperature exceed a predetermined level.

The case for the medium-speed engines in vessels, such as ferries, had not been made in the paper, because this was not within the terms of reference covered by the study on which the paper was based and, since their use for this duty was well established, there was little need for further comment. It was worth mentioning, however, that a significant number of such ferries was operating on heavy grades of fuel generally of up to 1000 sec Redwood No. 1 viscosity.

The desirability of being able to burn fuels of higher viscosity than 3500 sec Redwood No. 1 was very doubtful with today's bunker prices. In the Middle East, high viscosity fuels were just not available. The problems of burning higher viscosity fuels were largely common to both slow-speed and medium-speed engines and chief among these problems must be proper heating and treatment to remove water and solid impurities.

Mr. Lowe had rightly commented on the future increases in rating to be expected from four-stroke cycle engines and of course by no means all engine manufacturers were adopting the two-stroke cycle. The authors had knowledge of highly rated four-stroke engines, in addition to those listed in Table I, being developed by English Electric, Burmeister and Wain, Fiat and Deutz. Further, the Sulzer Z engine which was listed in Table I as a two-stroke engine could also be built in four-stroke form as explained in the recent technical papers by Sulzer. There was certainly no intention, by the choice of engine used as a basis in the paper, to imply that the two-stroke cycle engine was necessary to meet the required conditions for the propulsion of large ocean-going ships. It would seem to be foolhardy at this stage to predict the eventual supremacy of either cycle.

A breakthrough in the realm of heat exchanger equipment was everybody's dream, but one which so far looked like remaining unfulfilled for the next few years, at least.

Coming from one who had had experience of direct-drive and geared Diesel engines, Mr. Ellison's remarks were invaluable.

It would be to the advantage of the country as a whole if medium-speed geared engines of British design became established as a reliable prime mover to compete with the Continental engines and it would be particularly attractive if this were not at the expense of Britain's only indigenous slow-speed engine.

The authors would not agree that there was little argument in favour of geared Diesel engines. It was agreed that the reliability of the projected engines had still to be established. Although it was stated that the study showed no overwhelming case in favour of medium-speed engines, this also implied that there was no overwhelming case against them. In other words the authors believed that the geared machinery would compete with the slow-speed engine and should be considered for any new construction in the same way that the different designs of slow-speed engines competed with one another. In this connexion the remarks made by Commander Short in his contribution were relevant.

The friction clutches themselves would not smooth out torque variations and they should therefore always be associated with some form of flexible coupling. Three types of coupling were mentioned on page 98 and to these could be added the Holset coupling. Designs of Holset coupling were available which incorporated various proprietary clutch designs.

One, no doubt unintentional, omission from Mr. Ellison's list of features most important to shipowners was low capital cost. The amortization of this was probably the largest single item in the annual running costs of the ship and of course it was in this field where the medium-speed geared engine was likely to show to greater advantage.

It was noted that the geared engines used by Mr. Ellison's company did not fall into the same category as those discussed in the paper. The authors agreed that long periods of idling were not desirable, although through-scavenged two-stroke and four-stroke engines were likely to be less sensitive than the loop-scavenged engines of Mr. Ellison's experience. It was difficult to prove that failures or accelerated wear rates could be attributed to the frequent admission of starting air. However, manœuvring by direct reversing of the engines consumed a great deal of air, and receivers and compressors must be sized accordingly.

In reply to Mr. Butler, the authors said that, as stated in the paper, the lubricating oil consumption of trunk-piston engines was approximately three times that of a slow-speed crosshead engine. The fact that the consumption was high implied that the engine sump had to be regularly topped up and this process maintained the oil in an acceptable condition for a very much longer period than would otherwise be the case. Oil changes were not therefore required too frequently and, in view of the relatively small capacity of the engine sump, the oil quantity involved when a change was required was not great.

The question of engine power and type of direct-drive engine was dealt with in the reply to Mr. Thompson.

The question of electrical generation, waste-heat boilers and engine-driven pumps had been raised by Mr. Butler as well as by various other contributors. It was evident that the paper was not sufficiently explicit on this point, and the authors wished to take this opportunity to clarify the situation.

The type of electrical generating plant to be provided depended intimately on the details of ship operating routines and procedures. Theoretically each of the undernoted forms could be applied to both direct-drive and to geared installations:

Diesel generators;
waste-heat boilers supplying steam turbo-generators;
generators driven from the engine or transmission.

In the event it was important to remember that:

i) it was easier to arrange a mechanically-driven generator from a geared medium-speed engine installation than from a direct-drive installation;
ii) except in very special circumstances, Diesel generators had, in any case, to be fitted for use in port.

As a generalization, schemes using waste-heat boilers or mechanically-driven generators would have higher first cost, but lower fuel and maintenance costs, than the scheme relying solely on Diesel generators.

If engine-driven pumps were fitted, although all shipowners would not necessarily do this and the installed generating capacity was reduced as a result, then, whether electric power was derived from Diesel generators, mechanically from the main engines, or from waste heat, there would certainly be a saving in capital cost of the plant to be taken into account and, depending on the scheme adopted, possibly also a saving in fuel costs.

On the question of fuel consumption, the figures for specific fuel consumptions quoted in the paper were guaranteed consumptions with fuel having a net calorific value of 17 000 Btu/lb. Using the curve in Fig. 19 of Mr. Jackson's paper to the Institute in November 1964* and making allowance for the different calorific value of fuel used, a specific fuel consumption of 0·364 lb/bhp-h was obtained. If the usual five per cent tolerance was now added, as was done with the Ruston and Hornsby figures, a figure of 0·382 lb/bhp-h was obtained, which was a little higher than that used in the paper. When the fact that the lubricating oil consumption of the Doxford engine would almost certainly be higher than that of the Burmeister and Wain engine was taken into account, it seemed unlikely that the running costs of the Doxford engine would be lower than those used in the paper.

*Jackson, P. 1964. "The Doxford J Type Opposed Piston Marine Oil Engine—Testing Experiences". *Trans.I.Mar.E.*, Vol. 77, p. 102.

The comments on electric power generation given in reply to Mr. Butler answered part of Mr. Maddocks' question.

A comparison of electric loads was given below. In this comparison, the important thing about the electric load was not so much the total load, but the difference in load between the medium-speed and the slow-speed engined installation. This difference was obtained as follows:

| | *Slow speed* kW | *Medium speed* kW |
|---|---|---|
| Engine L.O. pump | 130 | engine driven |
| Camshaft L.O. pump | 3 | — |
| Gearing L.O. pump | — | 35 |
| Engine L.O. centrifuges | 5 | 6 |
| Gearing L.O. centrifuge | — | 3 |
| F.W. circulating pump | 55 | 55 |
| S.W. circulating pump | 55 | 55 |
| Air compressors | $136\times0{\cdot}6^*=82$ | $50\times0{\cdot}6^*=30$ |
| Fuel pump | 7 | 7 |
| Fuel oil centrifuges | 15 | 15 |
| Diesel fuel centrifuge | 5 | 5 |
| Fuel valve cooling pump | 2 | — |
| | 360 | 211 |
| Difference | +149 kW | Basis |

*Diversity factor of 0·6.

In a study of this nature, which was not to result in the building of an actual ship, it was not possible to pursue in detail every avenue. Propeller-excited hull vibrations were one such avenue which had not been investigated in detail. The main exciting forces arising from the machinery occurred at the following speeds:

a) engine rev/min;
b) propeller rev/min;
c) propeller blade frequency.

Items a) and c) involved the greatest forces. The out-of-balance forces of the medium-speed engines were small compared with those for the slow-speed engine and occurred at about 450 rev/min as opposed to about 114 rev/min. Propeller-blade frequency would be not less than about 270 c/min. In the medium-speed engined installation the greatest forces occurred at frequencies above the frequencies of the out-of-balance forces of the slow-speed engine and hence it was anticipated that hull vibration would be no more troublesome than in a slow-speed engined ship, bearing in mind that two-node hull vibrations occurred between 50 and 100 c/min in large ships.

Part of the object of this study was to highlight those areas where development was necessary to help make the medium-speed engines competitive. The hydraulic coupling was well known to members, hence its brief mention. Compared with a friction clutch, the hydraulic coupling was more costly, especially when the increased capacity of the associated lubricating oil system components was taken into account and, as stated, involved a continuous loss of power. The use of a friction clutch did, therefore, improve the economic position of the medium-speed engines. The paper stressed, however, that, particularly in the higher powers, further research and development of friction clutches was required to justify their use at sea and in fact such research and development was being carried out.

The description of the latest development in connexion with hydraulic couplings, by Mr. Maddocks' company, was of interest and added to the value of the paper.

The authors would like to make it clear to Mr. Victory that the paper was based on a study carried out for the Ministry of Technology to assess how the economics of ocean-going merchant ships propelled by medium-speed geared Diesel engines would be likely to compare with slow-speed direct drive engines. It was not therefore appropriate in the context of this paper, as its title implied, to examine in detail other than ocean-going ships.

The work was carried out under the direction of a steering committee representing the Ministry of Technology, the Board of Trade, the British Ship Research Association, Lloyd's Register of Shipping, the National Physical Laboratory, shipowners, shipbuilders and engine builders. It was the committee, in conjunction with the authors, who chose the ships on which to base the study. The ships chosen were of a popular type—what better basis was there to take?

The task was not made easier by the fact that new engine designs which had not had any sea service were being compared with engines which had been used at sea for many years. However, the authors wished emphatically to refute the charge that the assumptions and data had been chosen so as to minimize the advantages of the slow-speed engine and to maximize those of the medium-speed engine.

The authors did not understand Mr. Victory's suggestion that, going to a great deal of trouble to collect information which indicated that medium-speed engines required approximately 40 per cent more maintenance hours than slow-speed engines, was brushing the subject aside. It would have been of great value to hear from Mr. Victory details of any other maintenance comparisons which he might have available.

The questions regarding engine power and type of slow-speed engine had been answered in the reply to Mr. Thompson.

The question of electric power was covered in the reply to Mr. Maddocks.

The lubricating oil stowage was estimated on the basis of a complete change of oil plus make-up. Since the slow-speed engine oil system contained a large volume of oil it was inevitable that the amount of reserve oil carried would also be large

The main engine fuel consumption was the product of two items—the specific fuel consumption and the power developed. The specific fuel consumption of the medium-speed engine was a little higher than that of the slow-speed engine. In the case of the ship propelled by the medium-speed engines, the power required was increased to allow for gearing losses, but was reduced compared with the ship propelled by slow-speed engines for two reasons:

i) the medium-speed engined ship had a smaller displacement;
ii) the medium-speed engined ship had a slower propeller speed and hence more efficient propeller;

the net result being a reduced fuel consumption.

Mr. Fletcher's remarks served to indicate that the authors did not fall over backwards to put up a case for the medium-speed engines, as some contributors seemed to imply. Having selected a power of 16 000 bhp as the installed power it was unfortunate for the Mirrlees OP 16-cylinder engine and the twin eight-cylinder engines, that the power available from them was a little more than the power required.

As Mr. Fletcher said, a service rating of 90 per cent of the maximum continuous rating of each engine was used. The maximum continuous rating of the OP 12 engine was 15 000 bhp and hence the authors would have been obliged to assume that the service rating was 90 per cent of this, i.e. 13 500 bhp. Although this was very little short of the 14 040 bhp and 13 800 bhp service power required for the medium-speed engined ships, even if the authors were to do the work again it was questionable whether it would have been agreed to assume a single 12-cylinder or twin six-cylinder OP engine, however tempting it might have been.

With regard to the OP six-cylinder engine, as Table I indicated, the authors were not aware that it was intended that there would be such an engine. This decision presumably must have been made after the work reported in the paper had been completed.

According to Lindgren* and Sinclair†, for a ship of 140 000 to 150 000 dwt, it was estimated that for a reduction in propeller speed from 114 to 68 rev/min, the propulsive efficiency was increased by approximately 14 per cent as com-

*Lindgren, H. 1965. "Tanker Ship Propulsion and Contra-rotating Propellers". *Shipping World and Shipbuilder,* 1st July, p. 48.

†Sinclair, L. 1966. "Large Diameter, Slow-running Propellers". *Shipbuilding and Shipping Record,* 2nd June, p. 728.

pared with an increase of about 5·5 per cent assumed in the paper for the 55 000-dwt bulk carrier.

The authors were indebted to Mr. Jackson for an enthralling and witty contribution. They felt sure that Mr. Jackson did not mean to imply that it was a waste of time to study the expected economics of new projected engines.

Before embarking on a development project of any new piece of machinery it would seem a good plan to compare its economic potential with existing machinery performing a similar function. The designers of the projected medium-speed engines had no doubt carried out such studies before and during the development of their engines. The study upon which the paper had been based was such a study carried out for a government department.

The authors made no excuse for the fact that the paper was based on untried engines. It would have been interesting to learn why Mr. Jackson was so surprised that the AO engine was being applied to marine propulsion after completion of shore trials: this did not seem to be very different from the procedure adopted with new designs of Doxford engines.

The Mitsubishi UET engine was slower, twice as heavy and at least twice as long as the Ruston AO engine. Presumably the cost of the engine was also greater than the cost of the AO engine. As the paper showed, costs, weights and sizes were important and on this basis the Mitsubishi engine was less competitive than the AO engine. This would not encourage its use in ocean-going ships. Nevertheless, engines from the Mitsubishi UET range had been fitted in no less than 25 ocean-going merchant ships in the last three years. It was significant that the ships, which I.H.I. of Japan were building as replacements for "Liberty Ships" and of which 34 had been ordered, were fitted with medium-speed engines.

On the question of reliability, it might be pertinent to mention that the type of medium-speed engine under consideration was frequently used to propel large distant-water trawlers. There could be little doubt that these ships, which spent much of their time in stormy Arctic waters, must confront their propulsion machinery with some of the most arduous duties to be experienced at sea. Since there was normally only the one propulsion engine, its reliability must be of a very high order.

The examples of the Deltic-engined ships quoted by Mr. Jackson were hardly relevant to the present discussion since these high-speed engines were of a special type necessitating a repair-by-replacement philosophy and were not designed to burn heavy fuel.

Mr. Jackson's recollection of the discussions a number of years ago, between his company and various ministries and government departments, was of historical interest, but the authors felt that it was not their responsibility to comment on the outcome of these discussions as reported by Mr. Jackson, nor on his views thereon.

The reasons for the difference in electric loads were given in the replies to Mr. Butler and Mr. Maddocks.

There was greater justification in providing engine-driven pumps with medium-speed engines for at least two reasons:

a) the engine speed was greater and the pumps could be direct driven, as in the case of the Ruston AO engine lubricating oil pump, resulting in a reliable drive;
b) in the event of a pump breaking down the associated engine could be stopped while the ship proceeded on the remaining engine.

On the question of engine-room length, Mr. Jackson had stated that the choice of engine would not affect the length of the engine room. This was not so—the length of the engine room was determined by everything in it, including main engines and auxiliaries. A reduction in size of any of these would contribute to a reduction in the length of the engine room. Since the projected medium-speed engines plus gearing took up less space than a slow-speed direct-drive engine it was inevitable that the engine room for the medium-speed engine would be smaller than that for the slow-speed engine.

Mr. Jackson was mistaken in his statement that medium-speed engines had been burning heavy fuels for only three years. Mirrlees put their first heavy fuel burning installation into service in 1951. It was true that this was a land installation, but they had had ships at sea burning heavy fuel since 1955.

The question of noise had been discussed in the reply to Mr. Honour.

Mr. Welbourn was correct in his statement that a quillshaft was not required in conjunction with a flexible coupling, for the twin AO installation considered, in order to obtain the required torsional flexibility between the engine and gearing. The primary reason for the quillshafts in this instance was to provide the drive to the friction clutches which were mounted at the aft end of the pinion shafts.

It was possible, however, that in certain installations, particularly with four-stroke cycle engines, that when operating on the high stiffness portion of the coupling characteristic, low order critical speeds might be excited to relatively large amplitudes. Since a quillshaft would have a similar torsional stiffness to that of the high stiffness of the coupling, under these conditions, its inclusion would reduce the natural frequency of the mode of vibration concerned and thus lower the critical speed.

The difference in manœuvring requirements for coastal and ocean-going ships appeared to be a matter of degree rather than any fundamental difference. The coastal vessel was smaller than its ocean-going counterpart, the masses of the transmission system to be stopped and the ship inertia to be overcome were less and hence, in this respect, the clutch duties were less onerous. The coastal vessel, however, would enter and leave harbour more frequently and so the clutch would be used much more often. The authors saw no reason why friction clutches should not prove successful in this case. The clutch was often external to the gearbox, or was arranged at its aft end where access was good, and thus replacement of clutch linings was a relatively simple operation, even if this was required more frequently on coasting vessels.

The choice of engine for the smaller ship was greater and it was likely that a higher percentage of the geared installations would be single rather than twin engined. In the case of the single geared engine, a clutch was not essential as the engine could be direct reversed. Alternatively a reversing gearbox could be used, a number of which existed in tugboats, which would seem to be a good test for any system.

The use of controllable-pitch propellers purely as a means of reversing was difficult to justify on economic grounds in comparison with reversing engines and associated clutches. If, however, an installation was considered as a whole, it was possible, for example, to justify the use of controllable-pitch propellers to permit generators to be driven from the gearing at a constant speed, regardless of ship speed.

The March 1967 issue of *The Motor Ship* gave some interesting comparisons of installations with solid and controllable-pitch propellers, which indicated that for very little, or no, increase in capital expenditure, a worthwhile reduction in fuel, lubricating oil and maintenance costs could be obtained.

In reply to Dr. Watson, the authors said that firstly, to put the record right, the overall length of the medium-speed engines and their gearbox used in the study was 33 ft rather than 26 ft. The gearbox was designed to suit the low propeller rev/min and included friction clutches, both of which contributed to its length.

The main engines occupied, very approximately, half the width of the ship and it did not appear unreasonable to expect that, if the engines were shortened by 19 ft, as in this case, the overall saving in engine-room length could only be in the order of 10 ft.

The authors would agree however that, depending upon the type of ship in question, more advantage might be taken of the low headroom required for the medium-speed engines to make a further saving in space.

Dr. Watson was perfectly correct in drawing attention to the fact that the Burmeister and Wain engine had one of the lowest lubricating oil consumptions on record. If the comparison had been based on the Doxford engine, which had a higher lubricating oil consumption than the B. and W. engine, there were other factors which had to be taken into account, e.g. Table II showed that the Doxford engine would save on length and weight compared with the B. and W. engine. It would, therefore, appear to some extent at least, that what one saved on the swings one lost on the roundabouts.

The reduction in lubricating oil consumption referred to by Dr. Watson, which would be achieved in time, would be a valuable contribution to the economy of the medium-speed engine.

The authors shared Dr. Watson's disappointment at not being able to take into account the cost of replacement spare parts. An initial outfit of spares was, of course, included in the comparison. It was possible to get the costs of a number of replacement parts but the information that was most difficult to obtain was the cost of reconditioning of engine parts and the frequency at which these parts required reconditioning.

Mr. Snead's description of his company's activities in connexion with friction clutches made a useful contribution to the paper. The authors were pleased to see that clutch designs of the capacity required for the medium-speed engines were available and in service, at least on land installations.

The authors and the Institute would be indebted to Mr. Snead if he could at some future date provide the results of the tests at present being conducted on clutches in Japan.

To Mr. Yates, the authors replied that engines in Fig. 1 were shown at the minimum centres practicable. Being vee-engines, they were considerably narrower at the bottom than at the top. Sufficient access was available at the top to get to the cylinder heads etc., and ample space was available at the lower level in way of the crankcases.

The authors agreed with Mr. Yates' statement on the small savings to be gained by burning 3500 sec Redwood 1 fuel as opposed to 600 sec Redwood 1 fuel, and a number of shipowners saved themselves the expense of providing expensive heating equipment, insulation and trace heating pipes by restricting the fuel they bunkered to a maximum of 1500 sec Redwood 1. The type of fuel available however depended upon the trade routes of the ships in question and it was essential for the engine designers to assume that their engines might have to operate on the heavier grade of fuel and design them accordingly.

On the question of lubricating oil quantities, Mr. Yates' point was interesting since his argument was exactly the reverse of Mr. Victory's. However the authors could not agree that the initial oil charge was as low as he stated for the medium-speed engines. It was estimated that approximately 2500 gallons of oil would be required in the system of the twin AO engines.

The authors agreed that the medium-speed engine installation could easily be adapted to drive generators off the gearbox and hence reduce maintenance on the auxiliary Diesels. This question became quite involved, however, since other possibilities had to be considered to determine which was the more attractive and to a large extent what applied to the medium-speed Diesel applied to the slow-speed Diesel. Further remarks on this subject were included in the reply to Mr. Butler.

To date, the only means found of preventing the build-up of valve-seat deposits, particularly with high sodium/vanadium bearing fuels, was the cooling of these seats. On the Ruston AO engine this cooling was obtained by means of special passages through the flame plate, whilst in the Mirrlees K Major, passages were provided in the exhaust valve cage and thus the degree of additional complication was relatively small. It was agreed that water cooling of the valve itself, which involved a multitude of flexible connexions and seals, was not likely to be looked upon with favour by the operator, but this would almost certainly result in lower valve temperatures than any of the simpler methods. Mr. Yates might rest assured that the engine builders were very conscious of the problem of exhaust valve life and were continually keeping a check on this problem. Further remarks on this subject were contained in the reply to Commander Short.

The authors agreed entirely with Mr. Fothergill when he said that it would be unwise to accept the conclusions arrived at as being generally applicable. The paper gave a method of comparing slow-speed and medium-speed engines, but the onus was on the ship designer to use appropriate values of the variables such as fuel cost, interest rates, number of days spent at sea etc.

The question of propeller immersion was referred to in the reply to Mr. Thompson, and engine-room length was covered in reply to Mr. Jackson.

The specific fuel consumptions quoted in the paper were the makers' guaranteed figures, modified to allow for the calorific value of fuel assumed. It was appreciated, in practice, that somewhat lower specific fuel consumptions were frequently achieved due to:

i) a rather higher average net calorific value of heavy fuel than the value of 17 000 Btu/lb assumed;

ii) fuel consumptions were normally slightly lower than the figures guaranteed by the engine maker, since he included a contingency allowance in his guaranteed figure.

The figures from the published defects lists were indeed interesting and at least suggested that the reliability of the medium-speed engine was no worse than that indicated in the paper.

The ship speeds given on page 91 were nominal speeds based on the maximum engine power, while the speeds quoted on page 102 were based on the service power. The authors trusted that the two sets of figures gave rise to no undue confusion.

Mr. Grzybowski had asked for maintenance details for the Mirrlees K Major engine, comparable to those given in the paper for the Ruston AO engine, but based on shipowner's experience. Unfortunately, the K Major had not yet entered marine service and its service on land was of relatively short duration so that figures based on experience were not yet available. Figures taken from the earlier K engine would not really make a fair comparison since the K Major incorporated a number of important modifications to extend its overhaul life when burning heavy fuel, as discussed in the paper listed as reference (5) of the authors' bibliography.

It was difficult enough to give maintenance schedules for engines which had been built and run and hence no attempt was made to produce schedules for the OP engine. One point in favour of the OP engine in this respect, however, was that there were no exhaust valves to maintain.

The reason why costs of replacement parts had not been included was explained in the reply to Dr. Watson.

If the maintenance effort had been based purely on estimates the authors would be inclined to agree with Mr. Grzybowski. The estimates given, however, were relatively conservative figures based on the actual times taken for maintenance of the test engines which were running at Lincoln. Nevertheless, it was admitted that the times quoted were idealized ones and would probably need to have a contigency allowance added to bring them closer to the sort of times achieved by owners in practice. This applied equally well to the Burmeister and Wain engine times, although the appropriate contingency allowance would not necessarily be the same for the two classes of engine.

With regard to the question of engine-room manning, a number of British and foreign owners had adopted medium-speed propulsion machinery during the past few years. To the authors' knowledge none of these ships had increased engine room staffs, but despite this, in some cases the proportion of main engine maintenance carried out by the ship's staff as opposed to shore labour had increased.

As medium-speed engines were developed it was to be expected that the output per cylinder would increase. This would be done by increasing rev/min, b.m.e.p., and cylinder bore, and would of course reduce the number of cylinders required for a given power. The Mirrlees OP engine already showed a trend in this direction.

Mr. Major's comments on engine maintenance went some way toward answering Mr. Grzybowski's comments on maintenance and corroborated the authors' reply to Mr. Grzybowski.

With careful planning and by taking full advantage of the light weight of components and the fact that two engines were provided, there was little doubt that economies could be achieved which would reduce or offset the additional man-hours required for maintenance of the medium-speed engines compared with the slow-speed engine.

The point made by Mr. Major concerning the smaller out-of-balance couples which were produced by the medium-speed engines was an extremely valid one. Although the natural frequencies of hull vibration could be predicted with greater accuracy today, than in the past, the complete avoidance of resonant excitation in the region of the service speed was difficult to achieve and the large secondary couples associated with certain cylinder arrangements on slow-speed engines had sometimes given rise to severe hull vibration.

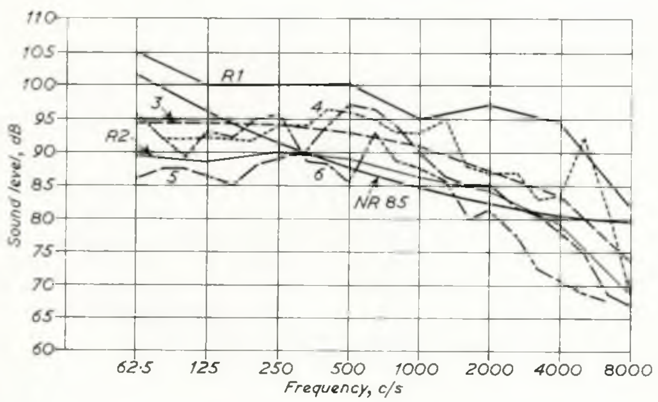
The figures quoted by Mr. Bridle no doubt included an allowance for the cost of replacement parts for the engines. A similar figure to that quoted by Mr. Bridle was quoted by Mr. H. Andresen of Götaverken in a paper to the Society of Naval Architects and Marine Engineers, Philadelphia, but this figure was based on maintenance contracts and could therefore be expected to be somewhat higher than the normal cost to a shipowner who carried out a fair proportion of the work himself. This being the case there was no reason to assume that the costs given in the paper should be increased *pro rata*.

Mr. Dunshea's remarks on medium-speed engines were based on his experience as a ship operator and consequently provided a valuable contribution to the paper which should give encouragement to the builders of such engines.

The authors were pleased to note Mr. Dunshea's opinion that periods between overhaul would be greater than those quoted in the paper. If not achieved immediately this would no doubt be the case as operating experience was gained.

Mr. Dunshea would see from Dr. Watson's remarks that intensive development was being applied to reduce the lubricating oil consumption of the medium-speed engine.

As in their reply to Mr. Fothergill, the authors agreed with Mr. Honour in stating that each shipowner must substitute his own data and values to permit him to make a fair comparison between medium-speed and slow-speed engines for use in his projected ships.




With acknowledgements to the *Shipping World and Shipbuilder*

R1, R2) Extreme values (transposed) measured in 15 ships with slow-speed Diesels of 4500 to 22 000 hp (Storm: Bulletin of the Norwegian Ship/Technical Institute, 1962).

3) Average value in the same 15 ships.

4) Propulsion machinery in escort vessel (medium-speed Diesels, 8000 hp at 465 rev/min), before modification.

5) Propulsion machinery in escort vessel (medium-speed Diesels, 8000 hp at 465 rev/min), after modification.

6) Engine room of steam-turbine escort vessel (20 000 hp at 358 rev/min)

FIG. 13—*Sound level in dB frequency in c/s—Engine-room sound spectra (one-third octave bands)*

If, as Mr. Honour suggested, some shipowners might provide lubricating oil storage for six months, then, based on the bulk carrier, for example, the amount of lubricating oil carried for the slow-speed engine would be increased by 14 tons and the amount of oil carried for the medium-speed engines would be increased by 43 tons. On this basis the amount of oil stored for the slow-speed and medium-speed engines would be similar.

This example showed how the comparison varied according to the basis assumed and, of course, was the reason why, as just stated, it was so important for the shipowner to insert the values to suit his own requirements.

To answer Mr. Honour's query on noise, ideally noise level readings taken in engine rooms with the slow-speed and medium-speed engines in question should be shown. At this stage this was not possible but reproduced in Fig. 13 was a curve comparing noise levels in the engine rooms of various ships, including slow and medium-speed engines. It could be seen that there was not a great deal of difference although, comparing curves 3 and 4, the medium-speed engines were the noisier.

It was recommended that for an exposure to noise for eight hours per day the noise level should not exceed NR85, which had been superimposed on Fig. 13 by the authors. It would be seen that over most of the frequency range indicated in this figure, the noise levels were well above NR85. If, therefore, the recommended noise levels for human exposure were to be taken seriously, whether slow-speed or medium-speed engines were used it was necessary to provide a machinery control room to isolate watchkeepers from the noise.

It was not expected that a simple acoustic screen between a running engine and a stationary engine would reduce the noise levels adequately to permit men to work for long periods on the stationary engine. Earmuffs would give a reduction in noise level of 15 to 20 dB and were the most effective acoustic protection for maintenance personnel.

The question of electric load raised by Mr. Honour had already been dealt with in the replies to Mr. Butler and Mr. Maddocks.

As stated in the paper, the use of friction clutches had been assumed and these of course involved no slip, or in other words they were 100 per cent efficient. At the present time, however, although suitable clutches were under development, none had had seagoing experience at the powers referred to in the paper. This was a factor which a shipowner would have to take into account in assessing the merits of the slow and medium-speed engines.

The authors were indebted to Commander Short for his valuable contribution giving a practical example to show how the advantages of the medium-speed engine could in some cases be exploited by the shipowner. This could only be done if the trouble was taken to examine the alternative ships in detail and it was hoped that the paper would encourage other shipowners to do so. It was to be expected however that in only a percentage of the cases examined would it be advantageous to install medium-speed engines.

Commander Short's description of maintenance and operating procedures and philosophies contained much valuable information and shed most welcome light on the views of a large shipowner. The authors were also grateful for the numerous facts and figures given in Commander Short's contribution, which made it of particular usefulness.

The authors would not dispute the fact that in a completely worked out installation a controllable-pitch propeller might be advantageous.

In a comparison of the nature described in the paper, a number of refinements could have been applied to either or both the slow-speed and medium-speed engined installations, controllable-pitch propellers, shaft-driven generators and steam turbo-generators driven by steam from waste-heat boilers, among them. It was felt, however, that the inclusion of these features would have little effect on the comparison.

The detailed discussion of measures to increase the overhaul life of components such as exhaust valves, was really outside the scope of this paper. However, the authors were under the impression that most engine builders were agreed that rapid deterioration of valve seats when operating on heavy fuels was due to the build-up of deposits which eventually led to the breakdown of the gas seal. It also seemed to be fairly universally considered that if valve and seat temperatures could be maintained at a sufficiently low level, the build-up of these deposits would be minimized. It was in the method of keeping the valves cool that the various makers differed and, in the opinion of the authors, provided that reliability was not impaired or costs increased excessively, the various methods should only be judged on their success in reducing valve deterioration.

The function of valve rotators was to prevent local spots on the valve seats and to even out deposits. If the valve seat temperature was high it was unlikely that the use of rotators alone would make a significant difference to the life. Similarly, if seat deterioration was due to excessive mechanical stresses on the valve seat, rotation of the valve was unlikely to be beneficial.

In reply to Mr. Sinclair, the authors said that they had no personal experience of the Geislinger coupling. Compared with couplings having rubber elements in shear, it was rather stiff and in many installations would not permit the removal of critical speeds from the running range. On the other hand it would probably be able to withstand operation at a critical speed for long periods to a greater extent than would the rubber coupling.

Mr. Sinclair had put forward an ingenious proposal to use a lock-out clutch for positive ahead drive and crash astern manœuvres, in conjunction with a relatively inefficient hydraulic coupling for use during normal manœuvring. Mr. Sinclair's proposal would seem to give the best of both worlds so far as manœuvrability and efficiency were concerned and appeared to be well worth considering.

The system was a little more complicated than the systems mentioned in the paper and, naturally, a careful comparison would have to be made of the costs of Mr. Sinclair's proposals with the alternative arrangements. The authors hoped to have an opportunity of examining a scheme of this type in detail in the not too distant future.

# INSTITUTE ACTIVITIES

## Minutes of Proceedings of the Ordinary Meeting held at The Memorial Building on Tuesday, 22nd November 1966

An Ordinary Meeting was held by the Institute on Tuesday, 22nd November 1966, at 5.30 p.m., when a paper entitled "The Use of Medium-speed Geared Diesel Engines for Ocean-going Merchant Ship Propulsion" by J. Neumann, B.Sc., A.M.I.Mech.E. (Associate Member) and J. Carr, A.M.I.Mech.E. (Associate Member), was presented by the authors and discussed.

Mr. R. R. Strachan (Chairman of Council) was in the Chair and approximately two hundred and fifty members and visitors were present.

Thirteen speakers took part in the discussion which followed.

The Chairman proposed a vote of thanks to the authors which received prolonged and enthusiastic acclaim.

The meeting ended at 7.35 p.m.

## Branch Meetings

### North East Coast

*Junior Meeting*

A junior meeting was held in the theatre of the Marine and Technical College, South Shields, on Monday, 13th February 1967, at 3.15 p.m. when Mr. P. Jackson, M.Sc. (Member of Council) adressed an audience of 490 marine engineering students.

After being introduced by the Honorary Secretary, Mr. A. J. S. Bennett, M.B.E., who was in the Chair, Mr. Jackson recounted the early history of the Diesel engine, the birth of the Diesel engine industry, the growth of Doxford's and the place of the large engine in modern ships. He then showed a number of slides illustrating the development of the Doxford engine up to the present day.

In conclusion a vote of thanks was proposed by Mr. F. Howitt, M.Eng., Head of the College's Mechanical and Marine Engineering Department, and carried by applause.

*General Meeting*

A general meeting of the Branch was held at 6.15 p.m. on Thursday, 2nd March 1967, at the University of Newcastle upon Tyne, when the paper "Operating Experience with Large Modern Turbocharged Heavy Oil Engines" by G. McNee, B.Sc. (Member) and J. McNaught (Member) was presented by Mr. McNaught.

Mr. A. W. Bell, B.Sc. (Chairman of the Branch) was in the Chair and ninety-eight members and guests were present.

Among those taking part in the discussion which followed the presentation were Mr. P. Jackson, M.Sc. (Member of Council), Mr. T. Matthew (Honorary Secretary), Mr. B. W. Martin (Member), Mr. R. J. Hook (Member), Mr. B. Taylor, B.Sc. (Member), Mr. W. Hewitson Menzies (Member), Mr. E. C. Cowper (Member), Mr. G. Dunn, and Mr. A. Abernethy (Associate Member).

The Chairman closed the meeting at 8.20 p.m. with a vote of thanks which was warmly endorsed with applause.

### North West England

A combined junior and senior meeting was held by the Branch at 6.45 p.m. on Monday, 27th February at the Manchester Club, 81 King Street, Manchester 2, when the paper "Motor Ships" was presented by Mr. E. Taylor (Member). Mr. J. K. O'Neill (Chairman of the Branch) was in the Chair.

After the presentation, which was illustrated with slides, the discussion was opened by Mr. O'Neill.

Later, in his vote of thanks to the speaker, the Chairman expressed his disappointment that only ten members were able to attend this most interesting meeting.

### Scottish

*General Meeting*

A general meeting was held on 8th February 1967, at 5.30 p.m. at the Institution of Engineers and Shipbuilders in Scotland, 39 Elmbank Crescent, Glasgow, C.2, when a paper, "The Future Marine Engineer and his Training" was presented by the author, Mr. J. G. Holburn (Member).

Mr. T. W. Liddell (Chairman of the Branch) was in the Chair and eighty-one members and visitors were present

Mr. Holbourn introduced his paper by stating that in the immediate post-war years it became apparent that the old traditional methods of educating and training marine engineers were out-moded and out-dated and were no longer producing men of the calibre required for ships which were becoming increasingly complex.

Technological developments had proceeded, and were proceeding, at such an astonishing pace that the position had been reached, if not passed, where some machinery installations were so sophisticated as to be beyond the complete understanding of all but a very few engineer officers, and were, indeed, beyond the understanding of many superintendents.

Mr. Holburn proposed fundamental changes in the present system of training marine officers. He suggested two systems—the higher system forming the elite of officer potential, capable of taking charge of all types of installation.

There was a strong case for the award of an entirely new Certificate. Mr. Holburn suggested that the Certificate which would appear to offer a better reward would be a marine equivalent of the City and Guilds Mechanical Engineering Technician's Certificate. He considered that it would be a function of the Institute to advise and administer such a course, in conjunction with the Board of Trade, educational establishments and shipowners' representatives. A panel should be set up to decide on a specialized form and desirable content of the examination. To widen the field and draw on the vast experience and knowledge available, the panel could be composed of representatives from the Board of Trade. Lloyd's Register of Shipping, the Institute and technical colleges. To ensure uniformity and unbiased assessments of the candidate's ability, considerable thought should be given to the possibilities of setting the examination by a computer.

The discussion which followed was opened by the Chairman of the Branch and proved most interesting and extremely controversial.

The meeting closed at 8.30 p.m. when Mr. F. Y. Whitham (Associate Member) thanked the speaker on behalf of the Branch.

*Thirteenth Annual Dinner*

The Thirteenth Annual Dinner of the Branch was held on Friday, 17th February 1967, at the Central Hotel, Glasgow, at 7.15 p.m.

Mr. T. W. Liddell (Chairman of the Branch) presided and 355 members and guests were present.

The President of the Institute, Sir Stewart MacTier, C.B.E., B.A., with Mr. Liddell, formally received members and guests at a reception prior to the Dinner.

*At the Annual Dinner of the Scottish Branch. The President of the Institute, Sir Stewart MacTier, C.B.E., B.A. (left), with Vice-Admiral Sir Raymond S. Hawkins, K.C.B., Fourth Sea Lord and Vice-Controller, Ministry of Defence (Navy)*

Following the Chairman's remarks the toast "The Institute of Marine Engineers in Scotland" was proposed by Mr. F. B. Bolton, M.C., President of the Chamber of Shipping of the United Kingdom.

Mr. Bolton stated that today was the day of the professional but one of the processes not very clearly evolved was the transition from being one kind of a professional, to another —in terms of shipping, from the technical department to management. He could not see why the training and experience of the engineer should not be a stepping stone to management. On the other hand, if the syllabus of training and examination became too specialized, it would not be so easy for the professional to acquire what was also claimed as the virtue of the amateur—a balanced mind.

Speaking of research, he said that shipowners were just as progressively minded as managements in other industries and had been trying over the past few years to produce a significant research effort.

"It is only too easy" he said, "to provide a willingness to research without the conviction that the effort is worth while—and to work really quite hard to think what will improve our engines and our ship operations without really ever getting to grips with what needs investigation and improvement. This is where your Institute comes in, since you are dedicated to the improvement of your art, and if you couple your professional knowledge and skill and your determination to achieve your stated objective, to the readiness which I know is there with owners, the shipowner research effort—which is already considerable—could become really impressive".

The President, Sir Stewart MacTier, replied to Mr. Bolton's toast. He said that success in the future would be achieved only if we adopted a far more scientific approach to our problems in the marine industries. In his opinion the marine engineering industries of the United Kingdom were not raising their standards of technological competence sufficiently fast to match their overseas competitors.

On the question of research, he suggested that, in the past, the research activities of the marine industry were too theoretical and too little related to current operating problems at sea. Again, in the past, there had been an unfortunate failure on the part of the marine industries to pool their knowledge and experience to their mutual benefit.

Sir Stewart congratulated Mr. Bolton on the considerable step forward that had been achieved through the activities of the research department of the Chamber of Shipping, and the collaboration of the Chamber with the reconstituted British Ship Research Association.

Mr. W. McLaughlin (Vice-Chairman of the Branch) proposed the toast "Our Guests" to which Mr. W. P. Walker replied.

The top table party consisted of representatives of every branch of shipbuilding, shipowning and engineering. The United States Navy was represented by Captain R. F. Woodall, U.S.N., Commander of Submarine Squadron Fourteen. Vice-Admiral Sir Raymond S. Hawkins, K.C.B., Fourth Sea Lord and Vice-Controller, Ministry of Defence (Navy) was also present.

The Dinner was followed by a Conversazione which continued into the early hours of the morning.

*Joint Meeting*

The Branch held a joint meeting with members of the Aberdeen Mechanical Society at 7.30 p.m. on Friday, 24th February 1967, at Robert Gordon's Institute of Technology, Aberdeen, when Professor Lars Th. Collin, M.Sc., presented his paper "The New Polar Four-stroke Engine". Sixty-three members and guests were present.

Mr. H. Hampson, B.Sc., M.Sc., President of the Aberdeen Mechanical Society, presided and after welcoming Mr. T. W. Liddell (Chairman of Scottish Branch) invited him to take the Chair.

Mr. Liddell thanked the Society for the opportunity of having this joint meeting and said how pleased he was to meet members of the Branch who might be unable to attend the meetings in Glasgow. He then introduced Professor Collin, who was assisted by Mr. Olsson.

Professor Collin gave a brief history of the development of the Polar engine, from the signing of a licence agreement in 1898 between Rudolph Diesel and a Swedish financial group. Diesel engines were manufactured and developed on the one hand by Ludwig Nobel and Co., in St. Petersburg, Russia, and on the other hand, by A.B. Diesel Moterer, later named A.B. Atlas Diesel, who introduced Polar Diesels shortly before the Great War. In 1925 the Nobel activities were taken over by Nydkvist and Holm A.B., or Nohab, in Trollhattan.

Professor Collin described the development of the two-stroke engine with special reference to the development of the N and T types, which utilized the special Polar scavenging system. He went on to describe some of the considerations which were responsible for the introduction of the four-stroke cycle.

He detailed the many points leading to the design of the present F type engine, with a stroke of 300 mm and a bore of 250 mm. He gave a very good description of the engine and, in particular, the accessibility of all parts for overhaul, illustrating his remarks by the extensive use of slides.

Professor Collin closed his presentation with the film *Diesel,* which showed the wide-ranging applications, afloat and ashore, of the Polar engine.

A very interesting discussion followed after which Mr. Hampson thanked Professor Collin for his most interesting paper. The vote of thanks was carried by acclamation and the meeting terminated at 10.00 p.m.

*General Meeting*

A general meeting of the Scottish Branch was held on Wednesday, 8th March 1967, at the Institution of Engineers and Shipbuilders in Scotland, 39 Elmbank Crescent, Glasgow, C.2, at 6.15 p.m. when a paper entitled "The New Sulzer 1050-mm Bore Engine" by W. Kilchenmann (Member), was presented by the author.

Mr. W. McLaughlin (Chairman of the Branch) presided at the meeting and welcomed the 162 members and visitors present.

Mr. Kilchenmann opened the paper by showing that the trend of modern shipbuilding was towards bigger and bigger ships. Only fifteen years ago the output available from an individual Diesel engine was limited to roughly 10 000 bhp. First by turbocharging, then by the introduction of large bore engines, this limit was raised to 27 500 bhp. The modern tankers and bulk carriers of 100 000, 200 000 and more tons, required still higher propulsion powers. Thus, the Diesel engine aimed now at an individual output of around 40 000 bhp where the limit seemed also to be set by the size of a reasonably efficient propeller.

The author went on to describe the four ways to increase engine output—increasing the number of cylinders, raising the piston speed and the mean effective pressure, and by increasing the cylinder diameter; the advances made in each case were also shown. The result was the design of a new, larger RND 105 engine incorporating simplifications and improvements based on practical experience accumulated in the years of successful service with the RD range. The most important point, among many new features, was the application of the constant pressure supercharging system which was explained in detail.

From the operating point of view, the most important change, made possible by the constant pressure system, was the elimination of the rotary exhaust valve.

An extremely vigorous discussion followed, in which Mr. Kilchenmann dealt with the questions in a very confident manner.

A vote of thanks to the speaker for the excellence of the paper and the outstanding way in which the discussion was handled, was proposed by Mr. T. W. Liddell (Member of Committee).

The meeting closed at 8.20 p.m.

## South Wales

*General Meeting*

A general meeting of the Branch was held on Monday, 13th February 1967, at the Gas Showrooms, Swansea, at 7.00 p.m., when a paper entitled "Developments in Marine Pumps and Compressors with Automatic Control of their Systems" by R. J. Gates, B.A. (Associate Member) and L. Sterling (Associate Member) was presented by the authors.

Chairman of the Branch, Mr. T. W. Major, presided and welcomed the thirty-three members and guests present. After reviewing some of the changes experienced with marine auxiliary machinery in the last decade, Mr. Major invited Mr. Gates and Mr. Sterling to deliver their lecture.

The paper briefly described changes brought about over the past ten years in selected designs of reciprocating compressors for starting air and general service duties and of centrifugal and screw displacement pumps.

The paper, which was amply illustrated with slides, went on to show the impact that automation has had upon the design and development of this machinery.

Finally, there was a brief review on the control of pumping systems for main and auxiliary engines, together with the development of automatic control for bilge and ballast pumping.

An enthusiastic discussion followed, regretfully terminated by the Chairman.

In proposing a vote of thanks to the speakers, Mr. R. H. Scott (Member of Committee) congratulated them on making admirably produced preprints of the paper available to the meeting. The vote of thanks was warmly seconded by all present.

The meeting closed at 9.10 p.m.

*Senior Meeting*

A senior meeting with an "Open Forum—Some Practical Aspects of Marine Engineering" was held on Tuesday, 14th March 1967, at the South Wales Institute of Engineers, Park Place, Cardiff, attended by forty-four members and guests.

In the absence of the Chairman of the Branch, Mr. T. W. Major, the Chair was taken by Mr. O. T. Griffith (Vice-Chairman).

Mr. Griffith explained to the meeting that as a result of illness and business commitments several expected participants were forced to withdraw.

It was originally intended that each participant would speak on a branch of marine engineering with which he was familiar and invite questions at the end of each short lecture.

Mr. G. S. Taylor (Member of Committee) opened the Forum by speaking of the problems encountered in the fleet with which he was associated, in particular the methods used to eliminate boiler cleaning at sea between drydockings, and discussed problems experienced in turbine bearings, with stannous oxide corrosion. His talk was amply illustrated with coloured slides and encouraged much discussion.

The Chairman spoke of hull damage sustained by one vessel of his company's fleet. He went on to explain how the damage occurred, what steps were taken to prepare the ship for dry dock and the repairs that were necessary to make the ship seaworthy. Mr. Griffith then answered questions on the repair.

Mr. W. M. Mathieson (Member), Principal Surveyor, Lloyd's Register of Shipping, spoke of a repair carried out on a Doxford crank web *in situ* whilst he was based in Australia. This repair aroused a lot of interesting discussion.

Mr. F. R. Hartley (Member of Committee) proposed a vote of thanks to those who had taken part in the Forum, some of them at such short notice and this was warmly seconded by all present.

The meeting closed at 8.00 p.m.

*Council of Engineering Institutions—Inaugural Dinner*

The Inaugural Dinner of the South Wales Committee of the Council of Engineering Institutions was held at the Angel Hotel, Cardiff, on Wednesday, 1st February 1967 and was attended by 171 members and guests. Speeches were made by Mr. Ifor Davies, M.P., Parliamentary Under-Secretary of State, Welsh Office; Mr. N. S. Williams, Chairman of the C.E.I. South Wales Committee; Colonel Sir Cennydd Traherne, T.D., LL.D., Lord Lieutenant of Glamorgan and Patron of the C.E.I. South Wales Committee; and Mr. C. N. D. Cole, Managing Director of the Western Mail and Echo Ltd. The South Wales Branch was represented by Branch officers and a number of members.

### West Midlands

A general meeting of the Branch was held at 7.00 p.m. on Thursday, 16th March 1967, at the Engineering and Building Centre, Broad Street, Birmingham, when the paper "Some Corrosion Problems in Naval Marine Engineering" was presented by the author, Mr. L. Kenworthy, M.Sc.

Mr. A. Fowler (Chairman of the Branch) presided and thirty-six members and guests were present.

Mr. Kenworthy described the various materials used for heat exchanger tubes and tube plates and the problems encountered with them in service necessitating investigation into new materials.

He referred to sea water systems and the problems associated with their layout affecting the suitability of materials.

In addition to dealing with heat exchangers, he spoke of the problems caused by fungoid attack within all systems, including equipment involving the safety of personnel.

Summarizing his lecture, Mr. Kenworthy highlighted a number of points which he thought should be given prime consideration in all future projects in order to avoid the continual rise in repair bills which were all attributable in one way or another to corrosion attack.

An extremely lively discussion period followed and the meeting closed at approximately 9.00 p.m.

### West of England

*Junior and Senior Meeting*

A combined junior and senior meeting of the Branch was held at 7.00 p.m. on Tuesday, 14th February 1967, in the Small Engineering Lecture Theatre, Queen's Buildings, University of Bristol, when the paper "Practical Welding and Applications" was presented by Mr. F. Pollard, B.Sc., A.I.M.

Before the presentation, the Chairman of the Branch, Captain A. A. C. Gentry, R.N., gave a warm welcome to the audience which numbered twenty-three and which included Mr. F. C. Tottle, M.B.E. (Local Vice-President, Bristol).

Mr. Pollard described the various methods of shielded arc welding, pointing out the advantages and disadvantages as applied to a wide range of metals.

The paper provoked much interest and a lively discussion period followed. Finally, the Chairman proposed a vote of thanks to the speaker and the meeting ended at 9.00 p.m.

*General Meeting*

A general meeting of the Branch was held on Tuesday, 14th March 1967, in the New Lecture Theatre, City of Bath Technical College, at 7.00 p.m. when a paper entitled "The Bristol Siddeley Olympus Gas Turbine" by W. H. Lindsey, M.A. (Member) was read by the author.

Captain A. A. C. Gentry, R.N. (Chairman of the Branch) presided and extended a warm welcome to the fifty-two members and guests present.

The author opened the paper with an account of the advantages obtained by using suitably modified aircraft jet engines of the Olympus type as gas generators for marine gas turbines and went on to describe this gas generator in some detail mentioning with particular emphasis the modifications made to ensure satisfactory operation at sea level and in a marine environment. The design features of the power turbine together with an account of the control system for the complete engine were also described in full.

Mr. Lindsey then discussed the performance of the engine with special reference to the effect of air intake and exhaust conditions on power output and specific fuel consumption, and he made some comment on the effect of service life on performance, engine installation arrangement and noise suppression. Finally, he spoke of the possible future development of this type of engine and the advantages both in specific power and thermal efficiency which could be obtained in the simple cycle engine by the use of exhaust gas heat recovery.

A discussion period of over an hour followed, with many members taking part. A vote of thanks to the author was proposed by the Chairman and the meeting ended at 9.00 p.m.

## Overseas

### Ottawa

*Annual Report*

The Ottawa Branch had sixty-five members at the close of 1966. This membership level is five higher than that recorded twelve months ago. The transitory nature of marine engineers in Ottawa is again reflected in the high turnover in members, some eleven having departed the area during the period.

Three technical meetings were held during the year, one jointly with the British High Commission.

A Cocktail Party and Buffet Supper was held at the *Bytown* Naval Officers' Club on Friday, 3rd June, and was attended by thirty-four members, wives and guests.

On Tuesday, 1st November, as an introduction to Canada's Centennial Year, Mr. Alexander Gilbert, Executive Manager, Cornwall Board of Trade, and member of the Centennial Speakers Bureau delivered a lecture to members of the Branch and their wives on plans for Canada's One Hundreth Birthday celebrations.

It is noteworthy that Lt. Cdr. D. H. Benn, R.C.N., was afforded distinction in that his paper "The Application of Reliability Engineering Theory to Warship Propulsion Plants with Special Reference to the *St. Laurent* Class Destroyer Escorts" was chosen by Council to appear in the bound volume of the 1966 Transactions.

Attendance at meetings has averaged 30 per cent of members, representing a reduction from a figure of 38 per cent for 1965.

During the year the Branch Membership Committee, through the Secretary for Canadian Affairs, sought Council's ruling on the acceptance of engineering and technical students of Carleton and Ottawa Universities and the Eastern Ontario Institute of Technology for the purposes of election as Student members. Council's ruling is still awaited.

E. N. King (*Chairman*)
M. C. Armstrong (*Honorary Secretary*)

*Annual General Meeting*

The Third Annual General Meeting of the Branch was held on Thursday, 12th January 1967, at the *Bytown* Naval Officers' Club, Lisgar Street, Ottawa.

Mr. E. N. King, M.Sc. (Chairman of the Branch) presided. Twenty-five members were in attendance.

Minutes of the Second Annual General Meeting, the annual report and financial statement were read and accepted.

Commodore A. G. Bridgman, C.D., B.Sc., R.C.N. (Local Vice-President) speaking on the degree of participation of Branch members, expressed the need to maintain a high level of attendance at meetings in view of the relatively small size of the Ottawa Branch. He considered however that the average attendance of some 30 per cent of the membership compared very favourably with kindred societies' meetings of his experience. He suggested that members of other engineering groups within the area be invited to participate, as guests, in Branch activities.

With regard to this year's elections, none of the Committee members had yet completed a three year term, the limit set by the local By-Laws, and all members had agreed to remain in office. No nominations had been received for the positions of Honorary Secretary and Honorary Treasurer and the present office holders were re-elected for a further term.

The 1967 Committee is therefore as follows:

Local Vice-President: Cdre A. G. Bridgman, C.D., B.Sc., R.C.N.
Chairman: E. N. King, M.Sc.
Vice-Chairman: *Appointment pending*
Committee: J. H. Birtwhistle
C. F. Collins
Lt. Cdr. T. A. P. Eyre, M.Sc., R.C.N.
Cdr. H. G. Gillis, R.C.N.
R. V. Smith
Honorary Secretary. M. C. Armstrong, P.Eng.
Honorary Treasurer: J. E. Blakey

The main item on the Agenda was "Where Are We Going"—a self-appraisal by the Branch. As an introduction, the Chairman gave a brief outline of the history of the Branch and the policy which had been followed in presenting papers and social programmes. The Committee was anxious to be guided by the membership, in the latter's reaction to past policies, before proceeding with future programmes. Cdr. H. G. Gillis then outlined future possibilities for social events and Mr. R. V. Smith spoke on the prospects for increased Branch membership.

In the following discussion on papers, the main points which emerged were 1) the need to improve the standard of papers, without necessarily restricting presentations to marine engineering topics, 2) the need for more advanced planning to ensure that advance copies were always available and formal written discussion prepared, 3) choice of an evening and time for presentation which would ensure maximum attendance and 4) the widest possible circulation to interested parties of meeting notices. To meet the above needs, the following suggestions received majority support, 1) a strengthened Papers Committee with co-opted assistance from the membership, 2) increased attention to advance planning, 3) co-operation with other Branches in seeking out and advising of high calibre papers, 4) greater effort to secure papers from within the Ottawa Branch, 5) meetings to be arranged to commence shortly after the end of office hours and 6) an increased mailing list to include S.N.A.M.E. and other kindred organizations.

Discussion on the social programme resulted in majority support for a cocktail party in the Fall as the year's main social event.

Queries were raised on the small numbers of new members joining the Institute in Ottawa. What could the Branch offer to prospective members? However, in view of forthcoming changes to Institute By-Laws, which would affect eligibility for membership, further discussion on this subject was held in abeyance.

Mr. T. M. Pallas, P.Eng. (Secretary for Canadian Affairs) then gave details of the Canadian Division meeting to be held in Montreal, in June 1967. This meeting would be a highlight of Institute activities in Canada. A contingent of U.K. members including the President, and Director and Secretary would be attending. Maximum participation of members from neighbouring Ottawa was anticipated and the need for early reservations was stressed.

## Vancouver

*Annual General Meeting*

The Annual General Meeting was held at the Sands Motor Hotel, Vancouver, B.C., on 19th January 1967.

The meeting was preceded by a social gathering. Dinner commenced at 7.00 p.m. under the Chairmanship of Mr. W. Dey (Chairman of the Branch), who welcomed the thirty-seven members present.

Scrutineers were appointed for the ballot of Committee members. The members elected were Messrs. P. A. Dale, D. I. McGuinness, B.A.Sc., K. Nicol, N. Sigsworth and J. Watson.

As Mr. W. Dey, Chairman of Branch, and Mr. J. Forsyth, Honorary Treasurer, were resigning, the Committee appointed Mr. D. I. McGuinness as Chairman and Mr. N. Sigsworth as Honorary Treasurer. Mr. R. W. Brown, Honorary Secretary, was re-elected.

The Committee for 1967 consists of the following:

Chairman: D. I. McGuinness, B.A.Sc.
Committee: T. F. Annan
G. Baldwin
J. P. Brydon
P. A. Dale
K. Nicol
J. Watson
Honorary Secretary: R. W. Brown
Honorary Treasurer: N. Sigsworth

Mr. J. A. Forsyth, Honorary Treasurer, presented his Annual Report.

It was reported that three meetings had been held during 1966, one jointly with the Society of Automotive Engineers. There was continued activity as regards membership applications.

Mr. Rennie, Local Vice-President, expressed his thanks to the retiring Chairman and Treasurer for their services and hoped they would continue to be enthusiastic members.

A general discussion was held concerning coming events for the year which the Committee noted. Mr. W. Dey, Chairman, outlined events concerning the Division Conference to be held in Montreal.

After the business meeting a film was presented on "The Generation of Electricity by Atomic Energy".

## Vancouver Island

*Annual Report*

Once again there was an appreciable increase in membership to ninety-one as of 31st December 1966. There has been a steady annual increase as proven by the following figures:

| | |
|---|---|
| Year ending 1963 | 63 members |
| Year ending 1964 | 70 members |
| Year ending 1965 | 80 members |
| Year ending 1966 | 91 members |

This year is the Canadian Centennial and the Branch hopes to celebrate this by increasing to 100 members.

Seven meetings of the Branch were held this year; three technical meetings, one jointly with members of S.N.A.M.E., the Engineers Institute of Canada, Professional Engineers, and Power and Hydro Engineers; the Annual General Meeting; the Annual Dinner in the Wardroom of H.M.C.S. *Naden* on 28th April; a technical visit to C.C.G. Weathership *Vancouver* on 20th October, and a film evening on 23rd November when members and visitors saw the B.C. Hydro Columbia River Development and Island Tug and Barge new film "Highways of the Sea".

Six Committee meetings were held during the year to maintain the business of the Branch during the season.

Letters of thanks were forwarded to the President of the Dockyard Officers' Club, H.M.C. Dockyard, Esquimalt, B.C., for the use of the Club facilities during the season, and to the Master Attendant for arranging clearance at the Dockyard Main Gate and for providing facilities for parking during meetings.

D. McKinnon (*Chairman*)
J. McPherson (*Honorary Secrctary*)

*Annual General Meeting*

The Annual General Meeting of the Branch was held in the Dockyard Officers' Club on Thursday, 26th January 1967, with Superintendent J. A. Reader, R.C.M.P. (Local Vice-President) presiding.

Business discussed during the evening:

Cdr. J. S. Osborn, R.C.N., appointed Branch delegate to attend the Canadian Division Business Meeting and the Conference to be held in Montreal 7th to 10th June 1967.

Reports of Officers were presented and the Officers thanked. In accordance with the By-Laws the Honorary Secretary and Treasurer retired from their respective positions and indicated their willingness for re-election.

Elections were held for the following positions:

Honorary Secretary
Honorary Treasurer
One Committee member

Nominated and elected were Mr. J. McPherson, Mr. G. W. Holme and Mr. H. B. Brett respectively.

Mr. H. B. Brett will replace Mr. W. White who retires from office.

### Singapore

*Annual Report*

The last Annual General Meeting of the Singapore Branch of the Institute was held at the Shell Theatrette, Shell House on Friday, 11th March 1966. The annual report was read and unanimously accepted. The annual financial statement was read and unanimously accepted. Under the heading of "Any other Business" reference was made to the proposed Institute of Engineers, Singapore, and other matters relevant to the Branch.

During the year the number of corporate members in the branch had reduced slightly due to transfers from the area. The total stood at fifty-six of which 25 per cent were floating staff. There was a slight increase in the number of Student members. Graduates and Students now total seventy.

The death of the Local Vice-President, Mr. S. A. Anderson, O.B.E., during the year was deeply felt throughout the Branch Members attended the church service held at the Presbyterian Church commemorating the many aspects of his active and useful life in Singapore.

Members welcomed the appointment on 29th July of Mr. J. M. Mair, Superintendent Engineer, Straits Steamship Co., Ltd., as Local Vice-President.

The Committee met on five occasions during the period of office to deal with matters relating to the Branch. It is regretted that with fewer corporate members in the Branch and increased responsibilities and travelling falling to those remaining, it was not found possible to organize any functions during the year.

Various professional engineering bodies in Singapore had combined to form the newly inaugurated "Institute of Engineers, Singapore". Members had been kept informed of developments and whilst the Branch had representation on this comprehensive body, it was being found that the qualifications for entry are higher than those which had normally been acceptable for marine engineers. For this reason there was every need for an active branch of the Institute in this area.

E. Daniels (*Honorary Secretary*)

### Sydney

*Annual General Meeting*

The Annual General Meeting of the Sydney Branch was held at Science House, Gloucester Street, Sydney, at 6.00 p.m. on Wednesday, 1st March 1967.

Captain R. G. Parker, O.B.E., R.A.N., Local Vice-President for Sydney, took the Chair and opened the meeting by extending a welcome to the seventy-four members and guests present.

The Minutes of the previous meeting were taken as read, confirmed and signed.

The Annual Report and Balance Sheet for 1966, which had previously been circulated to all members, were approved.

The Honorary Secretary then reported the result of the election of the 1967 Committee as follows:

Chairman: C. Bie
Vice-Chairman: H. Gerrard
Committee: F. W. Davies
Capt. P. G. Elliott, R.A.N.
K. R. Longes
J. A. McGillivray
Honorary Secretary: J. W. Lamb
Honorary Treasurer: K. McC. Murray

The Chairman opened his address by thanking the 1966 Committee for the support and assistance they had given during the year—particularly the Honorary Secretary and Honorary Treasurer, whose work behind the scenes contributed in no small measure to the smooth operation of the Branch. He also thanked members for their good attendance at meetings and asked them to continue their efforts in encouraging other members to attend, and other marine engineers to join the Institute.

In referring to his recent appointment as Local Vice-President for Sydney, Captain Parker assured members that he would promote the interests of the Institute to the best of his ability.

Captain Parker then invited the new Chairman, Mr. C. Bie, to take the Chair. At this point Mr. W. G. C. Butcher rose and said that on behalf of members present he wished to congratulate Captain Parker on his appointment as Local Vice-President, and thank him for his service as Chairman during the past year. (Applause.)

On accepting the Chair Mr. Bie said he would do his best to uphold the high standard set by his predecessors. He then introduced the speaker for the evening—Commander A. A. Townsend, R.A.N., who presented a film and talk entitled "Experiences in an American Shipyard".

The talk was followed by a short but lively discussion and a vote of thanks to the speaker was proposed by Mr. H. Gerrard.

The proceedings terminated at 8.15 p.m.

### Election of Members

*Elected on 6th March 1967*

MEMBERS

*Elections*

Donald Ure Alexander
Norman Arthur Andrews, Eng. Lt. Cdr., R.N.
Randall Berry
John David Armstrong Burn
William Cain
Francis Philip Crum
John Ellison Erb
George Keith Miller
Donald Alexander Orr
George Paton
Joseph William Pennyfather, Eng. Lt. Cdr., R.N.
George Rainy Peterson, B.A. (Hons.)
Keith Edward Piper
Charles Coats Purdon
Alfred Willmott Robinson
Harold Smith
Charles Edwin Sundbye
Alfred Harold Webb

*Transferred to Member from Associate Member*

John Clark Button
Kenneth Gibson Collinson
Bryan Patrick Robert Cumings
Charles Edmund Gay
Ronald Leslie Gray
William Brown Leitch
Thomas Orr Leith, B.Sc.
Ronald John Lidguard
Maung Sein Maung
James Ronald Parker
Frederick Edward Wood, Lt. Cdr., R.N.

*Transferred to Member from Associate*

Henry Russell Boyle, T.D., J.P.
Alan Wesley Brew
Robert Francis Manning

*Transferred to Member from Graduate*

Richard Embleton Burn
Ian Campbell, B.Sc.
Renato Faresi

ASSOCIATE MEMBERS

*Elections*

Leonard Anderson
Wilfred Edwin Robert Blacker, Eng. Lieut., R.N.

Ralph Geoffrey Burn
John Butler
John Terry Brunton Chard
George Clyne, B.Sc.
James Lees Derries
Roland Drew
Peter Durham
Robert Ellis
Fouad Iskander Gaied
Archibald Nigel Tony Harry
Gerald Anthony Hathaway, Eng. Lieut., R.N.
Leslie Holland
Geoffrey Stuart Hubbard, B.Sc.
Murdoch McDougall
Ivor Joseph Murzello
Alfred Charles Reeves
George Alec Shimmings, Eng. Lieut., B.E.M., R.N.
Eric Ernest Simpson
Colin Earl Spencer
Malcolm George Bridgeman Wannell
Thomas Michael Williams

*Transferred to Associate Member from Associate*
Robert Neale McKenney

*Transferred to Associate Member from Graduate*
Lawrence Brownson
Norman William Duke
William Little
Charles Harvey Meeke
Robert William Miller
Eric Macdonald Satterley
William Shou Chen Wong, B.Sc.

*Transferred to Associate Member from Student*
Ralph Ian Oxford
John Joseph Sullivan

*Transferred to Associate Member from Probationer Student*
Michael John Andrews

ASSOCIATES
*Elections*
Frank Charles Blight
Alfred Thomas Brooks
Anthony Claud Hope Childs, Lt. Cdr., R.D., R.N.R.
George Nicholas Christodoulou
Paul Antony Duggan
Michael Creasey Frost
Graham Garth
Norman George Grimes
John Cochrane Malleny
James Bogue Marjoribanks, Cdr., R.N.
Iona Moriel
John David Priestley
Gannavaram Narasimha Ravi
Kenneth Boyd Swanson
Yun-wing To
Roger Vickers
Kenneth Williams

*Transferred to Associate from Graduate*
Terence Charles Whitney Booth
Colin William Stuart Piggott
Alan Edwin Savage

*Transferred to Associate from Student*
Gordon Leslie Treliving, Eng. Sub. Lieut., R.N.

GRADUATES
*Elections*
Ibrahim Kavrak, B.Sc.
George David Kinrade
Howard Allan Mumford
Anthony John Watson
Stephen Young

*Transferred to Graduate from Student*
Peter Douglas Brock
Ian Turner Coffer
William Dickinson
James Henry Stoppa
Wan Wing-Kin
Allan Godfrey Willis

*Transferred to Graduate from Probationer Student*
Colin Avery
Hugh Arthur Comley
John English
Michael John Alexander Powell

STUDENTS
*Elections*
Peter Longmuir Balmain Anderson
Campbell Barrie
Sydney Wilson Berry
Allan Paterson Caldwell
Gordon Cameron
Alexander D. Campbell
Dale Clark
James Alexander Munro Cormack
Alexander Young Cuthbertson
Christopher John Evers-Swindell
Md. Habibullah
George Henderson
Alexander Graham Hutton
A.K.M. Shahidul Islam
Robert Graham Liddell
Alexander MacLeod
Donald Thomas Gordon MacRaild
Andrew Ingram Milne
Stanley Pallister
David Kenneth Riley
Alan John Trevelyan Robinson
William Angus Robson
Barrie McNaughton Sinclair
Soo Hoo San
Thomas Sutherland Wallace
Derek Harry Warner
Rodney Williams

*Transferred to Student from Probationer Student*
Malcolm Richard Bowman
Thomas Joseph Burridge
Howard John Cox
Wilfred George Turner

PROBATIONER STUDENTS
*Elections*
John Docherty Cameron
Anthony Michael Cantrill
Malcolm Charles Cater
Stephen Frank Fielder
Robert N. Fullerton
Matthew Gibbs
Bryce Gorman
Neil R. Grant
Robert Charles Hearson
Anthony Roy Hemsley
Paul Royston Howe
Paul Rodney Jarvis
Graham Mackintosh Lewis
Charles John McCrossan
Alun Hugh Morgan
Hamish Stephen Robertson
Colin Donald Ross
James Sibbald
Crawford Cilfillan Steedman
Ian Sutherland
Michael John Whittaker