Paper read at the Institute on Tuesday, 20th April, 1971

MEDIUM SPEED ENGINES IN SHIPS

Dipl.Ing. K. Luther*

Geared multiple engine plant came into use in ships some years ago, admittedly only in smaller vessels with relatively low propulsive output to begin with. With the development of all makes of larger engines with higher output, able to run on heavy fuel oil without difficulty, such plant has been introduced increasingly into larger and faster vessels of almost every type. Some examples of installations for such plant are given and the essential advantages of marine medium speed engine plant over alternative forms of propulsion outlined.

The design features of two medium speed engines developing 560 and 1000 bhp (metric) per cylinder are described and their operating data specified.

As special attention must be paid to the questions relating to maintenance and overhaul in plant comprising a relatively large number of cylinders, all the measures taken in these engines to facilitate such work are treated in detail. The damping of noise produced by such engines is also discussed. Details are given of the service life and periods between overhauls for the essential parts subject to wear and, in conclusion, comments are made upon the question of the reliability of these engines.

Mr. Luther

APPLICATION OF THE MEDIUM SPEED ENGINE IN SHIPS

In recent years, technical committees and journals have dealt, to an increasing extent, with marine layouts incorporating higher output, medium speed engines. While there is nothing new, in principle, about this trend, a new aspect is its application to all fields of shipping, with the general tendency to use increased outputs, employing, to this end, larger medium speed units of higher specific ratings.

DEVELOPMENT OF THE GEARED INSTALLATION

Geared multiple engine marine plants have been used for certain marine layouts for many years. The size of the ships ranged from about 4000 to 8000 dwt and the total output from 2000 to 6000 hp. Thus, a large number of these plants was built in Germany in the early post-war period, for which submarine engines built during the war were used. Later on, other plants were modelled on these lines. Since the design of the entire propulsion plant involved some difficulties—not actually due to the engines themselves—and since the output of two-stroke engines had increased—these engines being especially suitable for heavy fuel oil operation—the importance of medium speed engines and, consequently, geared engine plants, decreased again. However, engine manufacturers have, meanwhile, developed new higher output four-stroke engines, i.e., engines also suitable for heavy fuel oil operation. Although these engines are not very much larger than the previous ones, higher piston and engine speeds, higher mean effective pressures and higher degrees of turbocharging, together with the vee-arrangement of the cylinders, have resulted in higher outputs. Another important point to be borne in mind is that gearbox manufacturers are now in a position to produce reliable reduction gearboxes of the desired rating, and that oil companies have meanwhile developed lubricating oils enabling trunk-piston engines to run on heavy fuel oil.

PRESENT POSITION IN RELATION TO VARIOUS SHIP TYPES

In recent years, these plants, although with lower outputs, have mainly been used where the chief merits of a geared plant, i.e., its smaller size and lower weight, have been of particular importance. As well as in warships, geared plants were often installed in ferry boats, the number of which has increased considerably due to the rapid rise in the motor transport of passengers and goods. In ferry boats, a low engine height is of particular importance, as it facilitates the arrangement of one or several car decks extending over the entire length of the ship.

A steep rise in the number of ships with geared engine plants and the introduction of such plants into ships of all types has been made possible by the development of medium speed engines suitable for efficient heavy fuel oil operation. Consequently, these engines have now been introduced into normal cargo vessels and tankers used in merchant shipping. Fig. 1 shows that ferry boats with medium speed engines had, to a certain extent, reached saturation point in 1966. Furthermore, there was no considerable rise in the number of special duty vessels, such as dredgers and factory ships. A remarkable feature is the steep

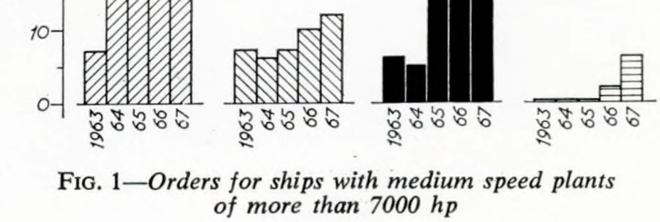

FIG. 1—*Orders for ships with medium speed plants of more than 7000 hp*

* Manager, Medium Speed Four-stroke Engine Dept., M.A.N.

rise in orders placed since 1965 for cargo vessels with geared medium speed engine plant. It seems that, after some initial hesitation, the great advantages of these plants were recognized.

SOME LAYOUTS INCORPORATING MODERN HIGH OUTPUT, MEDIUM SPEED ENGINES

What made a large number of economically minded shipowners choose geared medium speed engine plants? For such layouts this engine design offers many advantages. The author proposes first to give some brief examples of the space and weight requirements. In Fig. 2, an 18 000 hp propulsion plant, with a

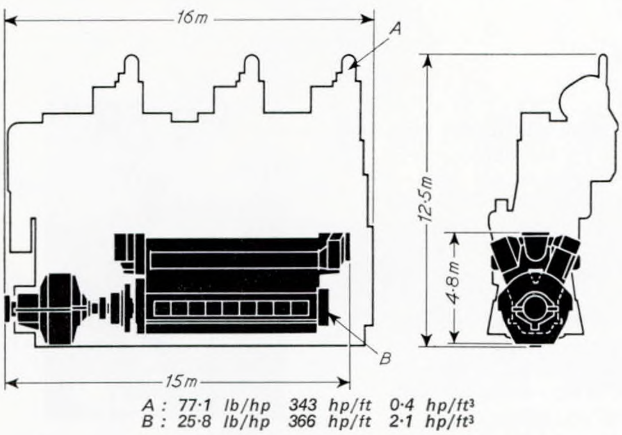

FIG. 2—*Comparison between an 18 000 bhp layout (V9V52/55) and a two-stroke unit*

modern crosshead engine, is compared with an 18 000 hp propulsion plant incorporating a medium speed engine of the latest design, coupled to a planetary gearbox with a built-on thrust bearing.

This gearbox is of advantage if it is important to save weight and space, if the shafts must not be out of alignment and if the gearbox does not have to provide for auxiliary power take-off. The advantages of the medium speed engines are particularly noticeable from the specific values, i.e., the power/weight ratio, the output in relation to the length of the entire plant, and the output in relation to the complete space requirement.

Other examples of medium speed engine layouts are shown

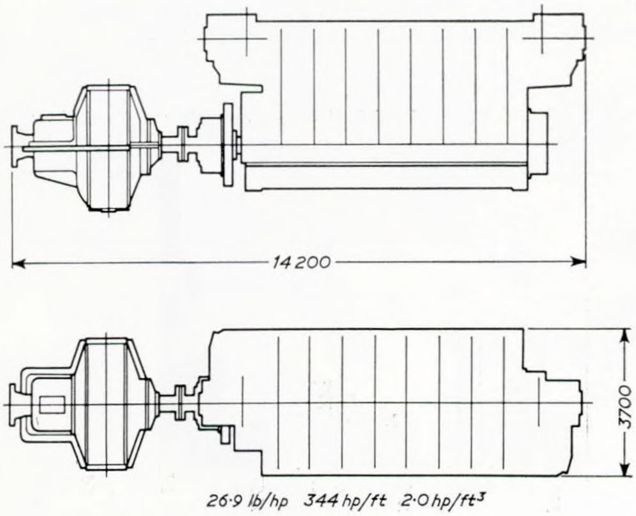

FIG. 3—*16 000 bhp V8V52/55 engine with epicyclic gear*

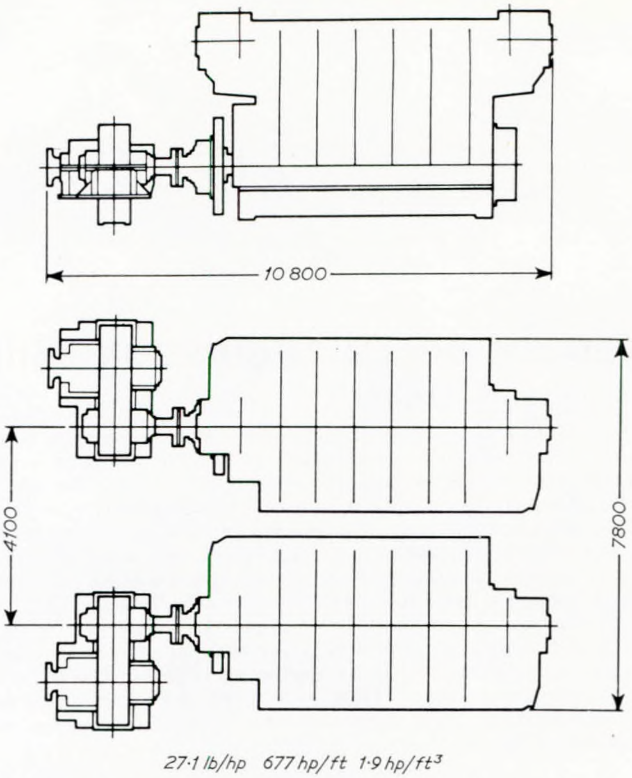

FIG. 4—*24 000 bhp, twin-propeller plant with two V6V52/55 engines*

in Fig. 3 (16 000 hp, 16-cylinder vee-engine with a planetary gearbox) and Fig. 4 (a twin-shaft plant with two 12-cylinder vee-engines and associated reduction gearbox for 24 000 hp). Here again the favourable specific values for space and weight requirement are remarkable.

ADVANTAGES OF THE MEDIUM SPEED ENGINE COMPARED WITH OTHER PROPULSION UNITS

A number of other features are worth mentioning which, when compared with various marine propulsion plants and stationary generating plants, are favourable to the medium speed engine, some applying to the four-stroke engine in particular. It is obvious, however, that they are not all of the same importance when considering one particular engine plant. Nevertheless, it is worth while taking a look at them.

The low weight and space requirement has already been mentioned.

The particularly low height, enabling shipbuilders to provide an additional, continuous deck in ferry boats, has been mentioned. In the case of other types of vessel too it is possible, for instance, to arrange the engine control room, the workshop, stores or auxiliary engines above the main engine room.

To make optimum use of a ship's cargo space, shipbuilders try to arrange the engine plant as far aft as possible. As the weight of the medium speed engine plant is considerably lower than that of other plants, even an empty vessel can be trimmed to an almost even keel, which is hardly possible in the case of vessels incorporating heavier engine plant. Consequently, the ship does not have to go into an expensive floating dock and can use a dry dock. It is also important that, with the empty ship trim, the longitudinal bending moment can be reduced by the lower machinery weight.

The low centre of gravity is advantageous when heavy cargo has to be loaded and unloaded. This is of importance also in container ships and all other vessels having deck cargo.

The reduction gearbox permits free selection of the propeller speed, according to the size and speed of the ship concerned. This ensures optimum propeller efficiency, which demands between about 130 and 140 rev/min in the case of outputs exceeding 20 000 hp and in small ships sailing at high speed, whereas in a large ship sailing at low speed, such as a tanker, a lower speed, between 80 and 90 rev/min, is required.

The possibility of using overlapping or contra-rotating propellers, now under examination by leading shipbuilding and propeller firms, is also increased by the inclusion of a gearbox.

In medium-speed engine plants, the main engines can be used to drive generators through gearboxes, not only at sea, but also in port, when adequate electric power is required for the ship's cargo handling equipment. Under certain conditions, the heavy fuel oil, normally used to drive the main engines, can also be used for this purpose. In most cases, a single Diesel generating set will then suffice to supply the electric power required in port when no cargo is being handled. For this generating set, a high speed unit involving low initial cost and installation charges can be used, as it will only be operating for short periods of time. Consequently, it is possible to save the expense of at least two auxiliary sets. In a geared plant of up to 16 or 18 main engine cylinders, a saving in cylinders in the entire installation, and, thus, in the associated maintenance costs and work, is possible. A 16 000 hp layout illustrates this point (Fig. 5). In tankers, these plants offer the additional advantage that one of the main engines can be used to drive the cargo and ballast pumps. Thus, the expense of separate pump drive units can be saved. In relation to the initial costs of the pump drives, the degree of utilization would have been very low. Fitting a steam turbine set (fed by the exhaust boiler) to the free shaft end of the geared generator would improve the economy and, in the event of an engine failure, the reliability. This turbine set would be fed by the excess steam generated by the plant under most service conditions and could even supply power to the propeller shaft (Fig. 6).

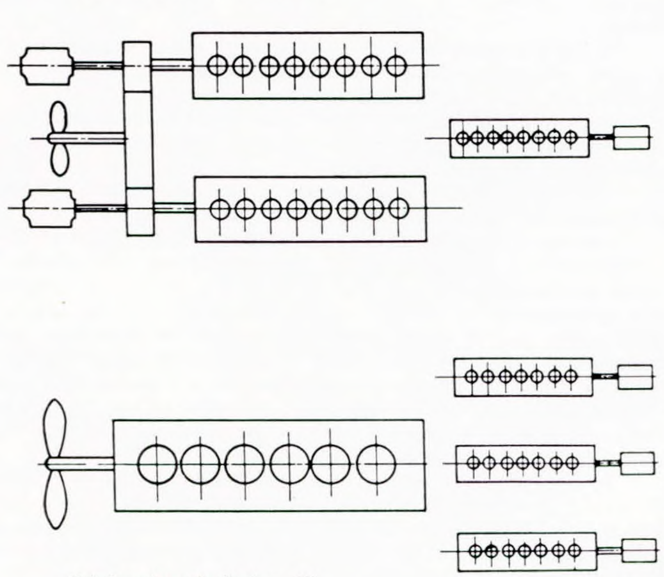

FIG. 5—*Comparison of total numbers of cylinders for 16 000 hp propulsion plants*

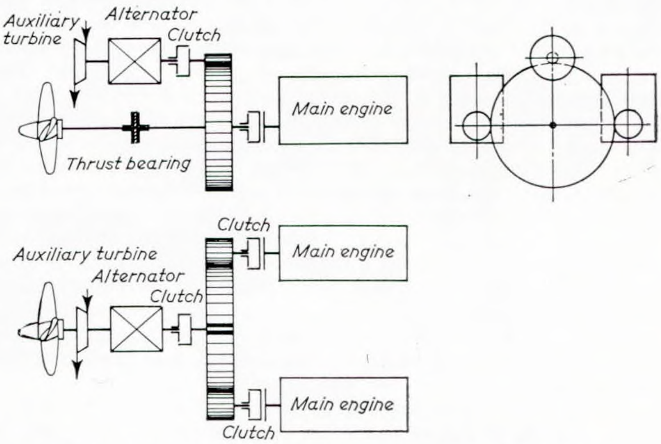

FIG. 6—*Medium speed geared propulsion plant*

Medium speed engines can not only be transported, almost completely assembled, by rail or road, but it is also possible to install complete engines in ships. The advangage of not having to consider the engine installation in the course of shipbuilding is that it simplifies the shipyard's planning work; also the engines

TABLE I—COMPARISON OF SHIP PRICES

Different propulsion arrangements in a 115 00 dwt tanker

	Steam Turbine	*Low Speed Diesel*	*Medium Speed Diesel*
Make	Kockum Delaval	MAN K7Z 105/180	MAN 2 × V8V52/55
SHP(mcr)/rpm	24 000/91·5	28 000/106	28 800/90
Service output, per cent	100	90	90
Speed at service output, kn	15·6	15·6	15·8
Alternators:			
steam drive	1 × 1000 + 1 × 2000 kVA		
Diesel drive		2 × 1000 + 1 × 2000 kVA	1 × 1000 kVA
gear drive			1 × 1000 + 1 × 2000 kVA
Cargo pumps:			
steam drive	3 × 3400 m^3/h × 130 m		
Diesel drive		3 × 3400 m^3/h × 130 m	
gear drive			3 × 3400 m^3/h × 130 m
Ballast pump:			
electric drive	1 × 3400 m^3/h × 30 m	1 × 3400 m^3/h × 30 m	
gear drive			1 × 3400 m^3/h × 30 m
Propeller	Fixed pitch	Fixed pitch	Variable pitch
Steam plant		2 × 8 + 1 × 2·5 t/h	5 × 8 + 1 × 2·5 t/h
Length of engine room	a	a + 1·6 m	a — 4·0 m
Moulded depth	67 ft 5 in	68 ft 3 in	66 ft 4 in
Index price, per cent	100	106	102 (100·7)

can be assembled under more favourable, clean workshop conditions.

The spare parts are not only cheaper but also easier to handle and to store aboard ship.

The first cost of a medium speed engine plant is, in most cases, up to 30 per cent less than that of other conventional layouts. However, this difference does not always have a decisive influence on the price of the whole ship. A comparison of prices often gives very different results depending on the size and type of the vessel. In this connexion, the comparison between three 115 000 dwt tankers is very interesting. A well-known Swedish shipyard prepared this on the basis of existing turbine plants and low speed as well as medium speed M.A.N. engines, making use of all the possibilities mentioned so far (Table I). In the case of the turbine ship, the generators, cargo and ballast pumps are driven by steam and no auxiliary boilers have been provided. The ship with the low speed engine has three Diesel generating sets, as usual, and three Diesel driven cargo pumps, as well as an electrically driven ballast pump and an auxiliary boiler. The ship with the medium speed engine has two generators mounted on the gearbox, a standby Diesel generating set, gearbox mounted cargo and ballast pumps, as well as an auxiliary boiler. The first interesting point is that the engine room of the ship with a low speed engine is 5·5 ft longer than that of the turbine ship. The engine room of the medium speed engined ship however is 13 ft shorter than that of the turbine ship. The difference in height is also noticeable. If the total price of the turbine ship is assumed to be 100 per cent, the price of the low speed engined ship is 106 per cent and that of the medium speed engined ship 102 per cent, i.e., it is only slightly above that of the turbine ship, despite the controllable pitch propeller used. Because of the slightly higher output of the medium speed engine selected, the ship is about 0·2 kn faster and, if this higher speed is converted to cargo capacity, the disadvantage in price becomes negligible.

The operating experience gained with four-stroke, medium speed engines in recent years has also shown that the wear of pistons, piston rings and cylinder liners is considerably lower than in two-stroke engines. Consequently, it is to be expected that the time until the pistons have to be pulled is longer and that it will be possible to have two-year periods between piston overhauls.

Table II shows the wear rates of M.A.N. medium speed

TABLE II—40/54 ENGINE—WEAR ON HEAVY FUEL OPERATION

	Average value in/1000 h	m.s. *Malmanger* in/1000 h
1. Piston ring rad.	0·00078–0·00118	0·00236
1. Piston ring groove	0·000039–0·000156	0·000196
Cyl. liner t.d.c. 1. ring	0·00039	0·00078

engines after a total operating time of approximately 400 000 h. The date relating to the engines of m.s. *Malmanger*, which were running on poorer quality heavy oils is shown separately to demonstrate that even with fuels of this class (ash content 0·06, insolubles in pentane 10, sulphur 3, vanadium 224, 1500 sec./Redwood I) the wear rate of vital components is extremely low.

There are also other advantages, not only over low speed two-stroke engines, but also over turbines:

The splitting of the main power plant into several engine units increases the ship's safety.

The use of one particular type of engine within a fleet permits a large number of different sizes and speeds, the engines having different cylinder numbers, but the same cylinder dimensions and, consequently, the same parts subject to wear. This simplifies overhaul work and the stocking of spare parts, making it easier to change staff within the company.

In oil ports or certain parts of Asia where very bad weather is likely to occur, the propulsion plant must always be ready for service so that in an emergency the ship can immediately leave port. This is rather awkward when planning overhaul or maintenance work on a turbine or low speed engine. In the case of geared multiple engine layouts, however, one can easily work on one engine and keep the others ready for service, according to the regulations.

It is also possible to carry out certain work on an engine while the ship is at sea, if necessary. This can also be done while sailing under ballast or, in ferry boats in the off-season, when the sailing schedule is not so tight and the full engine output not always required.

In the larger layouts, allowance is made in the design and the timetable, right from the start, for the routine maintenance of certain components. During the voyage, for instance, some of the pistons of one engine can be exchanged for reconditioned ones. The pistons or parts removed can then be reconditioned during the voyage and subsequently placed in a shipboard spare parts store. The loss in time caused by the stopping of the engines is allowed for when establishing the timetable of the vessel.

In the foregoing cases or under similar circumstances, such a layout approaches optimum efficiency in operation as individual parts of the plant can be shut down.

It is well known, too, that these layouts facilitate manoeuvring when running in confined waters, when engines can go ahead while the others go astern.

The cargo pump is operated at a considerably higher efficiency than those in turbine vessels. More than 10 000 hp is required for this purpose in large tankers. Owing to the considerably lower efficiency of the auxiliary turbine, the relevant steam consumption is very high as it is required for an output of approximately 36 000 hp from the main propulsion machinery. This means that it might even be necessary to dimension the boiler plant primarily with a view to meeting the steam requirement of the cargo pumps. In any event, it will be necessary to operate the whole main boiler plant, in order to be able to run the cargo pumps. This leaves little or no possibility of carrying out overhauls on the boiler plant in port.

So far the advantages of medium-speed engines have been discussed in detail. When discussing their disadvantages, the following arguments are usually put forward: high lubricating oil consumption, short service life of the exhaust valves in heavy fuel oil operation, higher cylinder numbers and, consequently, more maintenance work, etc, higher mean effective pressures, higher noise levels.

Whereas the advantages mentioned are either undisputed from the very outset or must be assessed by the owners in the light of economic evaluations and optimization for the type of ship concerned and its intended duty, the disadvantages mentioned concern the engine builder. These points will be dealt with when details of medium speed engine are discussed.

MAIN CHARACTERISTICS OF M.A.N. HIGH OUTPUT, MEDIUM SPEED ENGINES

The M.A.N. production programme for medium speed engines was extended to obtain higher unit outputs and, in the course of this, two new engine types were developed, with cylinder outputs of 560 and 1000 hp respectively, covering and engine output range of 3000 to 18 000 hp. Like previous engines, they operate on the four-stroke cycle since experience shows that at this size and speed they offer more advantages than two-stroke engines, particularly as regards thermal loadings, specific wear, lubricating oil consumption, heavy fuel oil operation and part-load operation.

The 40/54 engine (15·7 in bore by 21·2 in stroke, 430 rev/min) has been built since 1964, meanwhile becoming very popular and establishing an excellent service record in marine as well as stationary applications. So far, 250 engines of this type have been sold, totalling over 1·6 million horsepower. Out of this number a little less than 150 engines are already in operation.

Main Design Features of the 40/54 *Engine*

The pistons, arranged opposite each other in the vee-engine, drive an articulated connecting rod (Fig. 7), thus offering the advantage of an appreciable saving in engine and engine room

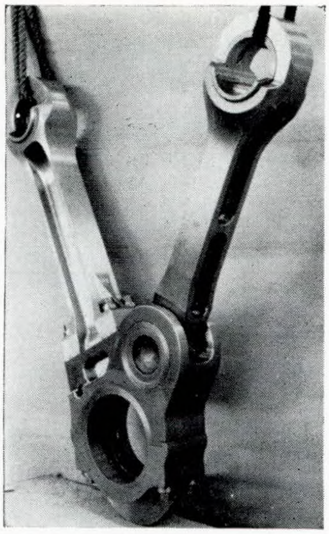

FIG. 7—*Articulated connecting rod—Vee-engine*

length as well as short symmetrical force paths in the frame; they have a steel crown and a light-metal skirt, a design ensuring very little change in the piston crown clearance between part load and full load, excellent cooling of the ring belt; hardened piston ring grooves to increase reliability in service, low wear and good part-load operation, particularly when running on heavy fuel oil (Fig. 8).

The engine has a separate cylinder lubrication system permitting small quantities of fresh lubricating oil of a quality suitable for the fuel used, to be admitted to the exact point where this oil is required. If the lubricating oil system of the entire plant should fail, independent cylinder lubrication would be of advantage. To provide for such a failure the engine main bearings have emergency provisions, but the piston and cylinder running faces would not be protected if they depended solely on the oil thrown up from the crankcase. Thus separate cylinder lubrication which continues to function irrespective of a failure of the forced-feed lubrication system until the engine is stopped, provides an added safety factor. As opposed to the conventional method, the lubricating oil is admitted from below, through

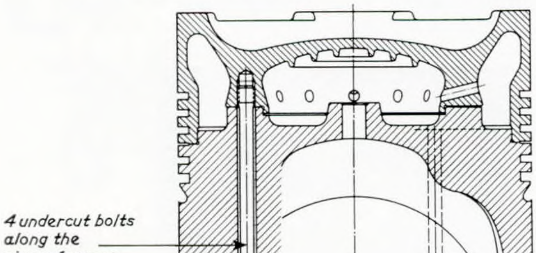

FIG. 8—*Piston—VV40/54 engine*

bores in the cylinder liner, so that it need not reach the hot upper part of the cylinder liner and, consequently, is not in danger of carbonization or separation of additives.

The inlet and exhaust valves are installed in cages, so that they can be replaced without disturbing the cylinder head. This facilitates maintenance work considerably. Intensive cooling of the exhaust valve seats in the cages permits reduction of the valve seat temperatures to a point where, in the case of heavy fuel oil operation, vanadium and sodium deposits can be reduced to a minimum so that the period between valve overhauls is considerably longer (Fig. 9). Cooling the cages instead of the

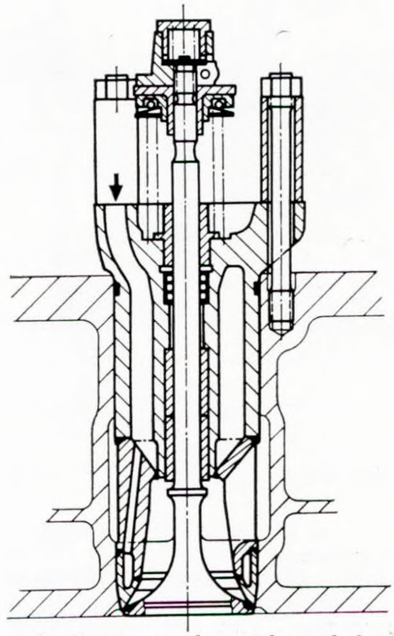

FIG. 9—*Exhaust valve with cooled seat*

valve cones not only results in adequately low temperatures but also avoids unnecessary complications in the supply, treatment and control of the coolant, also risks of lubricating oil dilution and welding on the valve cones. Moreover, the ability to rotate the valves in operation is maintained.

The fuel injection valves and—in the case of periodically or permanently unattended engine rooms—the fuel injection pipes are located in protective casings in which any oil leakage is collected and discharged separately. As there are also water connexions in the lubricating oil space of the valve drive on the cylinder head, the rocker arm lubrication circuit is separate from the general lubricating oil circuit.

Main Design Features of the 52/55 Engine

In May 1969 the first experimental engine of a new and larger type was introduced, a 12-cylinder vee-type V6V 52/55, developing 12 000 hp at 430 rev/min, cylinder bore 20·4 in, stroke 21·6 in. Since then, 35 engines of this type totalling more than 420 000 hp have been sold. The first of these engines, with 18-cylinders, for a containership, was shown fitted with a Renk planetary gearbox on the M.A.N. test bed in November 1970.

There were two main reasons for the design of this new engine. The first was to extend the output range of medium-speed engines to cover unit outputs of between 15 000 and 18 000 hp. The second was the considerable simplification of maintenance work that can be obtained by reducing the number of cylinders for a given engine output.

The design data of the engine were selected after a careful examination of the marketing and technical aspects. To a certain extent, the proven design principles of the 40/54 engine helped to minimize the time for designing and testing the engine and to make it as reliable as possible right from the outset. Nevertheless, full use was made of the latest experience and facilities, permitting the 52/55 engine to be of a considerably more compact and

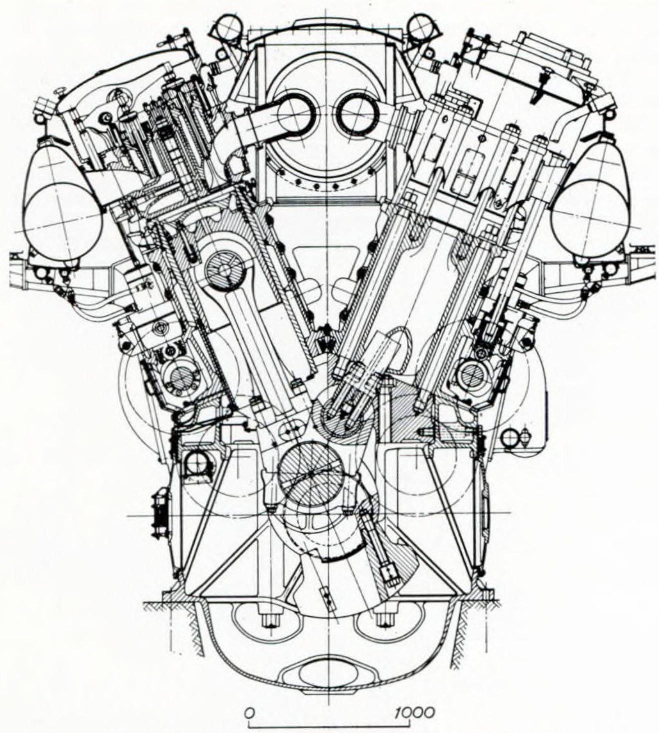

FIG. 10—*Section view of VV52/55 engine*

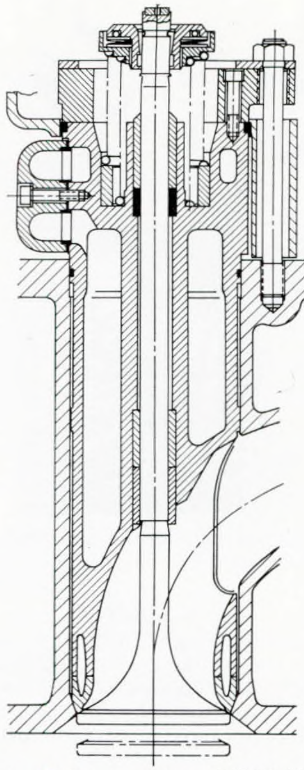

FIG. 11—*Exhaust valve—VV52/55 engine*

lighter design than its predecessor, the concept of which dates back quite a long time.

The cross-section of the engine (Fig. 10) shows basically the same design as that of the 40/54 engine. All proven design features have been incorporated. This also applies to the articulated connecting rod, since calculations showed that the extra design and production work involved is balanced out as far as the engine is concerned by the 12–13 per cent saving in engine length. In a 16-cylinder engine this results in the very considerable saving of exactly one metre in engine length when compared with the design incorporating parallel connecting rods.

The seating face of the cylinder liner resting on the cylinder block is positioned by means of a collar cast integral with the liner, so low that thermal deformation of the upper part of the liner cannot cause ovality. At the same time, the collar permits efficient cooling of the liner, in its hottest zone, so that the temperatures at top piston ring reversal point are of the order of 160°C (320°F).

The whole cylinder head area deviates from the 40/54 design. The head itself is constructed with a considerably higher degree of rigidity and has a relatively thin lower part. The stiffening element, a double ribbed cross, is in the upper area which is not subject to high thermal stresses. The intake and exhaust valves are fitted in cages and the valve drive has been made much simpler, again of benefit as regards maintenance work (Fig. 11). On this engine the fuel and cooling water systems have been completely separated from the rocker arm lubricating oil.

The water inlets and outlets for the exhaust valves are located outside the oil space and the fuel is led to the injection valve from the side through the cylinder head and not from above. This also applies to the coolant piping for the valve.

Scientific Design Methods

In designing new engines, M.A.N. use a number of modern methods to select the optimum design for highly stressed parts, before or during the design stage, and to recognize and eliminate any peak stresses. Thus calculations involving advanced strength analyses were carried out on a computer, permitting the examination of all possible variations within a relatively short period of time. This applies above all to the deformation and stresses of the piston crown, the dynamic loading of the connecting rods and the toothing of the camshaft drive. Photo-elastic examination was also applied in the design of highly stressed areas of the crankshaft, connecting rod, piston crown and piston bolts, frame and exhaust valves (Figs. 12 and 13). The design of the intake and exhaust ports in the cylinder head was based on flow investigations. The design of the bearings was based on theoretical examinations of the dynamic loading and the movements of the bearing pin (Fig. 14). During the testing of the experimental engine, the frame, cylinder head and running gear were strain gauged for stresses.

These methods cannot replace thorough testing, on the test bed and in practical operation, but, to a certain degree,

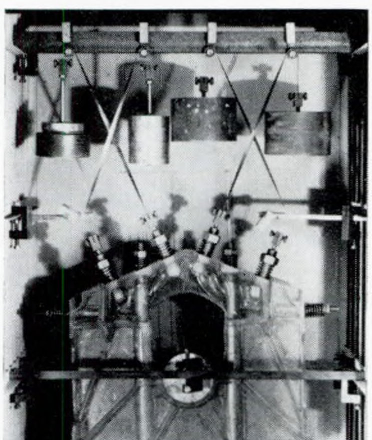

FIG. 12—*Photo-elastic model under load—VV40/54 crankcase assembly*

FIG. 13—*Photo-elastic test of piston crown—VV52/55 engine*

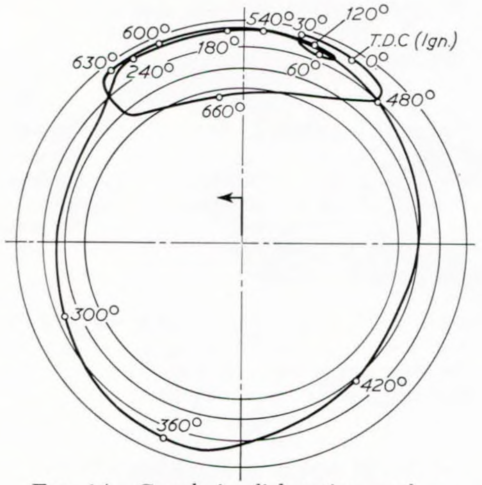

FIG. 14—*Crankpin dislocation path—VV52/55 engine*

reduce the testing time required for many parts and are conducive to additional operating reliability.

Operating Data

The most interesting operating data for the medium-speed engine are the fuel and lubricating oil consumption rates. Fuel consumption is 0·34 lb/bhp h. As is typical for four-stroke engines of higher rating, these rates apply to a very wide output range, roughly between half and full load.

The lubricating oil consumption rate is of particular importance to the economy of medium speed engine plant, where it is normally higher than in a low speed engine or turbine plant. For this reason, it is important that all factors affecting this consumption be carefully matched. An economic evaluation carried out by M.A.N. showed, for instance, that oil consumption of about 0·004 lb/bhp h cancels out the whole advantage a geared plant offers as regards the actual freight costs.

The lubricating oil consumption of the two engine types is 0·0022–0·0027 lb/bhp h which is therefore good. The larger number of engines in service has confirmed this figure for the 40/54 type over a fairly long period.

Exhaust temperature after the valve, at full load, is extremely low at 380°C (715°F) which to a large degree is due to the favourable air rate of 12·6 lb/bhp h. The firing pressure is 1·64 lb/in^2.

Maintenance Problems

In designing these engines, every effort was made to facilitate maintenance and accessibility. The following is a brief description of the most important measures taken.

The Cylinder Head. Electrically heated waisted bolts are used for mounting the cylinder head. These bolts have been employed successfully with the 40/54 engines. A new feature is a unit which makes the heating process much simpler and safer. By means of this all eight heating rods can be connected and controlled simultaneously.

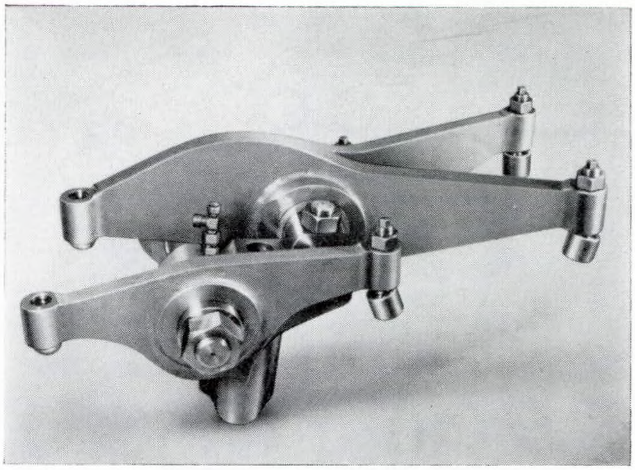

FIG. 15—*Inlet and outlet valve rocker arms—VV52/55 engine*

The valve gear mounted on the cylinder head can be removed as a complete unit after taking off only two nuts (Fig. 15). The inlet and outlet valves are arranged in cages so that the cylinder head need not be disturbed when checking or overhauling them. Cooling water circulation for each outlet valve is outside the oil compartment and by means of flexible hoses. After removing two screws, both cooling water connexions can be taken off without any difficulty. The cooling water inlet and outlet connexions for the exhaust valves can be shut off with a simple manual operation so that the valve can be removed without having to drain off any of the cylinder cooling water. The shut-off point can be monitored without much difficulty.

One of the most important problems associated with improving maintenance conditions for a four-stroke engine is the economical maintenance and repair of the valve seats. To this end two new-style valve grinders have been developed.

The Running Gear. To withdraw the pistons, the connecting rod bolts are removed with the aid of electrical heating rods. Prior to pulling the piston, a sliding piece is clamped to the connecting rod palm end which then, similar to a cam, forces the connecting rod into the required position relative to the cylinder liner so that the running face is protected against damage (Fig. 16). An attachment bolted to the piston crown permits suspension of the piston either vertically or at the angle of inclination of the cylinder. While turning the crankshaft, this attachment can remain fastened to the piston. A screwed in guide rod permits careful installation and removal without much physical effort. To exchange the connecting rod bearing shells it is not necessary to remove the cylinder head or connecting rod. On the other hand, removal of the piston/connecting rod assembly does not disturb the main bearing.

All movement of the main bearing covers is carried out by means of the engine turning gear.

* * *

When considering the maintenance required by various propulsion plants, a comparison prepared by the Swedish shipyard mentioned earlier must be borne in mind (see Fig. 17). Two 12 000 hp propulsion plants with an annual service period of 6000 hours are compared, i.e. one conventional plant with a six-cylinder crosshead two-stroke engine and three seven-cylinder auxiliary engines, and a geared twin-engine plant with two medium speed twelve-cylinder vee-engines as well as two geared generators and a port-duty Diesel. There is not much difference in the maintenance required by the two plants; the medium speed engine plant requires about eight per cent more. The customer for whom the exercise was carried out selected the medium speed engine plant and ordered 13 ships of this type, which have been in service now for some years with good results. They are bulk carriers of the *Malmanger* class already

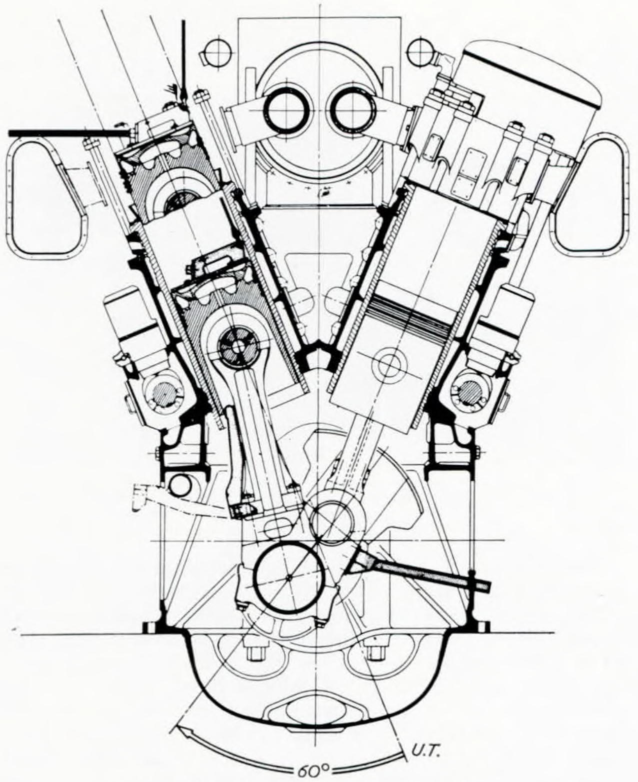

FIG. 16—*Removal of main piston*

mentioned. In future it will be possible to have a second variation with only half the number of cylinders, as 52/55 engines are now available, requiring far less maintenance.

Silencing

The problem of engine noise is often raised with medium speed engines. Here the turbocharging system is the main source of noise. In the M.A.N. engine effective sound-damping treatment has been applied at three points. If the combustion air, as is usually the case, is drawn from the engine room, the engine can be fitted with particularly effective intake noise silencers. In addition, the volute casing of the turbocharger has been given acoustic treatment. Furthermore, absorption type silencers can be mounted at the critical points in the charge-air pipes which were determined during tests. By these measures it has been

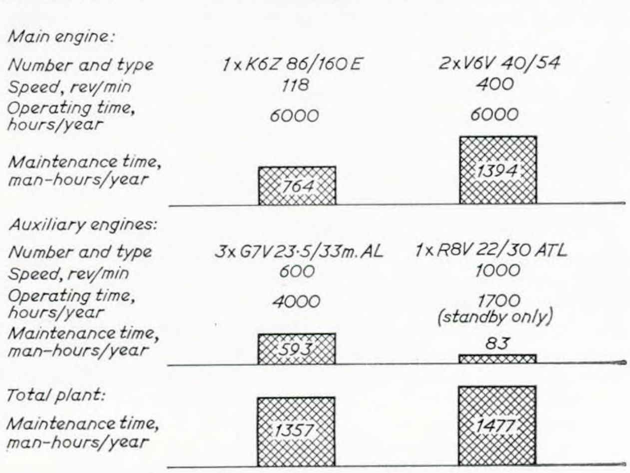

FIG. 17—*Maintenance requirements of two marine engine plants*

possible to reduce the turbocharger noise to the level of that of the engine. Consequently, the total noise level of each type of engine, including the new one, is below 105 dB (A), measured from a distance of three feet.

If special demands are made on these engines, they can also be seated flexibly on their foundations, despite the high power outputs. This results in good dampening of structure-borne noise, i.e. hardly any of the engine vibration is transmitted to the hull. As an example the four 40/54 nine-cylinder engines of the Dutch ferry boat *Koningin Juliana* were mounted on rubber elements, each of the four units weighing a little less than 100 tons which had to be so supported. In this way it was possible to reduce the structure-borne noise by almost 30 dB (A).

OVERHAUL PERIODS, SERVICE LIFE AND RELIABILITY

Table III gives particulars of the overhaul periods and the

TABLE III

	Period between overhauls (hours)	Service life (hours)
Pistons	10 000–12 000	30 000–50 000
First Piston ring		10 000–12 000
Cylinder liners	12 000 (random check)	30 000–60 000
Bearings	6 000 (random check)	approx. 30 000
Outlet valves	4000–7000	12 000–15 000
Injection valves	2000–2500	15 000–20 000
Exhaust turbochargers	12 000	

overall service life of engines of this type based on actual experience and wear rates measured. The figures refer to heavy fuel oil operation and are subject to variation within the limits indicated, depending on the fuel oil quality used.

Reliability

The reliability of engines with a mean effective pressure of 240–255 lb/in² is still sometimes looked upon with scepticism. Doubts of this nature are probably due to experience gained with engines originally designed for lower mean effective pressures and that were turbocharged more and more highly over the years. The engines described here were designed for these, and even higher, degrees of turbocharging from the very outset. The test engines alone have been operated for more than 20 000 hours at 255 lb/in² m.e.p. and, at times, even at 300 lb/in² m.e.p. In addition, the experience gained in marine operation with m.e.p. of between 240 and 255 lb/in² must be considered. The components of the new engine have been designed for an even higher mean effective pressure and the usual safety factors have been allowed for.

The loads and stresses resulting from the high pressures involved can and must be kept within reasonable limits by the engine design. As is well known, the mechanical loads imposed on the parts forming the combustion chamber are small when compared with the thermal loads caused by the irregular temperature distribution. These thermal loads, which are due to high degrees of turbocharging, must also be absorbed by an appropriate design, especially since, in large engines, the thermal loads, as a rule, cause more difficulties than the mechanical. The fact that the design in question has been successful in this respect is shown by the temperatures measured on the important components during operation. Thus, for instance, the temperature in the top piston ring groove is only slightly above 100°C (212°F). This is in the region of the value measured on large two-stroke units. Only 400°C (752°F) were measured on the armoured seat of the outlet valve. The temperatures measured at other points on piston, cylinder liner and cylinder head are extremely low when compared with those of larger two-stroke engines, so that one is quite justified in assuming high reliability and longer life between overhauls when compared with slow speed engines.

Owing to the steady increases in size, a modern ship represents an extremely valuable proposition. It is in this context that particular importance attaches to the reliability of the propulsion plant.

Discussion

Mr. A. N. S. Burnett, M.I.Mar.E., opened by complimenting the author on his paper and on the engines produced by his company, and went on to say that if shipowners would only carry out their marketing, planning and execution of ideas as well as the medium speed Diesel engine manufacturers, the shipping industry would be more healthy today.

Every new piece of machinery should be examined in the light of what the customer wanted to buy and not just as another more expensive toy, using perhaps further unproven new techniques. A shipowner looking for new ships of whatever size or type would like to see the following features incorporated: the smallest machinery space and the largest volume available for cargo per deadweight; the lowest cost per brake horsepower of the machinery plant; a simple machinery installation enabling completion of the ship in the shortest time; the simplest plant to operate, including ease of automation; high utilization of all machinery on board; excellent machinery reliability; improved ship availability and safety; ease of maintenance, with cheap and readily available spare parts; low maintenance costs.

Taking first the machinery space, there was no doubt that on this score alone most shipowners would wish to use a medium speed Diesel engine installation. The advantages of layout, as shown in the paper, were numerous and, perhaps above all, there was the importance of flexibility and choice of propeller diameter and speed.

The author had shown that there was a negligible difference in price per horsepower for a medium speed Diesel engine plant, compared to a low speed Diesel or steam plant. His company had already shown that these installations were simple for cargo ships, enabling shipbuilders to profit also from this factor. What they had not indicated was whether they were going to provide a total package, including modules for all the necessary auxiliaries, thus making it even more simple for the shipowner and the shipbuilder, for all sizes of ship, and more economical for the whole plant.

As far as ship operation was concerned, the main ingredients of automation (the use of improved measurement, observation and control systems) were simple to incorporate with multi-engined medium speed Diesel installations. The ship could leave harbour within minutes from cold, and, if a c.p.p. were fitted, ship manoeuvrability in restricted waterways and general operation were simple, either from the machinery control room or from the bridge. With a multi-engined plant the ship safety and availability throughout the year was superior even to that of a steamship—one of the main advantages put forward for steam installations. Mr. Burnett had had the privilege of operating a multi-engined medium speed Diesel installation, and had had experience of all these advantages. He had personally achieved full speed in 20 minutes in such a ship from an unprepared start in an open anchorage, and had operated a twin-shafted ship on one shaft in order to carry out maintenance to engines connected to the other shaft as well as to achieve economy in fuel at sea.

One of the most interesting and valuable parts of the paper outlined how easily electrical power could be supplied by either gearbox or engine drives, eliminating from the plant design the need for extra independent alternators with their prime movers. Every DM spent on any part of the machinery installation must be fully justified in service. Having pump and other auxiliary drives, or spare alternators lying idle at sea was poor utilization of capital. The development of this intriguing engine and gearbox auxiliary drive was something no shipowner could ignore.

Turning to maintenance, Mr. Burnett said that it was in this area that the real practical advances had been made. How nice it was to see design for maintenance being really practised instead of just talked about. No longer were exhaust valves awkward to remove. New cooling techniques for the exhaust valve assembly had been evolved, thus reducing valve seat deposits and lengthening the valve overhaul interval. Fuel oil leaks were not allowed to impinge on hot engine parts—important in unattended engine rooms. The methods incorporated for both piston and bearing removal were ingenious and simple.

The author had indicated in the paper that even the total maintenance load for similar ships engined with slow speed or multi-engined medium speed Diesel engines absorbed about the same manpower. He had, however, made no reference to availability and cost of spare parts. Could some further information be given on this?

Another untried method in the marine industry, but extensively used in the aero industry or in rail traction, was the use of microfilms for the maintenance task. This simplified the procedure both at head office and on board ship. Numerous advantages came from this, one of them being the ease of incorporating engine modifications into shipowners' systems, and for keeping licensees, builders and engine repair yards up to date with maintenance techniques and spare parts data.

Another device used by the aero industry and rail traction was critical path techniques for the overhaul of the engine. In designing a new engine it was important to incorporate in the design not only ease of maintenance but the maintenance methods to be employed in service; critical path techniques were a very important maintenance tool. Would the author say whether his company had developed a critical path method for maintenance which could be distributed to licensees and to repair yards for these new engines?

Noise and lubricating oil consumption were matters of vital importance to all users. There was no doubt that increased engine ratings meant increased noise, particularly at cylinder head level. It was also well known that flexible engine mountings and acoustic hoods could reduce noise levels, but involved some additional expense. A lot more work needed to be done in this area. Would the author care to explain how he hoped to reduce the noise level of his engine?

In the paper, the lubricating oil consumption of the 52/55 engine had not been confirmed. It was just stated that it was at an acceptable level. Would the author deal with this aspect?

There was no doubt that a great step forward had been achieved and that shipowners and ship designers would be forced to take note of the considerable advantages of such an engine. Multi-engined medium speed Diesel installations had come to stay. How refreshing it was to hear a manufacturer telling the industry exactly what he was doing and almost exactly how he was achieving advancement in design—and no doubt in profits. If shipowners could only come out into the open more often with information about their achievements, difficulties and methods, they would then have to return to their offices, as Mr. Luther would, to sharpen their pencils and be certain that they were equal to or better than their competitors next year as well as this. How refreshing that would be to the marine industry at large.

Mr. P. J. Adolph, A.M.I.Mar.E., emphasizing that the opinions given were his own and not necessarily those of his company, said that in the section on design features of the 40/54 engine the author had briefly outlined the separate cylinder lubrication and indicated that provision had been made to supply the main bearings in the event of lubricating oil failure. Would he give a brief description of this supply arrangement and also indicate whether it had been incorporated on the 52/55 engine?

Going back to the separate cylinder lubrication, the author had stated that "the oil is admitted from below so that it need not reach the hot upper part of the cylinder liner". At first glance this appeared to be a slightly strange statement, but by the "hot upper part" he took it the author meant the actual combustion zone, and with the top ring set fairly well down on the piston and since, on the 52/55 engine, the cylinder oil, he believed, was injected so that when the piston was on b.d.c. the oil entered between the second and third rings, on this basis the design would tend to give a slightly dry top ring, and thus justify the author's statement. It was noted that a scraper ring was fitted just below the compression rings, but the absense of a lower scraper ring on the piston skirt was noted with interest.

With the fitting of external cylinder lubrication—where the feed rate could be much better controlled than when lubrication was via the connecting rod and gudgeon pin—once again the possibility existed of using the optimum lubricating oil for the task involved. On this type of engine, on heavy fuel operation, the requirements on crankcase lubrication could generally be best met by an SAE 30 Supp. 1 oil with a TBN of between 5–10, while the cylinder oil requirements tended to dictate an SAE 40 oil of between Series 2 and Series 3 detergent level with a TBN between 25–40. However, all too frequently the shipowner called for a dual purpose oil, generally a compromise of an SAE 30 with a TBN 20–25, which clearly did not fulfil the ideal requirements of either the crankcase or the cylinders. Bearing in mind that a mid-alkaline dual purpose oil was somewhat more expensive than a low alkaline Supp. 1 oil, whether the saving on a storage tank and the fact that a larger bulk delivery of one grade of oil only was required, justified this technical compromise was a very debatable point, and he would very much appreciate the author's views on this subject.

Mr. G. B. Fraser, M.I.Mar.E., said that, contrary to the indications in the paper, there had been a certain amount of trouble experienced with medium speed engines and some difficulty in their application in ships.

The noise level in the engine rooms was so high in some cases that extra instrumentation was necessary because the operating engineers could no longer hear. Would the author care to comment on the amount of instrumentation required on these engines due to the noise level and increased speed?

The point was made that maintenance could be carried out on one engine while the other engine, or engines, were running. There again, the noise level and also the temperature were relevant. Did the author seriously consider this to be a practical proposition under the conditions that would prevail in the type of engine room described? Standard ear muffs, for instance, tended to fill up with perspiration very quickly and were obviously not designed for this type of use.

With medium speed engines now considered to be a regular feature on ocean-going vessels, it would be useful to have some indication of what "tropicalization" margins were considered to be practical for this type of engine.

Also, some trouble had been experienced with load control systems. Had the author any experience of how they were being set up, how the flexible coupling, clutch, gearbox and shafting losses were allowed for in these systems and also, the effect of propeller efficiencies? What were his views on the later maintenance and calibration of the load control system considering that neither indicated nor brake horsepower were at present measurable on these engines?

If given a choice, what type of propeller would the author choose with this type of engine installation—fixed pitch, c.p.p. variable speed, or c.p.p. constant speed?

As these engines were situated much lower down in the ship, how had the problem of ventilation and the air supply to the engines been overcome?

Mr. P. J. G. Mack, M.I.Mar.E., said that the paper was concerned principally with the advantages of the medium speed engine, with particular reference to the design features of the engines built by the author's company. However, in his opinion many of the desirable features postulated were incompatible and in fact unobtainable. For instance, it was not possible to have simplicity in engine design and pack in the power. As they developed, these families of engines became much more complicated, to the extent that the maintenance and operational requirements became more exacting. This was exemplified, for instance, by the need to resort to special arrangements—assorted pre-stressing techniques for bolted assemblies, special lubricants, micro-mesh filters and, he now noted, not without surprise, an emergency arrangement to deal with failure of the crankcase lubricant. Despite the present day "fashionable and rational" approach to design, the fact of the matter was that these "provisions" were mainly dictated by reduced safety margins. In the case of a thin walled bearing shell, relief from overheating by wiping no longer obtained and failure of such a bearing might sometimes mean renewal of the crankshaft.

Since 1966, the number of operational medium speed engined passenger/vehicular ferries had substantially increased and all nine passenger ships, at present building for British registry, were to be driven by medium speed engines with controllable pitch propellers. This transition had not been without operational defects and his personal comments, based on shipping casualty investigations and floor plate survey experience, were intended both as a caution to owners' superintendents and as a plea to manufacturers to improve the necessary reliability and safety.

The post-war development of steam and Diesel plant had had one common factor, which was the continued increase of fuel ratings, coupled with wide range remote control of plant output. The extent of the associated problems was common to both Diesel plant and steam plant, and, oddly enough, some of the major problems seemed to be associated with instability at low load conditions. In modern roof-fired boiler plant, for instance, the fouling of heat transfer surfaces, the burning out of economizers and superheaters, and the penal sanctions of the Clean Air Acts were all principally light-load operational problems which, even now, did not appear to have been fully resolved.

The medium speed engine likewise was not without its shortcomings and these were recognized by the author when he referred to the possible limitation of heavy fuel at low load, and again when the benefits of the composite piston designed were discussed.

If a high powered Diesel engine were to be coupled to a controllable pitch propeller, the requirements of the fuel injection arrangements would be more exacting than those associated with the same engine driving a fixed pitch propeller, particularly in the case of a ship which was required to manoeuvre frequently. In the former case, the machinery might be required to operate for protracted periods at idling speed (in the event of fog etc) and to respond from this condition when relatively cold to a full fuel rack displacement at the command of the bridge combinator controls, generally operated by someone with no comprehension of the limitation of machinery. In the idling condition it was desirable that all cylinder fired uniformly without "coking or trumpeting" of injectors and so assist the subsequent engine response to load and acceleration of the turboblowers. Variations of fuel pump performance at low loads might contribute to fouled injectors and, when coupled with fuel rack and propeller pitch displacement rates in excess of the turboblower response, would produce a transient condition of poor combustion and after-burning. This reaction was magnified when manoeuvring from ahead to astern or astern to ahead with ship way in either direction and manifested itself in many cases with varying hues of smoke at the funnel. It was a transient condition but very important in some cases. Balancing of these engines at power was generally done by adjusting the fuel pump rack positions against exhaust temperatures and maximum cylinder pressures at the expense, perhaps, of the performance of the fuel pumps at idling speed. The fuel pumps and injectors for each engine should, therefore, be matched and calibrated throughout the full rack range and, with two-stroke cycle engines particularly, it might be necessary to sacrifice power balance to achieve satisfactory low load operation.

There would also seem to be ample scope and a need to design the bridge combinator controls to match the engine characteristics and to cater for normal service deterioration of the engine components; for instance, a falling off in turbo-blower efficiency. The previous speaker had mentioned ventilation in the engine room, this, from the point of view of engine aspiration, being equally important. The provision of boost pressure attenuation of the pitch/fuel rack displacement rates might be a necessary refinement to embody in future control systems. Perhaps the author would care to comment on this.

Aluminium alloy trunk pistons in comparison, for example, with cast iron pistons of equivalent size, when operating under favourable conditions, appeared to be predisposed to scoring of the skirt thrust faces and distortion. This defect was usually attributed to inadequate lubrication associated with the design and condition of the oil scraper/control rings and possibly with the effectiveness of splash lubrication at idling speeds. The aluminium alloy trunk piston, however, had a high thermal conductivity, a high piston/cylinder liner thermal expansion ratio, an unsymmetrical cross-section, and was made of relatively low melting point material susceptible to tearing and the possible production of incendive sparks—it was not many years since the use of magnesium anodes in tankers was rejected for the latter reason. Given these physical characteristics and a rapid increase in heat flux during manoeuvring with cold pistons, coupled with inefficient combustion, the integrity of the boundary layer of cylinder lubricant and the piston ring seal might be impaired, giving rise to local hot spots and tearing of the skirts. The author appeared to be admitting to this vulnerability in terms of his comments on cylinder lubricators. Actually, in the event of total failure of the main lubrication, a trailing engine might be stopped within seconds by bringing the propeller pitch from the ahead to the astern position.

Unfortunately, however, the onset of piston seizure might also only take seconds, during which time the rapid generation of oil mist would take place, with the attendant risk of a crankcase explosion. In such circumstances, where the piston coolant was designed to be returned to the crankcase via an open ended orifice on the underside of the piston head, it appeared that, on account of the liberal quantity of lubricating oil exposed to the overheated piston, a rapid generation of oil mist in the over-rich phase might take place and that not until the engine was stopped did ignition occur. In such cases, the explosion might be very violent and it was essential that the gross areas of the crankcase relief valves were considered in relation to the strength of the crankcase doors in addition to the gross volume of the crankcase. There had been cases of crankcase explosions from such engines which were provided with relief areas in excess of existing requirements, where the relief arrangements were inadequate because the crankcase doors failed due to combined buckling of the plating and shearing of the fastenings. In such cases an oil mist detector might not give adequate warning. What were the author's views on this aspect of piston design; were subdivision flame screens contemplated in the crankcases of the new engines and why was monitoring of piston coolant return temperatures not considered necessary?

The foregoing comments were relevant to recent experiences in many different types of medium speed engine engaged in the exacting operating conditions associated with the U.K./Continental passenger/vehicular ferry services. The shortcomings referred to had been associated with scavenge fires, uptake explosions, crankcase explosions, excessive component wear rates, relatively short-term thermal fatigue fractures and even the inability of engines to accelerate from idling to load.

Given the option of an engine truly designed and built for the marine environment, was this the price shipowners were prepared to pay for having a gallon in a pint pot in pursuit of paper economies?

He made no apology for painting such a black picture because in so doing one might finish up with a light shade of grey.

Mr. D. Royle, B.Sc., A.M.I.Mar.E., said that the author had quoted wear measurements related to the 40/54 engine showing a difference between average values and those obtained on a particular ship using a fuel of 1500 sec Redwood No. 1 at 100°F. Photographs of the pistons would probably have shown their good condition after long periods between overhaul.

His company had recently inspected the 40/54 engines in a ferry after 10 000 hours operation. The fuel used was of 200 sec Redwood No. 1 at 100°F, but the voyages from port to port were of only three hours duration, so the engines were subjected to almost continual load variation, with a high proportion of part load operation. There were also frequent stops.

The maximum liner wear rate was 0·01 mm/1000 h—the average shown in the paper—and the top piston ring radial wear was 0·05 mm/1000 h. The higher than average piston ring radial wear was probably accounted for by the type of operation of the vessel. Pistons taken from 40/54 engines were invariably very clean and probably the use of separate cylinder lubrication was a contributing factor.

The author was very wise to make some remarks concerning the fuel and lubricating oil consumption rates for medium speed engines.

The lubricating oil consumption rate of 1·0 to 1·2 g/bhp h was considered good, whereas if the consumption rate rose to 1·8 g/bhp h, it would cancel out the whole advantage offered by a geared plant in regard to the actual freight costs.

Engine builders quite rightly talked about the lubricating oil consumption rate of an engine, whereas the shipowner was more concerned in the lubricating oil consumption rate of the vessel. The difference between these two figures was accounted for by losses in the use of automatic centrifuges and automatic filters, both necessary items with the lean engine room manning which applied in most ships.

In practice it would seem that many shipowners had overall lubricating oil consumption rates very near, if not above, 1·8 bhp h, but it would be extremely useful to hear some more comment on this aspect both from the author and the shipowners operating medium speed engine installations.

Mr. G. Geddes, M.I.Mar.E., referred to load control, which he regarded as important. Their experience of various ships with medium speed Diesels was that there was no means of checking the horsepower developed, in any way at all other than by adjusting the cylinders to suit exhaust gas temperatures. This was done by adjusting fuel pump racks when they got out of balance or phase with the main control for the racks and this movement of the main rack controlled the load controller of the propeller, together with a shaft rev/min indicator which controlled the constant speed of the engine to suit shaft alternators, when fitted. Could the author say how this system was controllable to prevent overload "coming back" on to the engine in any way at all, bearing in mind that a controllable pitch propeller was a standard manufactured propeller and could put up to 150 per cent or more "back on to the engine" if the pitch was not properly controlled?

Mr. G. R. Pringle, M.I.Mar.E., in a contribution read by Mr. A. E. Franklin, M.I.Mar.E., said that the author had given an excellent summary of the advantages of the twin medium speed geared Diesel drive on board ship. He acknowledged the contribution to this development made by a number of sections of the industry, including lubrication engineers, gearing designers and controllable pitch propeller manufacturers. However, the arrangement would not have been accepted unless uses were identified for it. One such use was in the coastal type of tanker where short port times were important for the economic viability of the vessel and, therefore, high pumping horsepowers were required. Although these pumping powers were high, compared with other auxiliary requirements, they were low compared with propulsion requirements. In fact, the requirement had generally been below the minimum power available when one main engine had been installed and, therefore, it had not been possible to run this engine satisfactorily in port for pumping only. The twin engine arrangement, on the other hand, permitted one engine to be run at an acceptably higher proportion of its power. His company had four tankers over 5000 dwt working with this system and a further two were being built. Experience in these ships had shown the need to fit a shaft brake, to prevent undesirable rotation of the c.p. propeller when the propeller was feathered, caused by drag in the clutches.

With the coming of the 1000 bhp/c engine, of which the 52/55 model was such an attractive example, they could look forward to an increasing number of larger twin engine installations. Such installations would be particularly suitable for the replacement of the 20 000–40 000 dwt tankers presently engaged in product movements and crude trans-shipment, but which were originally built in the 50s as crude carriers and were not ideally suited for their present employment.

Correspondence

Mr. R. M. Dunshea, M.I.Mar.E., in a written contribution, stated that the case for the medium speed engine had by now been convincingly established. From his experience, when compared with slow speed engines installed in ships as recently as seven years ago, medium speed engines could be less prone to faults than large two-stroke crosshead engines.

Particular reference could be made to crosshead bearing failures, and the heavy and exacting work involved in rectifying them, scavenge fires and their attendant dangers, heavy fouling of the gas side of turbochargers with resultant loss of power and burnt out exhaust valves often after periods of service of a little more than 1000 hours. He had experienced none of these troubles with medium speed four-stroke engines using residual fuel of a quality comparable with that used in slow speed engines.

The medium speed engine required a high standard of maintenance and cleanliness but given these, was extremely reliable.

It was felt, however, that the number of cylinders for main propulsion should not exceed 24 if possible and that 36 would be the greatest number. Above this a maintenance load of excessive proportions would be imposed on the average engine room staff.

Accessibility could be unsatisfactory in medium speed engines. It was pleasing to know that improvements in this respect had been sought in recent engines.

It was noted that in the author's engines heating elements were used for various tensioning functions. Could the author give an idea of the time after heating that cylinder head nuts made correct pressure contact with the head? An indication of the procedure, so far as the nut position relative to the cylinder head at the time elements were withdrawn, would be appreciated. With ambient conditions varying, to what accuracy could the tension in a stud be obtained? After experimenting with hydraulic tensioning of cylinder heads, Mr. Dunshea had found that a pneumatically operated impact wrench, with accurate hardening angles, was quick and satisfactory.

The author had referred to his engines having separate cylinder lubrication, making the point that an oil for cylinder lubrication could be better selected. In the case of engines operating on residual fuel, where an alkaline crankcase oil was necessary, would the author normally recommend different qualities of oil for cylinder lubrication and the crankcase? Failure of cylinder lubrication could be quickly disastrous in a highly rated engine and splash lubrication was at least a sure method of lubrication. How accurately did the lubricators on the author's engine meter the oil to each cylinder? Was there sufficient splash lubrication to avoid serious overheating should a lubricator to one cylinder fail?

Finally, he would appreciate having particulars of the tightening procedures for the exhaust valve cage studs in the 52/55 engine. How did the stress vary in the cage and the studs between the cold and running conditions?

Mr. J. B. Hill, B.Sc., M.I.Mar.E., wrote that as the author had pointed out, there was nothing new in the application of medium speed engines for main propulsion purposes; indeed his company had been developing this means of propulsion in parallel with their slow speed engines, since before the Second World War.

Particularly interesting early geared installations were those installed in the German pocket battleships and a series of fast cargo-liners, some of which were to prove the reliability of medium speed machinery by remaining at sea for long periods, as commerce raiders.

Post-war interest in geared installations would appear to have been rather inconsistent, various shipowners having favoured geared drive at one time or another, only to revert to direct drive in later tonnage.

It was correct to say that recently there had been an upsurge in interest, particularly in the case of ferries and roll-on/roll-off ships, but the percentage of other motor vessels having geared installations remained comparatively small.

One suspected that shipbuilders and naval architects were largely responsible for the almost universal adoption of geared drive in ferries and similar craft; designs being such that no other type of engine could possibly develop the power required and fit into the space available.

The current interest in medium speed engines for freighters was undoubtedly because of the large horsepowers required, which necessitated the shipowner having to choose between mammoth direct drive engines, medium speed engines and turbine machinery.

Expressing a personal opinion, it was suggested that for horsepowers above 28 000, the most ardent supporter of the internal combustion engine, might be forgiven for taking a serious look at the steam turbine.

One arrived at this conclusion, not because of lack of merit in the design of either the direct, or geared drive Diesel, but simply because of the manpower involved in maintaining huge Diesel engines, or multi-cylinder units.

It had been contended in the paper that a two-year period between piston overhauls might be expected from the latest vee-engines. Past experience would suggest that this hardly seemed possible, but assuming it to be feasible, one had not really gained any advantage over the latest slow speed engine, which, with less than half the number of cylinders, was also said to be capable of running for two years between piston overhauls.

Practical experience with geared installations indicated that piston maintenance of medium speed engines occupied almost as much time per piston, as the overhaul of large bore pistons, whilst attention to inlet and exhaust valves was appreciably more time consuming, as far as the smaller engines were concerned.

In support of this argument, he was able to offer some figures based on a number of years experience with a series of vessels, some gear driven and others fitted with direct drive Diesels of about the same horsepower.

	Direct Drive	*Geared Drive*
Average daily repair costs	£56·00	£94·00
Average daily cost of lubricating oil	£6·00	£15·00

One accepted that many improvements had been made to medium speed engines since those used in this illustration were constructed, but it must also be acknowledged that there had been similar advances in design and efficiency of the slow speed engine.

The lubricating oil costs he had quoted underlined a distinct disadvantage from which the medium speed trunk piston engine had long suffered. Lubricating oil consumption figures given in the paper were reasonable; could they be regarded as average over the life of the engine, or were they test bed figures which could only be maintained by meticulous attention to scraper rings, bearing clearances etc?

Reverting to the first section of the paper one read with interest the many reasons advanced for preferring geared installations to direct drive. Whilst space did not permit examination of more than a few of the arguments, one could not refrain from observing how leading advocates of both geared and direct drive installations had been able to employ much the same information to prove their respective types of machinery were superior to the other.

Weight and space considerations were probably the most common ground upon which the two sides did battle and when one attempted to look at the facts impartially, it was hard to decide which type of installation had the advantage.

The author had pointed out that the weight of the medium speed plant was considerably less than that of other plants, yet other authorities had demonstrated that the all-in weight of a direct drive installation might be only about 25 per cent more than medium speed machinery of corresponding power. Admittedly, this difference could amount to as much as 400 tons in a large installation, but was this figure a really significant factor, in deciding whether a large bulk carrier (say 60 000 dwt) would have to use a graving dock, or a floating dock?

Height, he agreed, was important in certain specialized craft, but not a deciding factor in bulk carriers, tankers and most types of cargo vessel.

Neither type of machinery would appear to have the advantage as far as installed length was concerned, the author's company claiming a reduction of one, or two metres, using an epicyclic gearbox, whilst the direct drive enthusiasts produced typical machinery layouts showing engine room length reductions of a similar order for their machinery.

In reality, one wondered whether the trend towards ever shorter engine rooms was not an embarrassment when it came to finding space for large gearboxes in the narrowest part of the engine room.

The author had laid some stress upon the advantages of being able to drive generators, cargo pumps etc, through the main gearbox. The attractions of adopting this method of auxiliary drive were obvious, but it should be borne in mind that in so doing one put almost all one's eggs in one basket, in so far as the integrity of the gearbox was concerned. There was also the hazard of clutch trouble to contend with. As difficulties in this respect were by no means unknown, could the author give some details of the type of clutch which would be used for the applications illustrated?

The use of main propulsion machinery in port, to drive auxiliaries, made one wonder how time was found for routine maintenance and whether it was realistic, from the noise point of view, to expect staff to carry out overhauling work within a few feet of an engine running at normal speed.

In connexion with manoeuvring, the possibility of running one engine ahead and the other astern was advocated, to facilitate rapid changes in direction of propeller rotation. This proposal had been made many times by manufacturers of geared machinery, but he had yet to meet a chief engineer who favoured the suggestion. When manoeuvring in this way there was an ever present danger of accidentally declutching an engine which was developing full, or nearly full, power. This could result in serious overspeeding of a light medium speed engine, unless the governor was instantaneous in its action, and one wondered if the author's company had developed a governing system which could cater for this possibility of sudden load shedding, or was there a system of interlocks which prevented such an eventuality?

The criticisms of geared drive installations made in this contribution were not in any way intended as direct criticisms of the latest VV engine, which Mr. Hill considered as one of the best of its type available. From the design and maintenance point of view the builders had taken maximum advantage of their very extensive research and test facilities and had evolved some particularly ingenious devices to aid the marine engineer in his maintenance work.

The author's company had expressed hopes of being able to establish the medium speed engine in the field of bulk carrier and tanker propulsion, and obviously much interest would centre upon whether this excellent heavy fuel burning engine could succeed, where others had failed, in ousting the direct drive Diesel from its predominant position.

Author's Reply

Dipl. Ing. Luther, in reply, thanked Mr. Burnett for his remarks, which indicated that M.A.N. were on the right lines with the medium speed engine.

One of Mr. Burnett's questions concerned the possibility of delivering total packages. The company was developing proposals for a package deal through a special group. It was necessary, especially in regard to medium speed engines, that such complete installations could be offered to shipyards and owners.

With regard to availability of service, speaking for his own company and for other medium speed engine producers too, there was a world-wide network of service stations, including the licensed companies, where all parts subject to wear were available. It seemed to be simpler to carry the smaller parts, which were easier to handle and to store aboard ship than in the case of slow speed engines.

The proposal for utilizing microfilms for maintenance duties was very interesting and would be closely examined by the company. Its adoption, of course, would require matched organizations used by both the engine manufacturer and the owner.

The critical path method had already been employed by the author's company for the design, development and construction of new engines, but not for the preparation of maintenance plans. However, the proposal undoubtedly was of interest, especially to the engine operator.

With regard to the counter measures against noise, he had mentioned this in the paper. It depended a little on the make of turboblower. Some turboblowers were noisier than others. The company had made various measurements and had developed insulation and damping for the different parts. The main noise source was the whole air arrangement of the engine, the air suction and delivery piping, not only the blower, but also the cooler and the manifold. These were the points at which the noise damping counter measures had to be applied. This also answered one of Mr. Fraser's questions.

In the case of the new and bigger engine, 52/55, there were the restrictions of the German equivalent of the Board of Trade to be considered. With this in mind a noise level of 103 dBa was attained.

The lubricating oil consumption was given in the paper only for the smaller 40/54 engine, but so far the same consumption had been measured for the bigger type, 1·0 to 1·3 g/bhp h.

With regard to Mr. Royle's contribution, in this same context, there was only one case with the 40/54 where this figure was exceeded, and there were special reasons on the engine for this. In all other cases they had 1·2 to 1·3 g, with medium speed. These figures applied only to the engine consumption itself. He concurred that, apart from the engine there were still a number of opportunities for lubricating oil losses, which Mr. Royle had already ennumerated. Time and again the company had received higher consumption figures from various ships and, after close examination noted, that losses outside of the engine were the cause for excess consumption. In this respect inspection and care by the ship's crew were considerably lacking in some instances.

Mr. Adolph had asked about cylinder lubrication. There must be a misunderstanding in this case. The author had stated that, in the event of failure of the oil pressure supply to the engine and thus the engine bearings, a separate cylinder lubrication system driven by the engine served as an additional safety factor, so that at least the pistons would be supplied with lubricating oil whilst the vessel was slowing down. From Fig. 10 it could be seen that the cylinder lubricating oil came in from the bottom, through the liner wall. The oil inlet in the liner was not between the second and third, but between the first and second rings and, in the 40/54 and the bigger engine, they had not found too great an amount of wear on the first piston ring or any signs of too little lubrication.

With regard to the type of lubricating oil, they were using the so-called mild alkaline oil, not the high alkaline oil, with a TBN of 20/25. It had been found, together with some oil companies, in tests on the company's test bed and in longer tests in several ships, that there were advantages regarding ash deposit and in connexion with lower wear if this oil was used.

Mr. Fraser had asked about noise. A very interesting comparison had been made on the M.A.N. test bed. The noise of the new medium speed engine and that of the new crosshead engine were measured, and in the vicinity of the blower, the noise was exactly the same in each case. With the crosshead engine the noise source at the engine top was much closer to the crew or passenger accommodation than with the low height, medium speed engine.

The idea of overhauling the engine while one or several other engines were still in operation in the engine room was not particularly his company's, as engine manufacturers. They did know some large shipping companies considering the inclusion of such an arrangement into the operational timetable of their vessels, at the very outset.

With regard to precautions in service under tropical conditions, normally the output of the M.A.N. engines did not have to be reduced, because the air cooler was already designed for these conditions. Attention had to be given to the temperature of the cooling water for the air coolers. It was well known that, with moderately high humidity, together with very low cooling water temperatures, the amount of condensing water in the air ducts could be too high and could cause difficulties, and although the air temperature had to be maintained at a certain level and had not to be too low, this was an important point in tropical conditions.

Load distribution, according to his company's experience, did not represent a problem for multiple engine plants. In such cases they were using speed governors set precisely and evenly to the speed drop and the speed set-point. Of course, there would be more costly and complicated solutions. However, the arrangement just referred to had always proved to be satisfactory for the large number of twin-engine plants equipped with their engines.

With regard to the question concerning a variable or fixed pitch propeller, the engine builder would seem to indicate a preference for the variable pitch propeller, but it was not absolutely necessary because the engine was reversible; in the case of multi-engined plant it was advantageous to have a variable pitch propeller because it was easier to adjust the output of one engine only in relation to the propeller conditions.

As to ventilation of the engine room, this was no different in the medium speed engine from the other engines. As far as he knew—and this applied to a large number of plants equipped with their engines, as well as with those of other makes—there were no problems with engine room ventilation in medium speed plants. They had always placed great emphasis on leading the fresh air supply close to the engine turbochargers by means of ventilation ducts. Needless to say, these ducts would not be arranged too close to the turbocharger intakes, since instances had happened where, in heavy seas, sea water was also sucked in by the engines.

The engine builders were given the specifications and to date there had been no claims from the shipyards, in this connexion.

Mr. Mack had made a number of criticisms. He did not know which was the bad engine of which Mr. Mack had had experience and felt sure it was not a M.A.N. product. Certainly he did not think it could be of the newer type, because some points mentioned by Mr. Mack could not relate to these engines. There had been difficulties with several ships of British Rail, but he could compare with this the very good results with a Channel ferry ship with medium speed engines of the type 40/54, on nearly the same route. But there was a big difference between these two engine types. The former were medium speed two-strokes and the latter medium speed four-stroke engines and there must be some reason why most of the medium speed manufacturers had chosen the four-stroke cycle.

Medium speed and trunk piston engines were not so insensitive to any kind of heavy fuel as the big slow running, two-stroke engines. On the other hand, according to the oil companies, as much as 90 per cent of big two-stroke slow speed engines were running on fuels of not more than 1500 sec Redwood, and here the conditions were nearly equal.

Mr. Mack had also brought up the subject of prolonged operation at low loads and speeds. Under such operational conditions the injection pump with two plunger sizes, used for the 40/54 engines proved very satisfactory. For instance, in the case of the *Malmanger* class bulk carriers, the power for the Munk loaders required in port was supplied by the main engines through gear-generators, with the power requirement amounting to only one-sixth of full engine performance. Beyond that, for the larger medium speed engine, they had now developed an injection pump with only one plunger, by means of which, due to a special design in the injection system following it, excellent conditions could also be attained at low loads and speeds.

Engines had been delivered for 13 bulk carriers—26 engines—and all these bulk carriers had generators and were using them, in most cases, in harbour for loading and unloading. This meant that the engines were running with mean effective pressure of 3–3½ kg/cm². Depending on harbour conditions, this could last from 24 to 48 hours, sometimes with heavy fuel, but normally with Diesel fuel. The results were good. Maybe one reason was that here they used, on the 40/54, the double plunger injection pump, which delivered an exact amount even with very small loads and very small amounts of fuel.

The problem of a drop in turbocharger performance primarily occurred when using low grade fuels and was quite well known to his company. Water washing the turbocharger on the turbine side represented an effective remedy for this. In turbochargers of different makes this practice had already been followed for some time. It was very important that, for this washing equipment, all requirements regarding operational safety, such as self-cleaning due to carbonization, should be taken into account. In this respect they had gathered a lot of experience in the past.

The aluminium piston, of all sizes, nowadays represented an element, which had proven itself over decades. Piston seizures—irrespective of causes—were possible, whatever material they were made of, and the resulting damage was more or less equally serious. Their build-up piston had the advantages already mentioned, as compared to the pure aluminium piston; among others there was primarily the advantage of considerably lower running surface temperatures, as well as the fact that difficulties in the ring area—whatever the nature—would not cause an undesirable temperature increase in the aluminium piston skirt.

In principle their engine crankcases were divided into sections. However, they were provided with fairly small recesses, necessary to accommodate the crankcase bearing covers. The company was, of course, fitting crankcase doors of sufficient strength, with relief valves approved by the classification societies, and screens for deflecting the flames to the bottom.

He thanked Mr. Royle for his comments, especially in the context of lubricating oil consumption and cylinder lubrication. As already mentioned, 1·8 g was an exception in M.A.N. engines.

Mr. Geddes had asked about measuring output, and he could confirm that in the case of higher speed engines it was difficult, if not nearly impossible, to draw up a diagram, and the output could only be checked by means of fuel admission, exhaust gas temperature, blower pressure and blower speed. The overload could be locked and it was impossible then to employ higher loads.

When using variable pitch propellers a protection device against overload was provided for all modern control systems by the manufacturers of such propellers. The propeller pitch would be controlled in dependence on the engine load and, when the engine was overloaded, the propeller pitch would be reduced correspondingly.

He was grateful for the interesting contribution from Mr. Pringle. It was very useful for the engine operator to know about the special conditions in the smaller tankers.

As mentioned already, there were definite possibilities for the modern medium speed engines to drive generators for loading and unloading in port. This applied, to an increasing degree, to tanker plants, since in this case the rating required for the cargo oil pumps compared favourably with the overall engine rating.

Mr. Dunshea's remarks, likewise, were of great interest. Regarding the questions in connexion with the heating bars for the cylinder head bolts, first the cylinder head nuts were tightened by hand and without using a spanner. Then the bolts were heated by the heating bars and after a certain time—set on an electric timing clock on the terminal box—had elapsed, the current turned off automatically. Then the nuts were turned through a specified angle marked on the face of the cylinder head to the prescribed degree, and after a cooling period of approximately 30 minutes, the cylinder head bolts were fully prestressed. During this period, of course, all the necessary connexions to the cylinder head could be made. Undoubtedly, an hydraulic or pneumatic prestressing device for these bolts would, in principle, be feasible. However, these devices had certain disadvantages compared with the electric method, since, in view of the size of the bolts, it would be relatively heavy and require quite some space.

The author admitted that in most cases on board a ship emphasis was placed on using only a single type of lubricating oil for cylinder lubrication and the gear. The metering and distribution of the cylinder oil quantity for the individual lubricating points was precise, since each lubricating point was supplied by a separate small plunger in the oil pump. Moreover, especially in the case of the medium-speed engines, the injection oil quantity supplied from the drive gear was ample since, due to the low speed required for the fixed pitch propeller installation, additional injection nozzles were used.

The heat expansion of the exhaust valve cages, which was not inconsiderable, would be absorbed by long and elastic expansion bolts designed especially for this purpose.

The author thanked Mr. Hill for his very detailed contribution, which mainly related the doubts of an advocate of low speed engines. Mr. Luther's comments were not meant to demonstrate the superiority of the medium speed engine in every field, but to show the reasons why it had become more and more popular in shipping and probably would continue to do so. Of course, for both types of propulsion system as well as for the steam and gas turbine, advantages and disadvantages could be ennumerated, which very largely depended on the intended use, i.e. on the type of the vessel, shipping route and many other factors. This was not the least significant reason, why his company had included both engine types in its production programme and continued to develop them.

The wear values with their built-up piston design, dependent on the fuel quality, made a piston overhaul period of two years quite feasible, if only because of the long life of the hardened piston ring grooves. Even if the medium-speed plant were only equal in this respect there were still other points in its favour.

It was certainly correct, if Mr. Hill was saying that, since his inquiries regarding repair costs, progress had been made. However, the author pointed out that in his presentation neither the higher lubricating oil consumption, nor the higher maintenance costs were questioned. The former could not be denied, yet could be kept within reasonable limits by appropriate and constructive measures. As clearly stated in the presentation, the latter could be put on to a similar basis by fully utilizing the geared plant, as with a low speed plant. In addition, attention was drawn to that part of the presentation which dealt with the operating figures of the M.A.N. medium speed engines. Here it was demonstrated, that engines of the 40/54 type in actual service showed the quoted lubricating oil consumption values of 1·0 to 1·3 g/bhp h over a longer operating period. This still applied even at present, after operating periods of about or more than 20 000 hours had been recorded for several twin-engine plants.

He did not know which medium speed plants Mr. Hill had used for his weight comparison, showing these to be lighter by only 20 per cent. All the comparisons, which his company had made with its engines, of both types, showed an entirely different picture. For instance, a comparison for a 35 000 hp plant, with clutches and gear included, showed:

Single-shaft		Twin-shaft	
crosshead plant	m.s. plant	crosshead plant	m.s. plant
1265 t	443 t	2 x 1200 t	2 x 461 t
35·2 kp/hp	12·3 kg/hp	35·2 kg/hp	12·8 kg/hp
Difference in weight: 822 t		Difference in weight: 739 t	

Of course, the height of the engine was not decisive in all cases. However, it was important in, for instance, container ships, since in there the auxiliary engine room, workshops and storage rooms were situated above the main engine, leaving additional cargo space, lengthwise, for containers. A comparison of 20 ft container capacity for a given size of ship and a c.s.r. of 40 000 hp showed:

Steam plant	Crosshead plant	m.s. plant
1600	1546	1676

Regarding sudden uncoupling of a potentially loaded engine, this applied also to stationary generator operation in power stations, where under some circumstances, sudden unloading of the generator could occur at full load. Since M.A.N. medium speed engines were also sold for such stationary operations, there was available, in case of overspeed, a quick-acting shut-down device, independent of the fuel control system, which could also be used for marine engines, provided they were equipped with a coupling, which could be disengaged while they were in operation.

Related Abstracts

Torsional Vibration in Marine Transmission Gear

This paper is not proposed to undertake a comprehensive analysis of all the problems under this head; it merely seeks to call attention to the conditions that ensure the correct interaction of gear wheels in the sphere of the application of toothed gears and elastic couplings in marine transmission systems. The increasingly frequent application of medium speed, non-reversing power plants in marine propulsion systems entails the use of reduction gears in the case of controllable-pitch propellers and reverse reduction gears in the case of fixed pitch propellers. The transmission system is liable to torsional vibration, to which gear wheels are particularly responsive. Although the classification societies stipulate the permissible stress levels that may accrue from torsional vibration in transmission shafting, they make no regulative references to either main or auxiliary gearing. The application of an elastic coupling in a geared transmission system can substantially amplify the flexibility of the whole system, which encourages a reduction in the frequency of free vibrations. Resonances with low node vibration harmonics detrimental to the system occur at times of low engine speeds, but the stresses in the shafting are well within those permissible. Only high node and high frequency vibrations at which critical speed values are close to those of the engine's nominal speed can become dangerous, particularly for the crankshaft. Often, subsidiary stresses accruing from these vibrations exceed those laid down by the classification societies, and the transmission system requires the application of a vibration damper. In order to ensure the quiet operation of gear transmissions, a certain stated condition must be fulfilled. Otherwise, the normal contact between meshing gears is disturbed by a frequency corresponding to that of the variable moment of the harmonic acting in the transmission. Consequently, gear operation is noisy and gears quickly wear. The variable moments acting in the transmission may be evaluated using Holzer's table and further calculations of torsional vibrations. An appropriate formula is given. The approach described is illustrated by two graphs plotted for a marine transmission system consisting of: power plant, elastic coupling, reduction gear, propeller shafting, and c.p. propeller. The first graph indicates that the transmission will operate more quietly above 108 rev/min and that this value must be adopted as the lower speed limit for the whole operation of the engine. At lower speeds, gear wheels will malfunction. An elastic coupling located between engine and transmission encourages a reduction in vibration frequency and variable moment values.—*Sokol J., Silniki Spalinowe Okretowe i Kolejowe, September 1970, Vol. 3, pp. 28–29; (in Polish); Journal of Abstracts, The British Ship Research Association, June 1971, Vol. 26, Abstract No. 30978.*

Pielstick-Engined U.S. Tanker

The first U.S.-built ship to be propelled by Pielsticks is the 37 276 dwt tanker *Falcon Lady*.

The ship is the first of a series of four ordered by Falcon Tankers Inc., for long-term charter to the then Military Sea Transportation Service, now the Military Sealift Command.

Each pair of S.E.M.T. Pielsticks for the Falcon ships are rated at 15 000 bhp and the S.E.M.T. machines are arranged for operation on heavy fuel.

Principle particulars are:	
Length o.a.	205·00m
Length b.p.	194·50m
Breadth, mld.	27·13m
Depth, mld.	14·27m
Draught, loaded	11·04m
Corresponding deadweight	37 276 tons
Light ship	8 601 tons
Cargo capacity	309 512 bb
Trials speed at 90 per cent load =13 500 bhp	17 knots
Service speed	16·5 knots

Falcon Lady has a continuous main deck with a poop, single superstructure aft, a forecastle, raked stem, bulbous bow and a clearwater transom-type stern. Construction has been to the American Bureau of Shipping classification ✠ A1 (E) Oil Carrier and in accordance with the rules and regulations of the U.S. Coast Guard, U.S. Public Health Service, U.S. Customs and Federal Communications Commission.

The cargo oil space is divided transversely into two wing tanks and a centre tank by two longitudinal bulkheads and is further subdivided in the usual way by transverse bulkheads, resulting in five wing tanks port and starboard and nine centre tanks. No. 1 cargo oil deep tank is further divided by a longitudinal bulkhead on the centreline giving a total of 21 cargo oil tanks. Clean water ballast is carried in the forepeak tank, the forward deep tanks and the aft peak tank.

A Flume-type stabilization system has been fitted to reduce rolling and enhance sea keeping and to sustain sea speed.

A cargo control room is fitted amidships for centralized control for all cargo and ballast pumps, valves and tank level indicator gauges.

A cargo tank ballasting system is provided with two sea valves, one port, one starboard, and a connecting sea main. The system is handled by No. 3 steam driven cargo pump and a motor driven ballast pump forward. Double valved cross-overs are provided between the ballast discharge main on deck and each cargo system filling and discharge main to provide a flexible system.—*Shipbuilding and Shipping Record, May 7, 1971, Vol. 117, pp. 25; 27; 29.*