Recent Development of the M.A.N. Marine Diesel Engine

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The rapid development of large marine Diesel engines during past years is discussed, reference being made also to the influence of the economic position in shipping.

The building programme of the turbocharged M.A.N. two-stroke Diesels is treated and special data are given on a type, the output of which has been increased two and a half times within twenty-five years.

The various scavenging and supercharging systems are examined with regard to:

a) Output attainable and thermal efficiency, respectively.
b) Influence on engine design.
c) Influence on weight and price.

Brief remarks are made as to what extent a further increase in supercharging is possible and advisable.

Special technical problems discussed in the paper, cover:

1) Crankshafts, torsional vibrations, longitudinal vibrations.
2) Transverse engine vibrations.
3) Further development of injection systems, study of the various possibilities.
4) Cylinder lubrication, timing, and automatic metering of the lubricating oil.

A short report is given on four-stroke dual-fuel engines up to 4,000 h.p. and on experiments with large two-stroke engines for dual-fuel operation. A report is made on very high powered four-stroke engines in Vee-form for special purposes, i.e. highly supercharged engines.

INTRODUCTION

During the past ten years the design of large marine Diesel engines has undergone a development which even the most optimistic expert could not have expected. It was the keen international competition in shipping which led to the following requirements:

i) to build larger and faster ships in order to improve the ratio between capacity and cost of transportation;
ii) to install units developing 15,000-25,000 h.p. with one shaft, without exceeding the requisite dimensions and weights to meet engine room conditions;
iii) to use the cheapest fuels of inferior quality at lowest consumptions and to keep expenses for maintenance and overhaul as low as possible.

By the rapid development of the turbocharged single-acting two-stroke Diesel, which is particularly suitable for heavy fuel operation, these requirements were met by the major European engine manufacturers. As a consequence the installation of steam turbines in the aforementioned power range could not only be checked but even considerably limited. Today a number of freighters, with single-shaft Diesel propulsion of 15,000-18,000 h.p. and large tankers of 50,000-80,00 tons, powered with engines developing 18,000-25,000 h.p. are already in service.

TECHNICAL DEVELOPMENT OF TWO-STROKE ENGINES

Fig. 1 gives the essential data of an engine the basic design of which existed already in 1936 and which is a typical example of the development of the M.A.N. Diesels. In 1936 the six-cylinder version of this single-acting two-stroke engine with a bore of 680 mm., a stroke of 1,200 mm. and an attached reciprocating air pump, developed 3,270 h.p. at 125 r.p.m. The same type, with an increased cylinder bore of 700 mm., has today an output of 6,700 h.p. at 125 r.p.m. and 7,200 h.p. at 135 r.p.m., which can still further be increased to 8,000 h.p. at 150 r.p.m. The main dimensions are smaller than before, the weight is practically the same. The speed of 150 r.p.m. is, of course, quoted with reservations since it can be considered only for very particular cases in shipping. At any rate, the increase in output by 105 per cent, 120 per cent and 145 per cent and the reduction of the power/weight ratio from 82·5 kg./h.p. originally to 39·9 kg./h.p. and 36·8 kg./h.p. and 33·1 kg./h.p. are the typical features of the aforementioned development.

Further particulars about the technical development of the

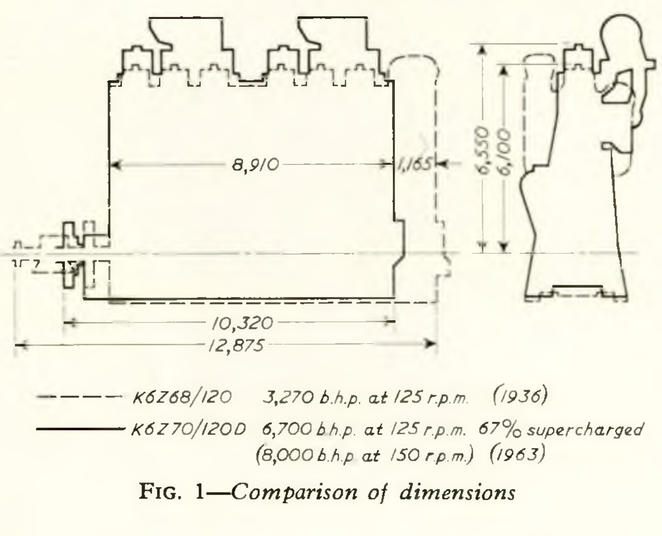

FIG. 1—*Comparison of dimensions*

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TABLE I.—TECHNICAL DEVELOPMENT OF M.A.N. TWO-STROKE ENGINES

	1) KZ68/120	2) KZ70/120	3) KZ70/120A	4) KZ70/120C	5) KZ70/120D
Output per cylinder, b.h.p.	545	665	700	905	1,200
R.p.m./m.e.p. (kg./sq. cm.)	125/4·5	125/5·2	130/5·2	130/6·8	135/8·7
Ignition pressure p_{max}., kg./sq. cm.	45	50	50	55	65
With or without exhaust slide valve	without	with	with	with/without	without
Crankshaft, diameter/axial thickness of the webs, mm.	440/265	445/275	445/275	465/285	510/308
Weight for 6-cylinder, (1–3) with crank for scavenge pump, kg.	29,200	32,500	30,500	32,300	40,300
Distance between cylinder centres, mm.	1,250	1,250	1,250	1,250	1,280
Distance between tie-rods, mm.	1,050	1,050	1,050	1,050	1,100
Diameter of tie-rods, mm.	115	120	120	120	125
Frame (bed-plate, columns, entablature)	cast iron	cast iron	cast iron later on welded	welded	welded
Material of piston crown	forged	cast steel	cast steel	cast steel	cast steel
Cylinder cover, upper/lower part	cast iron	cast iron	cast iron	cast iron	cast steel/cast iron
Weight of a 6-cylinder engine, tons (*without thrust-bearing)	270*	253	247	243	271

design will perhaps be of interest and have been compiled in the following notes and in Table I; The cross-section of the engine KZ 70/120 D (see Fig. 2) serves to explain the essential components:

1) *Engine KZ68/120 (1936-1940)*
Bedplate and columns of cast iron, tie-rods, long piston which descends into the crankcase, double-acting tandem scavenge pump at the free end of the engine, separate thrust bearing.

2) *Engine KZ70/120 (1948-1953)*
Welded bedplate, columns of cast iron with tie-rods, long piston and scavenge pump as in item 1) with exhaust valve in the exhaust duct, incorporated thrust bearing.

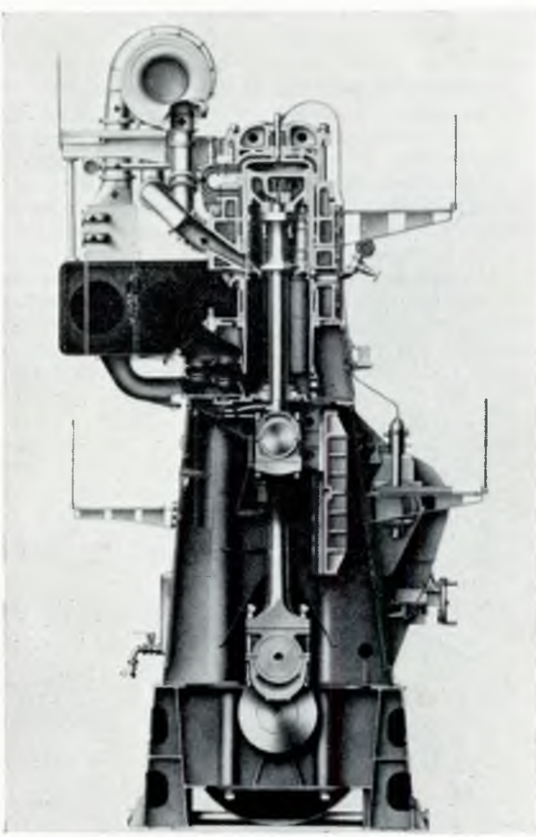


FIG. 2—*Cross-section of KZ 70/120D engine*

3) *Engine KZ70/120 A (1952-1960)*
Bedplate and columns as in 2), from 1958 with shorter welded columns and welded entablatures, tie-rods, exhaust valve as in 2), shorter piston, cylinder space and crank gear space separated by a stuffing box on the piston rod. All undersides work as scavenge pumps, only a small additional pump at the free end thrust bearing as in 2).

4) *Engine KZ70/120 C (from 1955)*
Exhaust gas turbocharging up to 40 per cent, frame and overall structure as in 3). At first with, later without, exhaust valve, only part of the undersides used, additional scavenge pump omitted.

5) *Engine KZ70/120 D (from 1962)*
Turbocharging increased to 65-70 per cent, speed 135 r.p.m. and in certain cases 150 r.p.m., frame and structure as in 4) but strengthened, distance between cylinders increased, crankshaft thicker.

The pistons of all engines are water-cooled through telescopic pipes, arranged in the recesses of the columns, with the design almost unchanged during the 25 years.

Fig. 3 is an evaluation of the statistics published yearly by Lloyd's Register of Shipping and The Motor Ship. It proves clearly that Diesel propulsion is continuously gaining ground in the power range for very large ships which, until 1960, were chiefly powered with steam turbines.

As regards the development work done by M.A.N. it is of special importance to point out that after the extremely

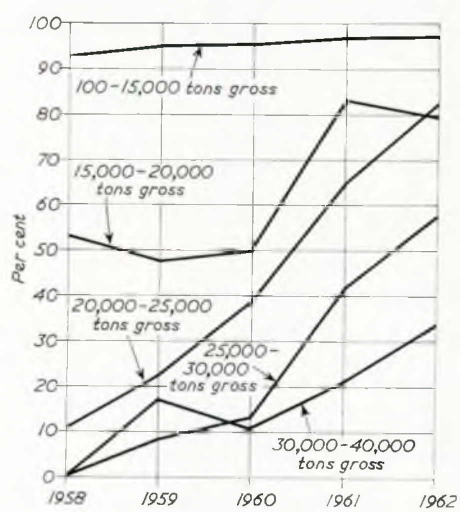


FIG. 3—*Proportion of motorships in the number of annually launched ships*

successful development of the double-acting two-strokes, M.A.N. Augsburg had already built large Diesels developing 12,000-15,000 h.p. in the period 1928-1940. These engines were partly built for peak-load plants and some were of extremely lightweight design for the propulsion of special vessels. After World War II, M.A.N. still held the opinion that the double-acting Diesel was the most suitable and most economical type for high outputs. It was only when shipping demanded operation on heavy-fuel and reduced maintenance, that they changed over to the single-acting engine, since the double-acting types did not quite meet these requirements. When in 1950, production of large engines was again started in the Augsburg works, M.A.N. very soon concentrated on the turbocharging of single-acting two-stroke engines. It became obvious with regard to space and weight requirements that the turbocharged single-acting two-stroke engine was more advantageous than the double-acting version. Turbocharging of two-strokes did not confront M.A.N. with any problem, since very successful trials had been made during the war (1942-1944), viz. a high-speed double-acting marine Diesel had been boosted from 10,000 up to 14,000 h.p. by means of exhaust gas turbocharging. An absolute series system was applied. The turbochargers were developed by a German firm which at that time specialized in this field, and a multi-stage axial blower was used. Unfortunately, all records on these comprehensive trials, which would nowadays be of great interest, were lost at the end of the war. At any rate, even at that time it could be proved that the M.A.N. loop scavenge system is suitable for exhaust gas turbocharging in contradiction to the assertion, repeatedly made, that supercharging would be economical only on engines with a uniflow scavenge system.

SCAVENGING SYSTEM

When comparing the different systems of scavenging, particularly uniflow scavenging and the various scavenge systems with ports, the technical experience should be kept in mind, according to which, any technical solution consists of a compromise of various advantages and disadvantages. Opinions are very often closely related to tradition-bound viewpoints. What matters is the fact that the scavenge efficiency meets the requirements. In former times comprehensive and thorough research work was carried out by taking gas samples from the cylinders in order to try to determine which scavenge system was the most efficient. The results of this research are shown in Fig. 4 (scavenging efficiency in dependence on the air delivery in percentage of swept volume, based on the scavenge air condition before the ports. The dash-dotted line represents the theoretical value for "perfect scavenging"). Rather wide ranges have been given on purpose, for the different scavenging systems, because it has finally been recognized that gas samples only indicated the rate of purity of the charge-air, just at the point where samples had been taken and that, at any other point of the cylinder, the purity can be quite different. The decisive factor is, however, that the engine can sustain a high mean effective pressure with an exhaust free of smoke and that the fuel consumption remains low in association with reasonable exhaust gas temperature. It may be claimed that these requirements are met in every respect by the M.A.N. Diesels with the loop scavenge system and that there are no essential differences in comparison with the uniflow system.

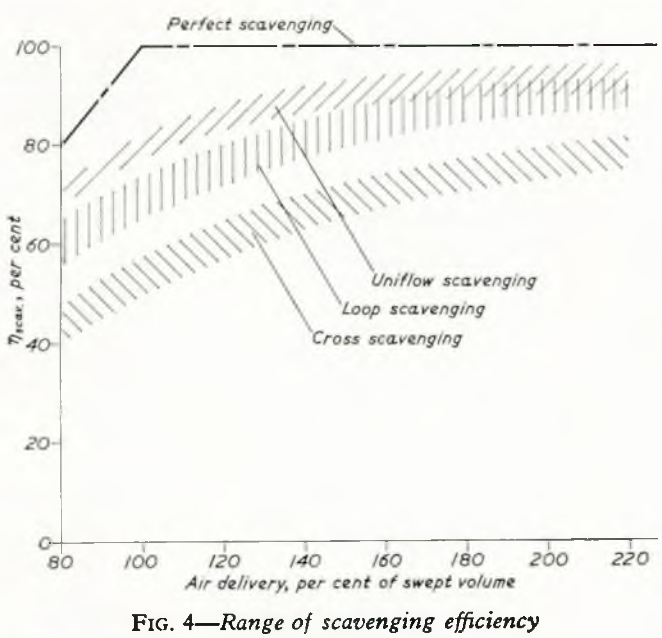


FIG. 4—*Range of scavenging efficiency*

It is, of course necessary to direct the scavenge air very carefully in order to make sure that it takes the desired path. It is for this reason that, at the Augsburg works, scavenge trials are carried out for every type with full scale models. It turned out repeatedly that an increase of the cylinder bore or a modification of the stroke is not associated with a simple geometrical increase of the ports, in order to obtain best results. With every increase of the dimensions it becomes more difficult to achieve "stability" of the scavenge air flow. On the other hand, it must be taken into consideration that, with simple flow trials at low air velocity and almost atmospheric pressures, it is much more difficult to obtain stability than with a momentary scavenge process at high velocity and a pressure of 1·5-1·9 kg./sq. cm. abs. In consequence the results, obtained in practical service with the engine, were always better than could be expected according to the stability of scavenging obtained during the flow trials.

The expert freely admits that with uniflow scavenging it is much easier to obtain stability and that, owing to the shorter distance the scavenge air has to cover, the flow resistance in the cylinder is slightly less than with the loop scavenge system.

This advantage is, however, not fully effective since the area of one or of several exhaust valves is always smaller than that of the fully dimensioned exhaust ports. The uniflow scavenge system, with exhaust valves in the cylinder cover, affords, moreover, the advantage that an increased quantity of exhaust gas energy can be supplied to the chargers by opening the valves at a very early moment, which is of course, feasible only at the expense of the thermal efficiency of the combustion processes in the cylinder. These advantages involve, however, considerable technical expense in the form of rather complicated cylinder covers, as well as exhaust valves with drive mechanisms. In the case of engines with one exhaust valve, it is necessary to provide two or three laterally positioned fuel valves which impair the combustion considerably. It is, moreover, known that the maintenance of a great number of exhaust valves takes up a very considerable part of the maintenance work, when using heavy fuels of poor quality. On the other hand, the extraordinary simplicity of the M.A.N. engine with the loop scavenge system is repeatedly welcomed by a great number of customers. It should be noted in this connexion that the cylinder cover, of simple design, can be removed very quickly, especially because of the fact that it is not connected to the exhaust duct.

The scavenge system has, of course, an influence on the entire design of the engine, particularly on the cylinders as well as on the height and the weight of the engine. Exhaust ports require, as a rule, an increase in height of the cylinder block. If no special control elements are provided in the exhaust channel, a long piston skirt is required to cover the ports during the entire stroke and as a consequence the engine height is slightly increased. According to the available particulars and to a comparison of weights of the various types, the aforementioned design involves an increase in weight which is greater than the weight of the exhaust valves and the associated drive mechanism. According to M.A.N. experience with large ships, powered with large low-speed engines, a difference in weight of 25-50 tons is of no importance whatever. The decisive factor is the price, which is practically the same for

TABLE II.—M.A.N. TWO-STROKE CROSSHEAD ENGINES.

<table>
<tr><th></th><th>Type</th><th>R.p.m.</th><th>Mean piston speed m./sec.</th><th>B.h.p./cylinder</th><th>M.e.p., kg./sq. cm.</th><th>No. of cylinders</th><th>Maximum b.h.p.</th><th>Length between cylinder centres mm.</th><th>Specific weight kg./b.h.p.</th></tr>
<tr><td>A</td><td>KZ57/80D</td><td>225</td><td>6·0</td><td>750</td><td>7·35</td><td>6–12</td><td>9,000</td><td>1,000</td><td>27–28</td></tr>
<tr><td>B</td><td>KZ60/105D</td><td>167</td><td>5·8</td><td>900</td><td>8·17</td><td>5–9</td><td>8,100</td><td>1,050</td><td>31–33</td></tr>
<tr><td rowspan="2">C</td><td rowspan="2">KZ70/120D</td><td>135</td><td>5·4</td><td>1,200</td><td rowspan="2">8·68</td><td rowspan="2">5–9</td><td>10,800</td><td rowspan="2">1,280</td><td>36–39</td></tr>
<tr><td>(150</td><td>6·0</td><td>1,335</td><td>12,000</td><td>32–35)</td></tr>
<tr><td rowspan="2">D</td><td>KZ78/140D</td><td rowspan="2">118</td><td>5·5</td><td>1,335</td><td>7·6</td><td>6–9</td><td>12,000</td><td rowspan="2">1,420</td><td>42–45</td></tr>
<tr><td>KZ78/155</td><td>6·1</td><td>1,540/1,600</td><td>8·0/8·25</td><td>6–10</td><td>15,400</td><td>40–42</td></tr>
<tr><td rowspan="2">E</td><td>(KZ84/160</td><td rowspan="2">115</td><td rowspan="2">6·1</td><td>1,800</td><td>7·9</td><td rowspan="2">6–12</td><td>21,600</td><td rowspan="2">1,550</td><td>43–46)</td></tr>
<tr><td>KZ86/160</td><td>2,000/2,100</td><td>8·4/8·85</td><td>25,200</td><td>39–42</td></tr>
<tr><td>F</td><td>KZ93/170</td><td>112</td><td>6·3</td><td>2,500</td><td>8·7</td><td>6–12</td><td>30,000</td><td>1,650</td><td>39–42</td></tr>
</table>

the different types, since the lower weight for exhaust valves and drive mechanism is made up by the higher expense for machining, in comparison with the expense for the heavier, but quite simple elements for cylinder block and frame. A thorough and detailed examination would be extremely difficult and of little use. The best proof is given by the fact that still today, as for years past, approximately the same number of large engines are being built with the port scavenge system as those with the uniflow scavenge system.

TWO-STROKE ENGINES

For a detailed discussion of the development of the M.A.N. two-strokes the authors refer first to Table II, a list of standard marine Diesels. For the purpose of simplification, the short designations for the individual types and/or groups of types such as A, B, etc. in the first column will be used in the following discussion. All types are equipped with the M.A.N. loop scavenge system and are supplied only as turbocharged engines (the naturally aspirated version of these types is no longer in demand). Owing to these features and the M.A.N. tradition, all types are to a large extent of similar design, i.e. it can be said that M.A.N. have a standard programme for the power range 3,000-30,000 h.p. Variations which are worth mentioning exist only in types A and B, the pistons of which are cooled by means of oil (through pipe linkages) and on which the camshaft runs along the entire length of the engine at the level of the cylinder block. All other types have fresh water cooled pistons (through telescopic pipes) and a short camshaft at about midway level of the engine.

Piston Coolant

As regards the piston coolant, the viewpoints of engine manufacturers have differed greatly during past years. However, common views have been approached gradually, which may be summarized as follows:

a) The cooling effect of water, without doubt, is better than that of oil. It is only by specially arranged oil leads with large quantities of oil, that approximately the same effect can be reached. The bigger the piston and higher the heat transfer the more difficult this becomes. Nevertheless, the danger of carbon deposits on the cooling surfaces is not avoided with certainty, particularly when the oil is not treated with the necessary care.

b) It is true that the means required for oil admission to, and discharge from the pistons, are simpler and cheaper than those for water, but the expense for accessories (coolers, tanks, oil pumps and oil quantities) is considerably greater.

c) In favour of oil cooling it is stated that leakage of water into the crankcase oil, which may have serious consequences (crank gear corrosion), is impossible. However, the fact that several firms (among them M.A.N.) have been using water cooling for many years and that other firms also show a tendency towards this cooling system, is sufficient proof that these dangers can be overcome by a suitable design.

Because of these considerations M.A.N. designs types C-F, in Table II, with water-cooled pistons. Type B will in all probability be changed to water cooling in the near future. Oil cooling will be retained for type A because it would be extremely difficult to find operationally safe elements for the speed of 225 r.p.m., not only in respect of the sealing elements but also because of the danger of cavitation in the pipe system. The fact that the M.A.N. Diesels are all to a great extent of similar design, and M.A.N. tradition, afford a degree of standardization in construction which will hardly be matched elsewhere. As an example it may be pointed out that of type C alone, 255 engines have been built at Augsburg (during the period of development, starting in 1951 and up to today), with a total of 1,784 cylinders. If construction by licensees is added, the total of engines built amounts to 480 with 3,347 cylinders.

Supercharging

The types specified in Table II have, of course, passed various stages in the course of development, especially since the introduction of supercharging and the continuous increase of the rate of supercharging. An example is given in the data in Table I. Nevertheless many parts of the engines remained unchanged and thus the spare parts, too.

The system and particularly the degree of supercharging are of special importance for these types. The fact that the standard types have no uniform mean effective pressure can be explained by various weighty reasons. An increase of the degree of supercharging requires a corresponding increase of the mean indicated pressure and of the maximum firing pressure. This calls for reinforcement of certain components, either because of the requirements of the classification societies or in view of the permissible stresses. In some cases the degree of supercharging was determined with due consideration of the extent of modifications of the design to be carried out. In other cases redesigning would have been of little use because of the completeness of the type programme. As regards type A, which operates at a higher speed, a restriction of the degree of power boosting is advisable in order to avoid thermal overstress of the oil-cooled pistons.

In this connexion it would, of course, be of interest to discuss the basic principles as well as the various systems of supercharging. The authors wish, however, to refer to the excellent papers read during the C.I.M.A.C. Congress held in 1962 (A1-A6 and A11), where the firms concerned gave detailed explanations on the basic principle and particularly on the increase of the rate of supercharging. An evaluation of the information given during the C.I.M.A.C. Congress reveals a perfect conformity of opinion regarding the control of supercharging (*vide* the compilation made in the diagrams of airflow, charge-air pressure, exhaust gas temperature before the turbines and maximum combustion pressure, see Figs. 5 and 6).

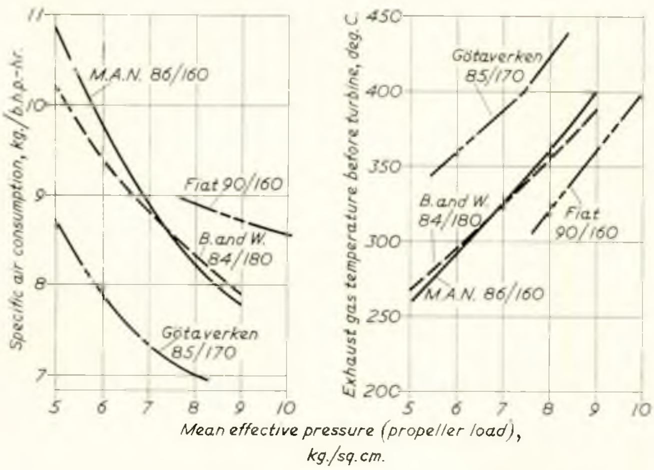


FIG. 5—*Results from some types of large bore engine*

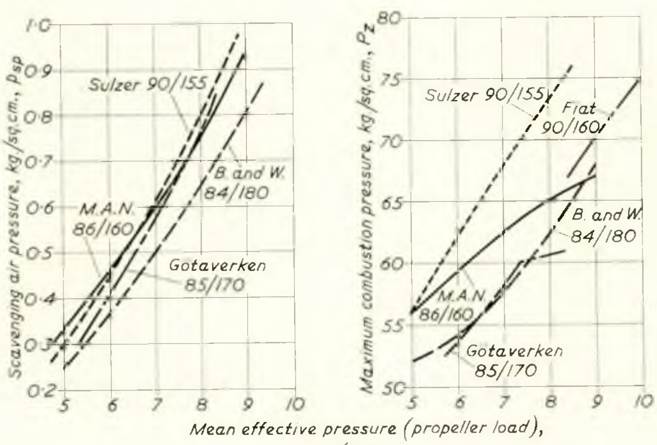


FIG. 6—*Results from some types of large bore engines*

It is remarkable, for instance, that the large engines built by Burmeister and Wain and M.A.N. are entirely different in their design and scavenging process, but have practically the same curves for the specific air consumption (kg./b.h.p.-hr.) and the exhaust gas temperatures. The remarkable features of the Fiat engine are higher air consumption and lower exhaust gas temperatures, but these are probably necessary in view of the scavenging process (cross-scavenging). This requirement can be met by the selection of the series process, for which the quantity of air is not only determined by the capacity of the turbochargers, but also by the dimensions of the scavenge pumps operating in series. With the four types, the lines of the charge pressure over the mean effective pressure are very close to each other, irrespective of the fact that the turbocharger characteristics can be chosen within a rather wide range.

The differences in the firing pressures represented in Fig. 6, are rather striking although charge pressure and firing pressure must increase approximately proportionally to the mean effective pressure if good thermodynamic efficiency is to be maintained. It can be assumed that the mechanical stress that can be allowed for a certain existing design has an essential influence in this respect.

The following important features should be pointed out in this connexion:

1) An increase of the rate of supercharging, i.e. an increase of the mean indicated pressure, requires an approximate proportional increase of the charge-air pressure and of the maximum firing pressure.
2) An increase of the thermal stress of the engine, and/or of the exhaust gas temperature, can be avoided only by increasing the quantity of air supplied to the cylinders in relation to output and/or burned quantity of fuel. At a mean effective pressure of 8·5-9·0 kg./sq. cm., approximately 8 kg./b.h.p.-hr. of air are required.
3) From the facts mentioned under 1) it may be deduced that, with higher boost, the admitted and discharged gas volumes are not increased and, as a consequence, the passages for the air flow may remain unaltered, or only increased within the scope of the increased air quantity according to the conditions mentioned under 2).

A thorough examination of the various systems of supercharging (there are five or six different systems) led the authors to the conclusion that all systems are suitable almost to the same extent for an increased rate of supercharging provided that the engine is of suitable design, an adequate turbocharger is used and additional air, which may be required, is supplied in a most economic way. Each engine builder is fully justified in stressing the particular advantages of the system employed with his engines. On the other hand it should however, be pointed out, that any technical solution is only a compromise of certain advantages and disadvantages, the original standard design of the engines and the tradition being of great influence. The fact that the various systems of supercharging are maintained by the different firms can be considered as best proof that the results do not differ so much. It has already been pointed out earlier that in comparison with the constant-pressure system the advantage afforded by the pulse system, with regard to the turbine output, decreases with increased rate of supercharging. In the case of three-cylinder groups with a 120 deg. firing distance, this advantage is still afforded at 60-70 per cent power boosting, since the exhaust gas is fairly evenly admitted to the turbines. In cases of other numbers of cylinders (5, 7, 8) M.A.N. have come to the conclusion that the constant-pressure system gives better results under full load conditions. This includes also the advantage that the constant high pressure in the exhaust pipe ensures better supercharging upon the termination of the scavenge process, i.e. after the scavenge ports have been covered. This opinion was fully confirmed by thorough investigations made with several large engines with five, seven, and eight cylinders and at a rate of supercharging of 50-70 per cent.

These results are in no way generally valid because there is a great number of factors that influence considerably the turbine output, the turbocharger efficiency and the quantity of air supplied.

The following influences have an effect on the:

Pulse-system

a) The firing sequence and the turbocharger grouping derived from it—the smaller the number of pulses per revolution the lower is the efficiency of the turbine. As is known, the most favourable conditions are achieved with the three-cylinder group.
b) The volume of the exhaust gas piping between cylinder and turbine grid—smaller volumes result in higher pressure peaks and consequently in a higher turbine output. The limit is determined partly by the flow area as excessive gas velocities result in energy losses.
c) Further important influences result from the dimensions and timing of the exhaust, the flow resistance

of the engine and the respective areas and efficiencies of the turbines.

Constant-pressure system

a) Firing sequence and turbocharger grouping are not of decisive importance, but careful attention must nevertheless be given to them to prevent pressure waves, that may have originated in the exhaust gas piping, from interfering with the scavenging of certain cylinders. Large exhaust pipes are more favourable in this respect.

b) Charge pressure and pressure level can be well regulated by dimensioning the turbine area accordingly. With the same input of air weight at the same temperature, the turbine efficiency is considerably better than in the case of the pulse system.

A further investigation is unfortunately impossible within the scope of this paper. The results obtained by M.A.N. can best be illustrated by the comparative diagrams in Figs. 7 and 8.

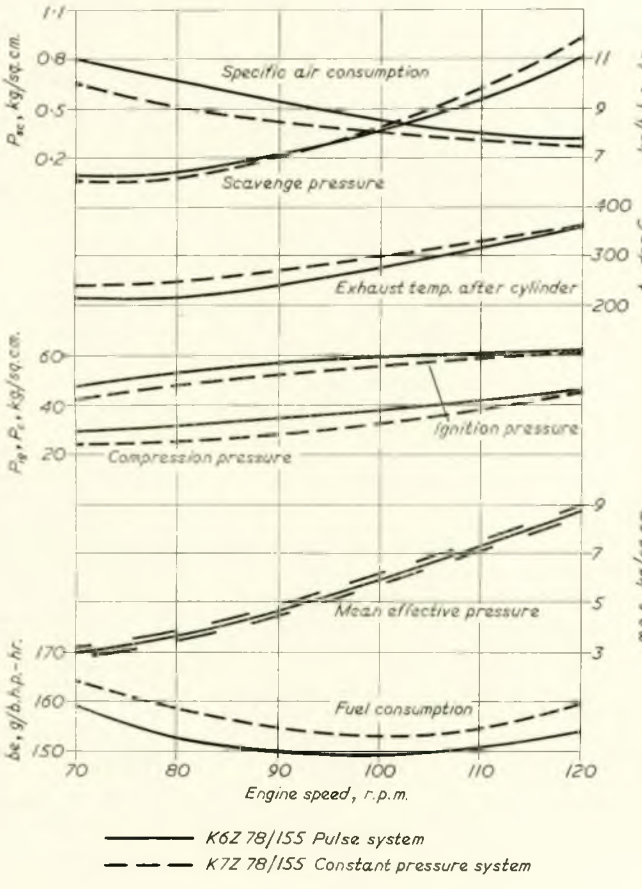


FIG. 7

With each of the two engines in Figs. 7 and 8 two turbochargers have been used in the most favourable arrangement, with throttle discs to dampen unfavourably positioned pressure waves fitted in the constant-pressure piping between cylinders 3 and 4 and cylinders 4 and 5 (firing sequence 1-6-3-4-5-2-7).

It is also noteworthy that the most favourable results of engine K5Z70/120 D (Fig. 8) were obtained with only one turbocharger at the end of the exhaust gas pipe.

It must however be taken into consideration that, at reduced load, i.e. at low speed, with the constant-pressure system the air quantity supplied by the turbines is not sufficient. Therefore, a system must be chosen which ensures the supply of additional air in this range. This is easily possible with the series-parallel system by using piston-underside pumps

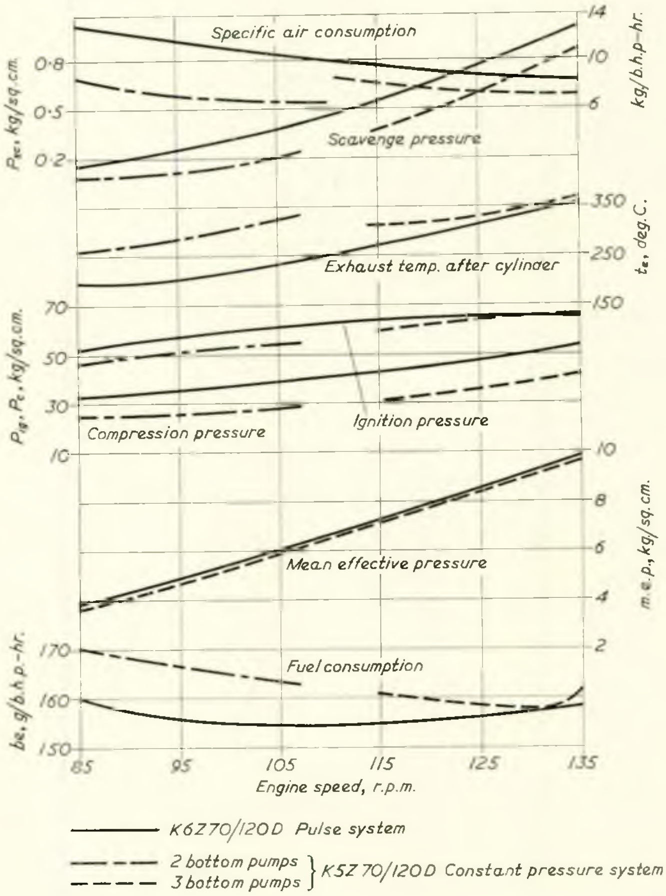


FIG. 8

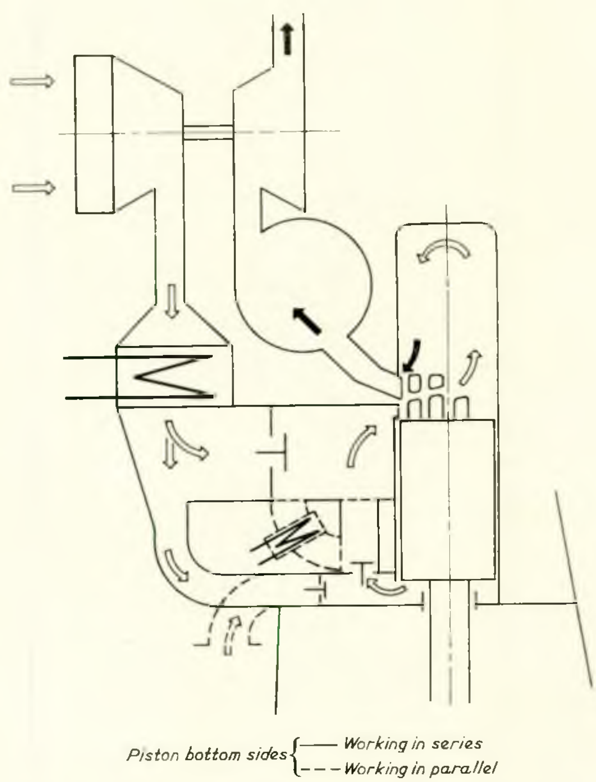


FIG. 9—*M.A.N. turbocharging series-parallel system*

accordingly. As a consequence M.A.N. builds as a rule, all engines with a rate of supercharging of 50 per cent and more at the exhaust side as follows.

i) All engines with 6, 9 and 12 cylinders: pressure-pulse system.
ii) All engines with other numbers of cylinders: constant-pressure system.

At the scavenge-air side, all engines have piston-underside pumps operating on the series-parallel system which is represented schematically in Fig. 9. The scavenge air pipe is divided into two compartments, with large non-return valves opening inwards installed in its partition wall. Approximately 40-50 per cent of the undersides operate in parallel, i.e. they draw the air from outside and deliver it through coolers into stage two (see dotted line). The remaining undersides operate in series, i.e. they draw from the first stage of the scavenge air pipe and deliver into the second stage.

This arrangement offers the great advantage that at low load, perhaps also in the case of complete failure of the turbochargers, the air supplied by all undersides is available. The engine can thus deliver 40-45 per cent of the full output, which means approximately 78 per cent of the full speed.

As soon as the air volume supplied by the turbochargers exceeds the suction volume of the undersides connected in series, the non-return valves between stages one and two will open at a very low pressure gradient of about 100 mm. w.c. From then on the undersides connected in series operate only as supply pumps with a very low pressure gradient. This point is met at about 50 per cent load. This system can be made even more flexible if one of the undersides is provided for change-over from series to parallel operation and *vice versa.*

The authors do not claim that this system is the only possible and suitable one. The absolute series system, e.g. as it is used by Fiat and Götaverken, affords the advantage that a copious supply of air is ensured by providing additional scavenge-air pumps of adequate capacity, and no difficulties are encountered as regards operation of turbochargers and additional scavenge-air pumps. In the meantime M.A.N. have also investigated other systems by means of which in spite of some additional scavenge-air pumps operating in parallel the danger of surging can be avoided in the range of low load.

At Augsburg various trials have been carried out, i.e. the additional air was not led into the scavenge-air receiver, but directly to special sections of the nozzle ring of the turbines, without cooling. These sections of the nozzle ring can be so dimensioned that more energy is admitted to the turbines, even at the lowest speeds and, as a consequence, the blowers are able to supply a sufficient quantity of air. Owing to the fact that the additional air is not admitted to the blower side, there is no danger of surging. A schematic sketch of this system is given in Fig. 10. Tests have also been conducted where part of the underside pumps supply towards the turbine segment while the other pumps supply the air into the scavenge-air pipe, with the number of the pumps employed being varied.

The results shown in Table III were obtained with the three-cylinder trial engine KZ84/160.

At 50 r.p.m., which is the lowest speed for the three-

TABLE III

	Booster drive 1 underside		Parallel operation 1 underside	
Speed, r.p.m.	118·8	50	119·1	50
Engine output, b.h.p.	6,190	446	6,290	474
Scavenge air pressure, mm. Hg.	646	34	692	17
Specific air quantity, kg./b.h.p.-hr.	7·13	19·7	7·0	13·0
Exhaust gas temperature before turbine, deg. C. (deg. F.)	419 (786)	162 (324)	423 (793)	207 (405)
Fuel consumption, g./b.h.p.-hr.	151·6	215	150	235

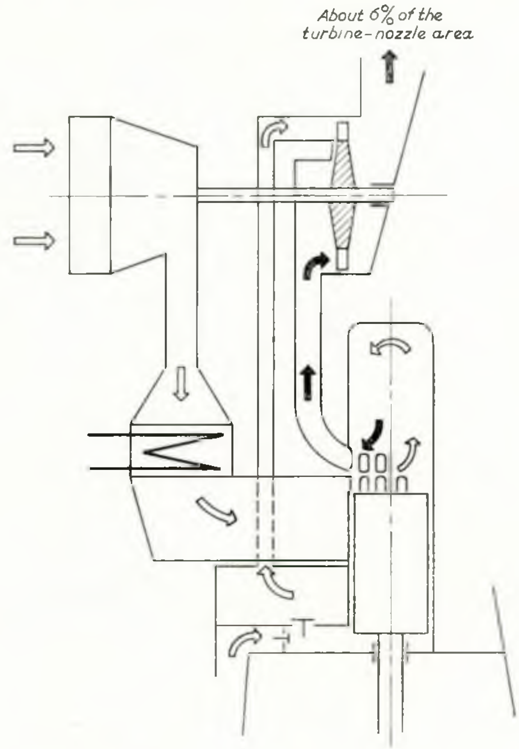


FIG. 10—*Booster drive of the turbine by compressed air*

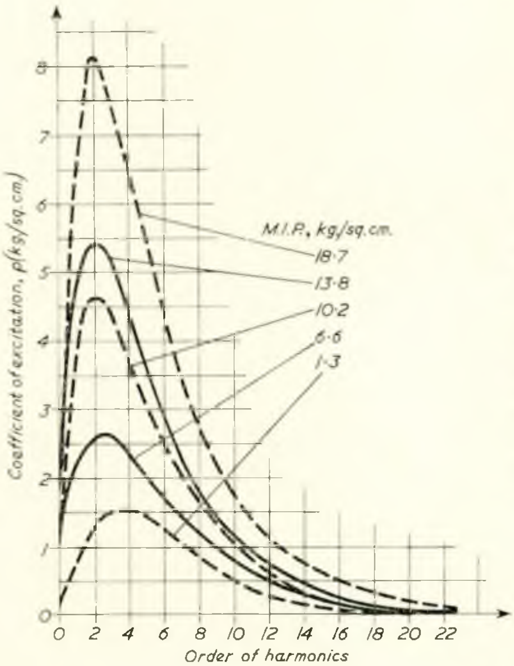
cylinder engine, the booster drive supplies approximately 62 per cent of the entire energy available at the turbine; at 119 r.p.m. this figure is only about 7 per cent.

Tests were likewise carried out with a five-cylinder engine KZ70/120 D. Here, four underside pumps were employed, two of them supplying to the turbine segment and the other two into the scavenge air pipe, and entirely surge-free operation was achieved from the lowest speed to overload. The engine could be run at the lowest speed, 30 r.p.m. with an almost invisible exhaust (when considering this lowest speed, it should be noted that the maximum speed of the engine is 150 r.p.m.).

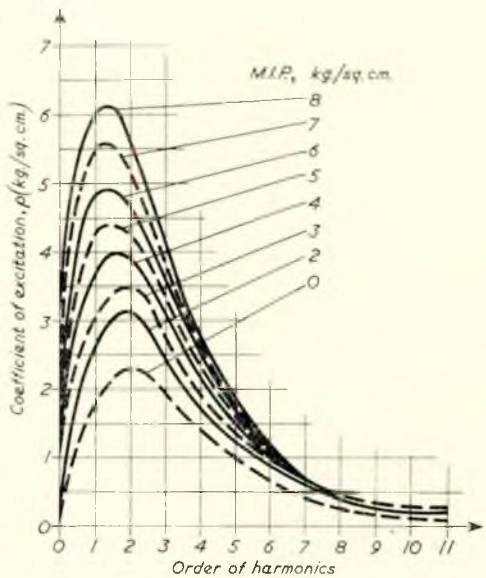
Messrs. Kockums have successfully tested a system where part of the additional air is admitted through injector nozzles right behind the blower outlets. In this case even at lowest engine speeds, the air is prevented from flowing back to the blowers, i.e. stability of operation is ensured throughout the entire speed range.

Shipowners and others have repeatedly raised the question of whether a further increase of the rate of supercharging of two-strokes would be possible. However, prophecy is not a matter for the technical man. The way of further development can, however, easily be derived from what is said under items 1) to 3) on page 5. As matters stand, the turbochargers must be designed for increased pressures on the turbine side and on the blower side. This is quite feasible. They would, however, require redesigning with due consideration of the requirement that, even at increased pressures, the total efficiency reached hitherto should remain more or less unchanged. This is especially important, since higher quantities of air than previously should be obtained. Moreover, increased quantities of air and higher pressures require considerable enlargement of the air coolers as well as an increase in the requisite quantities of cooling water.

As the running gear and the frame will have to be of still stronger design, an increased engine weight is unavoidable. As a consequence, part of the theoretical gain in the specific power/weight ratio is lost. We are approaching



Single-acting four-stroke



Single-acting two-stroke

FIG. 11—*Harmonics of one cylinder for different m.i.p. (without inertia forces)*

the limit at which the economic side of a further increase in output is questionable, especially when taking into consideration the expense involved by completely redesigning the engines.

The authors fully agree with what Mr. Jackson recently said about the development of the Doxford engines*. There they hold the opinion that, for practical reasons, for the time being, they should not concentrate upon a further increase in the rate of supercharging of two-strokes.

Vibration

The considerable increase of output, with a given swept volume and often with slightly increased piston speeds, has had a great influence on many technical problems which have hitherto been considered as fully solved. This applies, for example, to the safe control of the critical torsional vibrations of the engines. It has turned out that, in the diagrams with increased mean indicated pressures and higher firing pressures, the harmonics which excite torsional vibrations are often much higher, as is shown by the curve of the harmonic tangential forces, in dependence on the mean indicated pressure, given in Fig. 11.

With some numbers of cylinders, the critical speeds showed consequently higher stresses than were expected according to experience. In some cases critical speeds of the 8th, 11th or 12th order occurred which would have required a barred range and this would, of course, not have been popular with shipowners. In some cases although the critical speeds did not subject the crankshaft to undue stress, the irregularities of the movement of the running gear caused vibrations in the vessel. It is of course possible to control such critical speeds without difficulty by means of a vibration damper at the end of the engine. Investigations made by M.A.N. revealed, however, that it was possible to completely eliminate such vibrations by means of a flexibly coupled flywheel with additional damper effect. Fig. 12 shows the design of such a flywheel disc, with a geared rim for the turning gear, and which acts simultaneously as a damper. After having overcome some difficulties with sealing the passages for the lubricant required for the spring packages, this solution has given good results in practical service, i.e. critical speeds and vibrations can be completely eliminated.

Recently various engine builders have made and published thorough investigations of longitudinal vibrations of crankshafts. The authors' company have also attached increased importance to this problem and many years ago developed a complete system for calculating the natural frequencies of longitudinal vibrations of crankshafts.

The procedure has been developed from the usual method of calculating torsional vibrations. Instead of the chain of rotating masses, with drilled torsion shafts in between, there is a vibrating-system consisting of masses and shaft sections that are elastic in the longitudinal direction. The vibrations of the system are then calculated for various frequencies until those frequencies are encountered where the end conditions of the natural frequency are fulfilled.

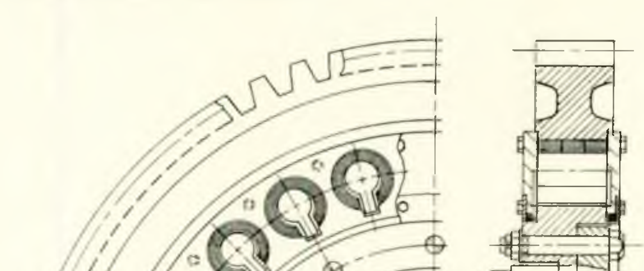


FIG. 12—*Elastic flywheel-turning wheel*

* Letter to the Editor, The Motor Ship, December 1962, No. 509, p. 397.

When determining the system, the following problems arise:

1) The advance calculation of the elasticity of a throw in the longitudinal direction is possible, but it is difficult and unreliable. M.A.N. have placed an actual throw under strain by means of hydraulic equipment and have calculated the elasticity factor from the measured deformation and force.
2) The most difficult question is—which free vibration form is possible? It is reasonable to consider the thrust bearing as a fixed point, i.e. an end condition would be that the amplitude there is zero. A fully rigid thrust bearing is, however, impossible. This is proved by the fact that thrust variations, excited by the propeller, will often cause longitudinal vibration resonances in the crankshaft. This is possible only if the point of attack (the thrust bearing) can likewise move, even if only to a very small extent.
   To be certain, the natural frequency for two extreme assumptions is determined:
   a) The thrust bearing is completely rigid, i.e. it is the fixed point of the longitudinal vibrations of the crankshaft.
   b) The thrust bearing is entirely free, i.e. the entire system of crankshaft, shafting and propeller is considered to be a freely vibrating system. Experience has shown that the relation of the two natural frequencies of these systems is approximately 1:1·4 and that the actual resonance mostly lies 6-12 per cent above the natural frequency a).
3) The third problem is—which exciting forces can produce noticeable or harmful resonances?

The most important influence arises from a strong torsion critical in the crankshaft. Crankshaft torsion under the influence of vibration moments produces a deformation of the throw in the longitudinal direction. This deformation effects a strong excitation with the same, sometimes twice the frequency as that of a torsional critical. If it is near the natural frequency of the longitudinal vibration, a heavy resonance may occur. Even if these two natural frequencies are found at a greater distance, there will be a considerable longitudinal vibration stress outside the resonance range, the amplitude curve of which closely resembles that of the torsional vibrations.

The radial stress exerted on the throw by firing pressure causes an axial motion of the shaft journals (with a throw of engine KZ70/120, 465 mm. in diameter, the web deflexion measured at standstill under the load of the highest firing pressure of 55 kg./sq. cm. was 0·5 mm.). This motion will be considerably smaller during operation owing to the dynamic influences.

The strongest excitation is represented by the impulse speed times the number of cylinders with uniform crank angles. These resonances were found with engines having higher cylinder numbers (9-12).

Finally, the aforementioned excitation caused by the propeller must be considered, the impulse number of which is speed times number of blades and a multiple of the resulting product. This has been apparent in several cases. To judge the additional shaft stress, the maximum crank deflexion is determined from the measured end amplitude and the calculated vibration form. The additional stress is then computed with the help of a formula by Professor Kjaer, which is suitable for this purpose:

$$\sigma_B = \frac{500{,}000.d.f}{R(L + 1{\cdot}5R)}$$

where,

σ_B = bending stress in the crankpin, in kg./sq. cm.
d = crankshaft diameter, in mm.
f = crank deflexion (double amplitude) measured at the distance R mm. from the crankpin.
L = length of the crankpin in mm.

An additional stress of 50 kg./sq. cm. is considered permissible. To date no difficulties whatever have been encountered with longitudinal vibration, even on engines of up to 12 cylinders. There are, of course, longitudinal vibrations reaching amplitudes of ±1 mm. at the free end, on large engines up to 12 cylinders. The additional stresses calculated were negligible in comparison with the normal service stresses of the crankshaft.

Recently, great attention has been paid to transverse vibration of the engine frames in the vessel. The possibility of vibration is increased with the relatively high engines with a long stroke, on which the arrangement of the supercharging equipment at the level of the cylinder covers impairs a favourable distribution of the masses. On the other hand, the increased mean effective pressure causes an increased excitation.

The design of the ship's foundation and of the entire hull in way of the engine room, has a decisive influence on the position of the natural frequencies. Unfortunately this cannot be determined by theoretical calculation. It is for this reason that with all large engine plants, lateral support of the engine frame in the ship is provided or installed from the very beginning. To ensure best results from such a lateral support it is, of course, necessary to provide a perfect connexion between the support and engine frame and to connect the support to such parts of the hull which have sufficient rigidity to distribute the forces in question evenly over a large area.

Fuel Injection

M.A.N. have hitherto employed cam-controlled fuel injection pumps, as have most of the other engine builders. In the high power range of turbocharged engines the injection pressures are rather low at reduced outputs. On the other hand, in the range of full load, relatively high injection pressures are obtained. Though the mechanical stress of the pumps is fully controlled, there is a considerably increased noise level which can be eliminated only with difficulty. In this connexion the stresses in the camshaft drive mechanism, caused by the high injection pressures, must not be neglected and require considerable reinforcement of all elements.

It is for these reasons that M.A.N. thoroughly studied the possibilities of other injection systems, the basic principles of which have been known for a long time.

At first, with large cylinders (780-840 mm. diameter), the common rail system was tried. As accumulating pumps, a number of standard fuel pumps with helical control edges was used, driven by eccentric cams. By turning the plungers the quantity of fuel supplied into the fuel accumulator was so regulated that the actual requirements were met.

With the cam-driven control valves (see Fig. 13), great importance was attached to the requirement that, in the case of one valve sticking, the fuel should not be injected in an

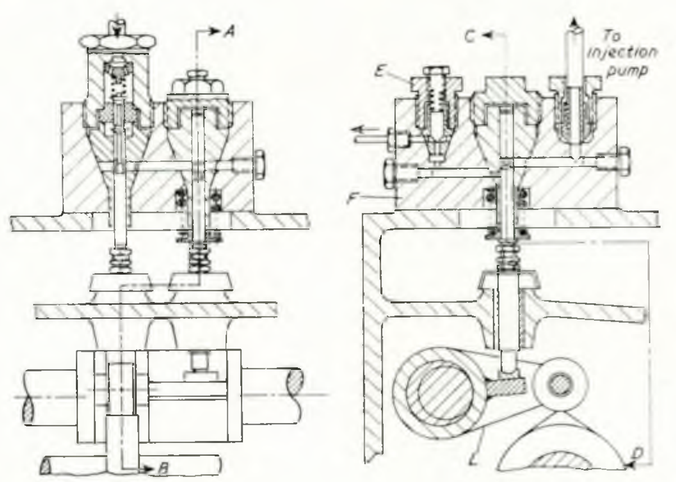


FIG. 13—*Timing valve for constant pressure injection*

uncontrolled manner and in an unrestricted quantity. For this reason two valves are connected in tandem and, moreover, behind the valves, a control pin, actuated by the same drive, is provided for the regulation of fuel admission, so that even in the case of both valves sticking, not more than the maximum overload quantity can be injected. Moreover, in the "out-of-operation" position, the control pin opens the communication F leading to the relief valve E, i.e. between injections the pipe system, control valve and needle valve are pressure-relieved to approximately 100 kg./sq. cm. In order to avoid injection of fuel while the control valve is open, in the case of a sudden shut-down of the engine, e.g. when a wrong manœuvre is carried out, a further device is provided which sets the fuel admission to zero when the speed drops below approximately one-fifth of the rated speed.

The range of injection and the fuel quantity are regulated by the eccentrically seated lever L. The regulating range can be increased further by setting the fuel pressure accordingly. The regulating diagram which is given in Fig. 14 is self-

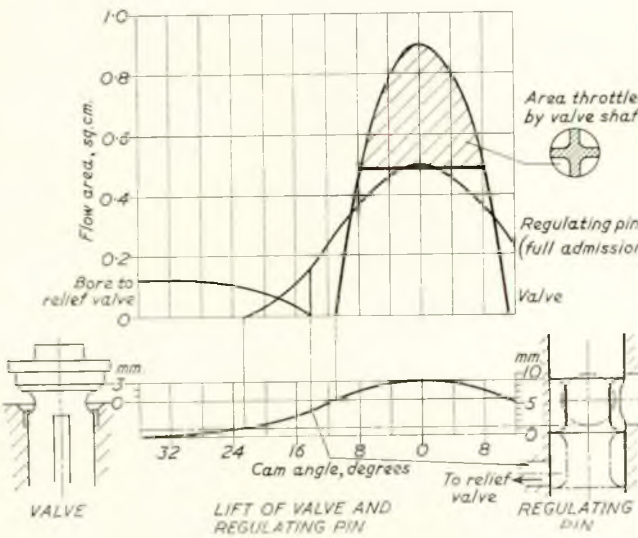


FIG. 14—*Regulation diagram for constant-pressure injection*

explanatory. It is possible to connect the regulating mechanism with the reversing mechanism, since it requires only a very slight change of the injection for an astern run. Reversing can, however, easily be combined with a change of the ignition timing during operation. The best way to achieve this is by turning the light camshaft. As a fuel valve, the standard M.A.N. needle valve was used, however, with slightly larger nozzle orifices than used for pulse injection. Excellent results were obtained with this principle and the same diagrams as well as the same consumption figures, as with pulse injection, were reached.

Comprehensive trials with two engines revealed also some disadvantages which will, however, be easily overcome. It turned out that the supply pumps were of poor efficiency in the full load range, because of extremely excessive losses caused by leakage from the pump space through the helical grooves to the suction space. This can be explained by the fact that the pump is subjected to the full pressure through 180 deg. by the aforementioned drive mechanism, whereas with the pulse pump the full pressure lasts only between 5-8 deg. of crank angle.

Moreover, it was found that the reaction of the regulating gear was rather high, which had an unfavourable influence on the governor and required the installation of stronger servo-motors.

Further development of this injection system has been slightly neglected during the past year since, upon a suggestion made by one of the licensees, Messrs. Kockums, M.A.N. concentrated more on an injection system by means of compression pressure. This system has been employed by Kockums, for many years, for the conversion of four-strokes which were originally provided with air injection, similar to the Archaouloff principle of earlier times, and is known also in England where it is built under Kockum's licence by Wilson and Kyle Ltd.

In co-operation with Kockums, trials were carried out on this principle, on large two-strokes where, as shown in Fig. 15, the fuel quantity is regulated inside the compression pump and the beginning of the injection is controlled by relieving an additional spring in the fuel needle valve, or by opening a small valve in the delivery pipe between pump and needle valve. In both cases the elements are actuated through a linkage by the movement of the guide shoe while close to T.D.C. This principle was tested at Augsburg on a trial engine.

With a ratio gas piston to fuel piston area of approximately 11 : 1 and the injection being controlled from the crosshead by

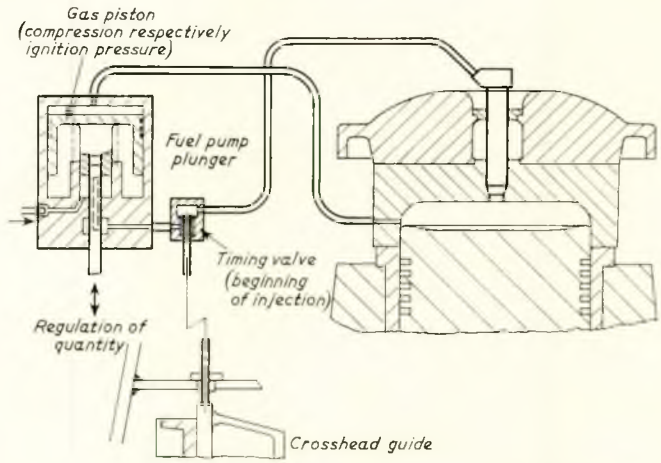


FIG. 15—*Scheme of a gas-pressure fuel injection system*

relieving an additional spring in the injection valve, the following results were obtained:

Fuel pressure at full load (pe = 8·42 kg./sq. cm., n = 115 r.p.m.) 507 kg./sq. cm., fuel consumption 154 g./b.h.p.-hr.

The consumption was inferior to that of jerk injection with injection pumps having helical control edges, by approximately 5 g./b.h.p.-hr., due to a lower injection pressure, longer injection period and some difficulties in controlling the firing pressures.

The gas pressure in the pump cylinder follows the pressure in the main cylinder with a certain time lag so that the injection is continued even after the beginning of firing.

An engine installation built by Kockums, with the injection controlled from the crosshead through push rods (engine type K6Z78/140 C) was subjected to practical testing for more than 920 hours on an ocean-going vessel; operation was on heavy fuel oil up to 2,600 sec. Red. The installation operated satisfactorily, including manœuvring.

Operation results: speed 115 r.p.m., output 6,120 b.h.p., pe = 5·96 kg./sq. cm. P_0 = 39·9 kg./sq. cm., P_z = 50·1 kg./sq. cm., be = 158·8 g./b.h.p.-hr., opening pressure of the needle valves = 220 kg./sq. cm.

A critical analysis of the results led to the following conclusions and/or demands for improvement:

1) Because of the high mean pressures used nowadays with turbocharged two-stroke Diesels, the ratio between exhaust gas piston and pump piston must be considerably increased in comparison with the values used hitherto.
2) The regulation of the fuel quantity inside the pump has to be further improved.
3) The mechanical drive for controlling the injection timing covers a relatively long distance and leaves some doubts as to reliability and life.

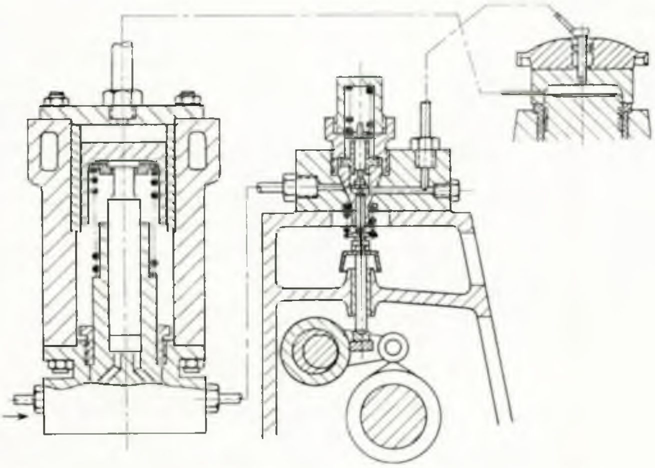


FIG. 16—*Arrangement for gas-pressure injection*

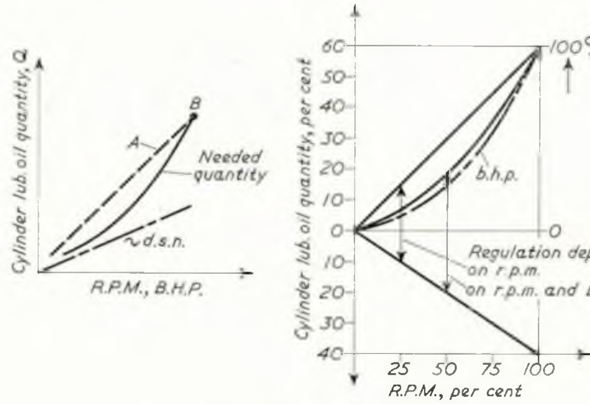


FIG. 17—*Regulation of cylinder lubrication*

Therefore, a revised version of this injection system was developed, which can be called a combination of the constant pressure injection and the aforementioned system (see Fig. 16). The fuel pump, which is driven by the compression pressure, has no quantity regulation, as a consequence the pump is of extremely simple design. Quantity and beginning of injection are regulated by a control valve which is actuated from a camshaft similar to the common rail injection. The design is, however, much simpler, since all safety devices, provided as protection from excessive injection, can be dispensed with, as the pump can supply only its maximum capacity at the time of the highest firing pressure in the cylinder. The simplified valve which is almost hydraulically balanced has only a very slight reaction on the regulating linkage, so that special servomotors are not required.

This system requires again, however, a camshaft close to the cylinder cover, since the control valve must be positioned as close as possible to the fuel pumps and to the needle valves. This camshaft and its drive mechanism would, however, be of very light design and may be used simultaneously for the control of the starting-air pilot valves and, if necessary, also for the control of lubricators and indicator devices.

This system was provisionally installed and tested, with excellent results, on a three-cylinder large bore engine, the maximum fuel pressure was lower than before by approximately 100 kg./sq. cm. owing to the longer pipes and the additional bends. The fuel consumption was, however, practically the same because the firing pressures can be better regulated by adjusting the cams for the timing valve.

At 50 r.p.m., which is the lowest speed of the three-cylinder engine, the fuel pressures were only slightly above the opening pressure of the injection valves, so that the injection was delayed and consequently the fuel consumption increased.

Cylinder Lubrication

Recently various investigations have been made with regard to the most suitable type of cylinder lubrication for large engines. On turbocharged engines, this problem is rather difficult, since the power ratio between maximum and minimum output is increased more and more and lubrication must be regulated accordingly to meet the demands. In order to determine the requisite quantity of lubricating oil, it can first be assumed that for maintaining the oil film only, a quantity of lubricating oil is necessary which is proportional to the surface covered by the piston rings per time unit, i.e. is in proportion to d.s.n. (diameter, stroke, speed). Now, part of the lubricating oil is, of course, burnt during combustion, the higher the output, i.e. the higher the temperature, the greater the amount of oil burnt. Moreover, the fact that with increased output the quantity of acid combustion residues increases accordingly, must be considered. As part of the lubricating oil is required for neutralizing the acids, an additional quantity of oil is required to maintain ample lubrication, as shown in Fig. 17.

The normal cylinder lubricators, which are driven from the camshaft or the running gear, supply lubricating oil at constant setting, only in proportion to the speed (see curve A, Fig. 17). As a consequence, it would be necessary either to provide additional adjustment of the lubricators or to set the lubricators from the very beginning in such a way that, at maximum output, lubrication corresponds to point B of curve A. This would naturally lead to excessive lubrication in the low load range. In order to overcome these difficulties, M.A.N. recently introduced a device for regulating the lubricators, in the case of turbocharged engines, which accords approximately with the diagram at the right of Fig. 17. Since lubricators, driven through a lever from the indicating gear are used, it is quite simple to provide an adjusting link in the drive mechanism, by means of which the angle of the drive lever is increased with increasing engine output. Originally it had been intended to provide manual adjustment of the lubricating oil from the manœuvring stand. Later on it was decided, however, to make this regulation automatic in connexion with the fuel admission. This system has given good results, it permits a reduction in cylinder lubricating oil consumption and, moreover, makes possible the adjustment of the quantity of lubricating oil according to the condition of the piston and piston ring or the quality of the heavy fuel used.

For more than 25 years the problem of precise control of cylinder lubricating oil has been tackled, i.e. to supply the oil to the cylinder at exactly the same moment that the piston rings pass the lubricating oil passages. Thorough investigation has revealed, however, that this is extremely difficult, unless special pumps, with a very short supply pipe, are used, i.e. a system which is closely similar to that of the fuel injection. Investigations revealed that troubles with cylinder lubrication arise mostly when hot exhaust gas enters the lubricating oil bores. The oil in these passages is then heated up to a fairly high temperature and is splashed into the combustion space by the expansion of the gas at pressure relief. The authors' company has, therefore, recently applied a number of methods of preventing the ingress of hot combustion gas into the lubricating oil passages. The safest way to do this is by providing a non-return valve close to the outlet in the lubricating oil passage (see Fig. 18). Such valves have been installed in several plants and have given good results, though apprehension may, of course, be felt because of their position, where control is rather difficult. For the rest, by a suitable design of the lubricating oil passage and by means of reliable non-return valves at the connexion of the lubricating oil pipes, gas can be kept out of the lubricating oil passages to a large extent.

It can be said that cylinder lubrication is an operational and economic problem rather than a technical problem of design. The maintenance of a proper lubricating oil film on the liners, and reduction of the rate of wear, depend only

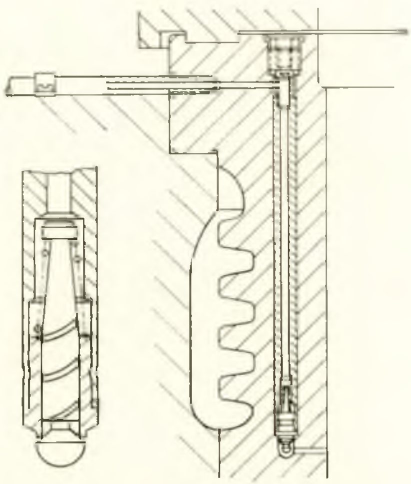


FIG. 18—*Non-return valve for cylinder lubrication*

partially on the perfect operation of the cylinder lubricating system. Other factors are also of great importance:

i) Quality and quantity of the lubricating oil used.
ii) Quality of the fuel, especially with regard to its sulphur and asphalt content.
iii) Load and temperature condition of the engine, also during shut-down and manœuvring.
iv) Stability of the engine and state of the sea (rolling and pitching of the vessel).

This was shown by the investigations discussed in the paper No. A 15 at the C.I.M.A.C. Congress, 1962.

It is extremely difficult, for inspectors and chief engineers alike, to assess these factors perfectly, since it is hardly possible to obtain a clear foundation by measurement. Experience, a certain intuition, and trials which are thoroughly planned and carried out are the only possible means. But even then, it is possible that the results, regarding the lubricating oil which is best suitable and the requisite quantity of lubricating oil, are superseded when another fuel of quite different composition is used (e.g. when changing over from a fuel of Eastern origin to a fuel from American refineries). It should be noted in this connexion that the lubricating oil is not only required to obtain the lubricating oil film but, for operation on heavy fuels, it is required also for neutralizing the acid combustion residues and, moreover, for dissolving and flushing out of residues containing asphalt.

It is the shipowner's problem to balance the costs of cylinder lubrication and those of an increased rate of wear. In the authors' opinion, for example, a reduction of the oil consumption by about 15 per cent would be uneconomic if an increase of the rate of wear of about 15 per cent has to be endured. It is not only the costs of the cylinder liner and its exchange, but also the work and time required for pulling and overhauling pistons and piston rings, that must be taken into consideration. The period between the requisite overhauls of pistons plays nowadays quite an important role when comparing the economic side of turbine plants and Diesel engine plants. According to experience with large-bore Diesels—the associated results are in the possession of the authors—it can be said that in the case of proper maintenance and cylinder lubrication it is possible to extend the periods between piston overhaul to 5,500-7,500 hr. even when operating on heavy fuels. In other words, with tankers or bulk carriers, operation of approximately one year is possible without having to inspect the pistons, i.e. the pistons can be overhauled simultaneously with the routine overhaul of the vessel in dock, etc. As a consequence the opinion that large Diesel plants permit of less service days a year than steam turbine plants, which can still occasionally be encountered, is losing ground considerably. It is admitted that such results cannot always be obtained, especially in cases where heavy fuels of rather poor quality are used. Many refineries, especially Western refineries, exploit the fuels by cracking to such an extent that the residual products, which are diluted with some gas oil in order to obtain a suitable viscosity, have rather a high percentage of asphalt, sulphur and vanadium salts. In these cases it is difficult to obtain clean combustion and, despite the use of high quality lubricating oils, the rate of fouling of pistons and cylinders is increasing rapidly. However, in the authors' opinion, it should be possible to prescribe fuels of better quality in such cases if it is impossible to tolerate reduced periods between piston overhaul. The economic side of heavy fuel operation would hardly be impaired by this, since there is only a very slight price difference in the quality range which would enter into consideration.

Maintenance

Maintenance and overhaul work with large Diesel engines often involves considerable physical exertion on the part of the engine room staff. It is for this reason that when designing engines, great importance is attached to the fact that components, which are subject to wear, are easily accessible without having to carry out a great amount of dismantling work. Moreover, special tools and devices are provided to facilitate such work. In addition to the hydraulic tie-rod pre-stressing device, which has been used with great success for a long time, nowadays other vital bolts are tightened by means of an hydraulic device. Besides facilitating the work to be carried out by the engine room staff, these hydraulic devices afford the important advantage of ensuring the correct pre-stressing of the bolts. A further hydraulic device is nowadays available for pulling the cylinder liners.

In this connexion it should be pointed out that the shipyard and shipowners also should support the efforts to facilitate overhaul work. Where space in the engine room is restricted overhaul work is extremely difficult and can cause a great deal of discomfort to the engine room personnel and shipyard workmen. Sufficient means for putting down large engine components should be provided and the spare parts should be so stored that easy access is ensured. A travelling crane in the engine room and facilities permitting the use of additional lifting tackles would assist overhaul work to a great extent.

Some of the engine components, e.g. the pistons, require overhaul ashore after long service periods. Therefore, facilities permitting easy transport of such heavy components from the engine room to shore without involving considerable dismantling work or detours should be provided. Experience has shown that in cases of urgent repair work, a considerable part of the time available is taken up by the transport of parts aboard the ship.

Control

Recently, a new development of the control of marine Diesels has started to make its way, i.e. the centralized control of the entire engine plant and remote control from the bridge. M.A.N. developed in co-operation with A.E.G., such an installation which was subjected to thorough trials.[1]

Discussion with a great number of shipowners revealed that the opinions regarding possible reduction of personnel differ greatly. This is quite understandable in view of the different operating principles involved.

A fully automatic control system also includes complete control and remote operation of all auxiliaries such as pumps, compressors, separators, etc. To a limited extent such systems are being used on some Japanese vessels and the results obtained during practical service are being watched with great interest.

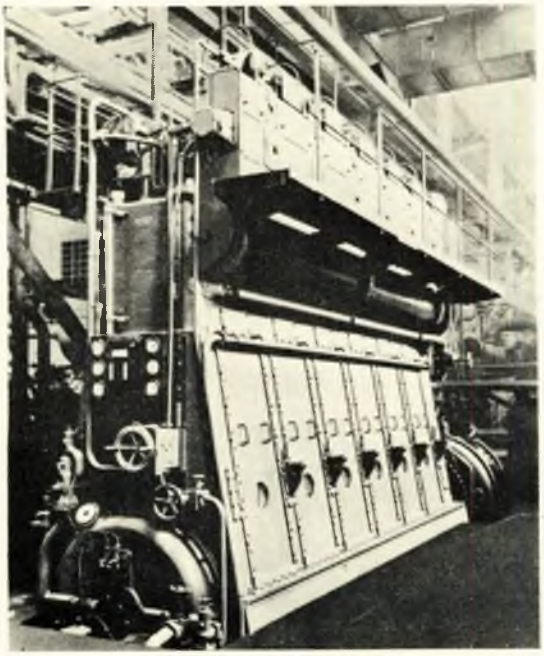
It is of great importance whether and to what extent, control of the engine plant should be effected from the bridge or from a central control room. In the authors' opinion the man in charge on the bridge should control the engine only indirectly, with such measuring instruments as are absolutely necessary, whereas the control of the entire plant should be effected from the central control room. The authors agree with the opinion held by many experts that it is necessary to maintain direct control of the engine plant, by an engineer-in-charge, from the engine room. The tendency to achieve, as a final solution, the push-button remote-control of large engines, operating without control from the engine room is, however, clearly noticeable.

DEVELOPMENT OF FOUR-STROKE MARINE DIESEL ENGINES

As is known, subsequent to thorough and successful trials with high-pressure charging systems, M.A.N. built and supplied a number of high-pressure charged four-stroke engines for the merchant marine in the years 1954-1956. These engines have a piston diameter of 450 mm. and a stroke of 660 mm., the six-cylinder version of which develops nearly 3,000 h.p. at a mean effective pressure of 16 kg./sq. cm. and a speed of 250 r.p.m. (see Fig. 19). The fuel consumption of these engines is 140-145 g./b.h.p. (metric)-hr. M.A.N. and M.A.N. licensees built 24 engines of this type all operating on heavy fuel. Up to date these engines have run for approximately 30,000 or more operating hours. Though at the beginning certain teething troubles were encountered, the results were satisfactory to such an extent that M.A.N. decided to proceed with this line. At present two engines are running at the Augsburg test bed, one eight-cylinder in-line engine and one 12-cylinder Vee-type engine with a bore of 400 mm. and a stroke of 540 mm. These engines are also intended for commercial purposes. They are designed for a mean effective pressure of 16-18 kg./sq. cm. and a piston speed of 7·2 m./sec. (see Fig. 20).

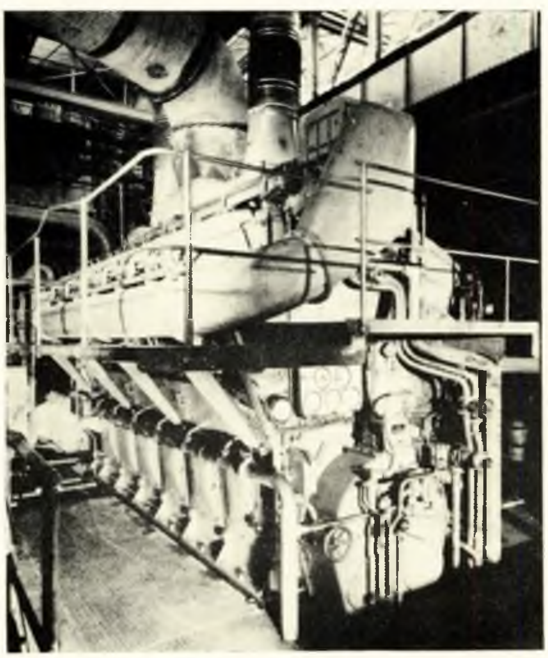
One engine of this type is already in service as a gas engine at the works of one of their licensees, however, with a slightly reduced output (see Fig. 21).

This is noteworthy since high powered engines are also of interest for gas tankers. The aforementioned engine still develops approximately 5,000 h.p. even at reduced mean effective pressure. Moreover, it should be mentioned in this connexion that recently M.A.N. delivered three dual-fuel engines with a piston diameter of 520 mm. and a piston stroke of 740 mm. (see Fig. 22). Each of these engines develops approximately 4,000 h.p. and it is considered quite possible to connect two engines of this type through a gear train to one propeller. Dual-fuel engines—only such engines are admitted for gas tanker service—operate according to the Diesel principle. On demand they can at any time be supplied with a system



n = 250 r.p.m.
m.e.p. = 16 kg./sq. cm.

FIG. 19—*K6V45/66 High pressure charged*



n = 400 r.p.m.
m.e.p. = 16 kg./sq. cm.

FIG. 20—*V6V40/54 High pressure charged*

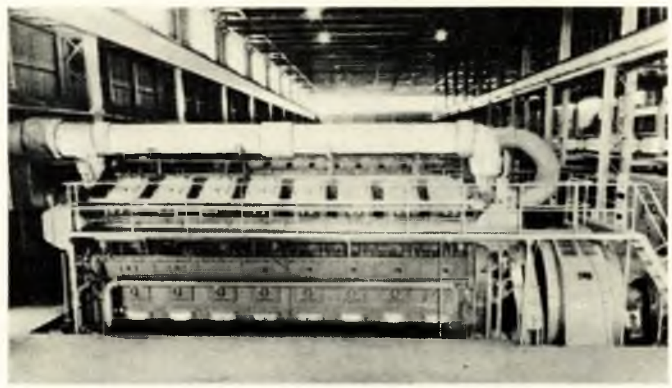


FIG. 21—*V8V 40/54 M.A.N.—licence gas engine*

FIG. 22—*G 10V 52/74 Diesel-gas engine*

for fully automatic change-over to straight Diesel operation. Any ratio of gas/liquid fuel mixture is possible and is even automatically adjusted when the gas supply decreases. At the Augsburg works, trials are being carried out to operate large two-strokes on natural gas.

At the same time M.A.N. designed highly supercharged engines for locomotives and marine purposes, of which about 40 engines have already been supplied and are, for the greater part, already in service. These engines run at a mean effective pressure of 17·2 kg./sq. cm. and a piston speed of slightly more than 9 m./sec. They are of Vee-type with 2 × 8 cylinders with a bore of 240 mm. and a stroke of 300 mm. (see Fig. 23).

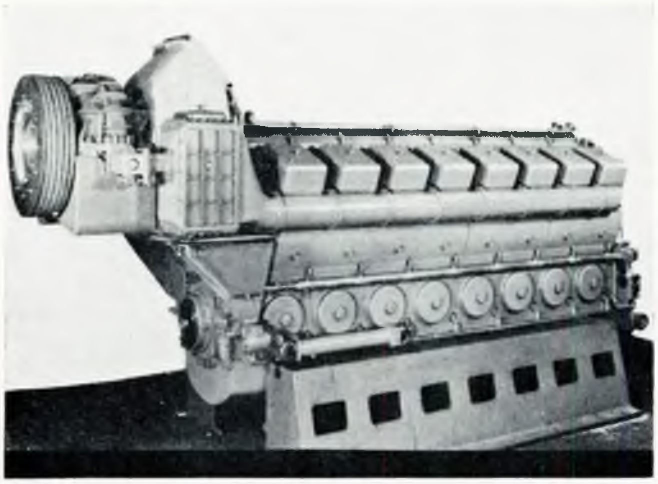


FIG. 23—*Motor V8V24/30*

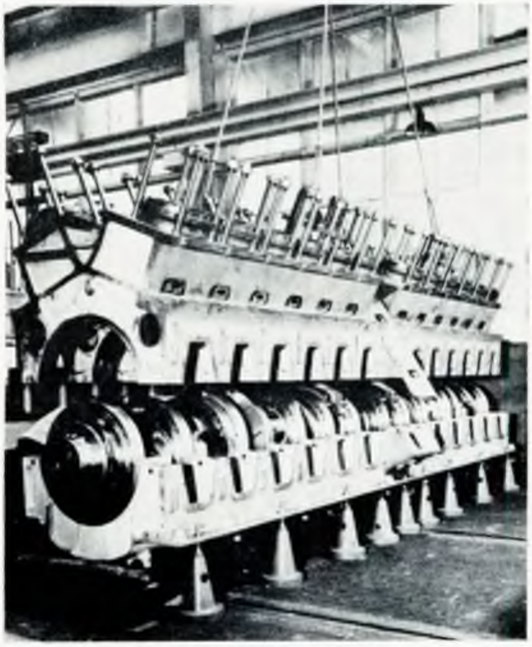


FIG. 24—*Assembling of crankshaft and casing V12V26, 5/30*

Development of this type has in the meantime made further progress. At present trial runs are being made with a prototype for similar purposes with 2 × 12 cylinders with a bore of 265 mm. and a piston speed of 11 m./sec. The maximum mean effective pressure is about 19 kg./sq. cm. The most remarkable feature of this engine is the fact that its crankshaft is seated in roller bearings in order to keep the distance between the cylinders as small as possible and to obtain a somewhat high rigidity of the crankshaft (see Fig. 24). In other words, this type represents the world's highest powered engine, the crankshaft of which is seated in roller bearings. Its maximum output is more than 9,000 h.p. (see Fig. 25).

This engine, especially, clearly shows the development in the design of high speed engines with very high outputs. While at the end of the war the output was 150 h.p./cu. m. of space required (double-acting two-strokes with 2 × 12 cylinders), ten years later, about 1954, 190 h.p./cu. m. and a further ten years later approximately 220 h.p./cu. m. were reached, which corresponds to an improvement of roughly 50 per cent in comparison with the engines built in 1944. At the same time in comparison with the aforementioned two-stroke Diesels, the fuel consumption could be reduced from 195 g./b.h.p.-hr. to less than 150 g./b.h.p.-hr., i.e. by roughly 25 per cent. A similar reduction has been attained for bulk and weight when compared to engines of about the same output.

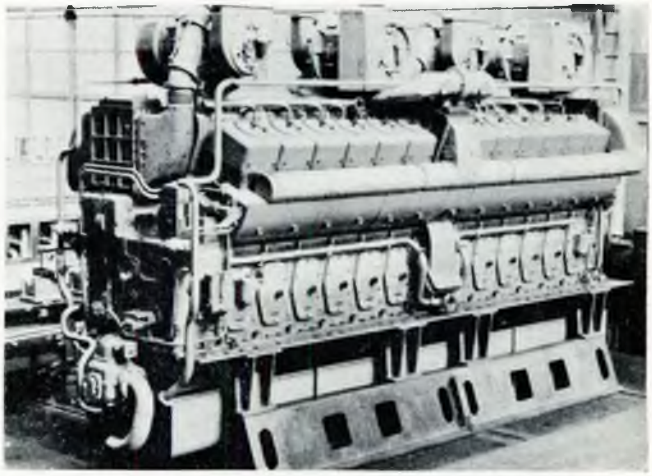


FIG. 25—*V12V26, 5/30-24 Cylinder engine high pressure charged*

With normal engines, as well as with engines for special purposes, the mean effective pressure has been raised considerably during the past 20 years whereas there has been only little progress made in speed increase. This is not so much due to technical difficulties, but rather to the simple considerations that increased power boosting affords an improvement in fuel consumption, whereas increased speed leads automatically to an increase in fuel consumption, owing to the higher frictional losses and the increased work for the suction and exhaust stroke.

REFERENCE

1) "Automated Bridge Control of M.A.N. Engines". *The Motor Ship*, July 1963, p. 173.

Discussion

DR. A. W. DAVIS (Member) said that Professor Sörensen and Dr. Schmidt had come over here and given a most interesting paper, perfectly delivered in English. The paper was the more interesting for the frank way in which virtues and difficulties alike were presented. It was not his intention to dwell in discussion upon any features which were competitive as between engines of one stable and others, although perhaps some of the speakers who followed, more directly representative of rival interests, would produce some of the questions which always provoked the dissemination of engineering experience of the basic order, and provided for an evening full of good-natured argument.

So great had been the rate of development of all the Continental slow running supercharged two-stroke engines, altering the whole pattern of ship propulsion in the process, that the user, which meant the shipbuilder almost as much as the shipowner, must endeavour to see someway into the future so that his present day planning might be as soundly based as possible. In this connexion, he could not agree with the authors when they said that prophecy was not a matter for the technical man. Prophecy there must be and this must be made on the best technical knowledge of the present.

The rapidity of this development had been so great, and in some quarters proceeded apace, that it was a little surprising to read that the authors believed that the limit was being reached to the economic advantage of adopting higher loadings. The reasons which they attached to this opinion would appear to be associated with immediate problems of the anatomy of a particular design rather than to any basic factor. In another setting the view had been expressed, and this had been confirmed by Professor Sörensen's verbal presentation, that crankshaft design would shortly become the obstacle to further development. This suggested a field for urgent co-operative research and perhaps a reconsideration by the classification interests of scantlings that, in the age to come, must surely be more attuned to the best of design and manufacturing techniques rather than the practice that could be tolerated among the producers who were least well equipped and were working to designs which did not perhaps carry the best co-operative experience both in relation to vibration and formation characteristics.

The authors' remarks on axial vibration would seem to suggest a propensity to difficulties in this direction that might be unusual with so relatively stiff a crankshaft. In turbine work there was no doubt in his mind that such difficulties were frequently associated with main thrust block seatings of inadequate strength. The basic truth was camouflaged by the fact that a misleadingly large (or altogether overlooked) proportion of the axial deflexion of a thrust seating was due to shear and not bending; this could only be minimized by thick fore and aft structural plates which were not perforated by any large holes for lightening, plumbers' pipes, or any other purpose. The structure within and below the Diesel engine adjacent to the thrust carrying face must be subject to the same consideration.

The scavenging arrangements the authors described were fascinating. In thinking of scavenging efficiency, however, although it might be conventional, he found it hard to accept that this was usefully described as air delivery as a percentage of swept volume. Clearly account must be taken of the usefulness of the air in its passage through the engine and there was no simple method of defining this by a criterion number. The authors' further remarks on this point would be most interesting, particularly if these could be associated with some information of cylinder liner temperatures in the region of the exhaust and scavenge ports. Could he also have the authors' views of the relatively high rate at which the specific air consumption was falling away at the top mean effective pressures on Fig. 5.

It would also be interesting to know the authors' thoughts, even though they be prophecy, on the application for the propulsion of general purpose cargo ships and tankers inherent in their very fine range of medium speed four-stroke engines.

MR. C. C. POUNDER (Past President) said that, with reference to Professor Sörensen's verbal summary of the paper at the meeting, he doubted if he had ever, at any time, listened for half-an-hour to such a wise, commonsense dissertation upon engines and engine design.

Professor Sörensen talked about the need for simplicity in design. The present speaker was in the fullest agreement with him for, as a machinery designer himself, he had always before him the working ideal of simplicity in all things. Anybody could make a complicated piece of mechanism, a complicated engine, a complicated machinery and pipe arrangement layout, or whatever it might be; but it required a master-craftsman—that was, a master-designer—to transform a complicated mechanism or system into an irreducibly simple construction.

Upon this most excellent paper there must be many men who desired to comment. Accordingly he would confine his remarks to two brief questions. Both of them were on the fringe of the paper and neither was likely to be mentioned by later speakers.

About eighteen months ago, at the close of a discussion with a powerful shipowner, a man for whom he had the highest personal regard, the present speaker was asked if scavenge fires were ever experienced with the type of engine for which he had been responsible. His reply was that there had been too many scavenge fires.

The shipowner next asked if he, the present speaker, would agree that these fires were inevitable. To this question he gave a categorical negative and, further, he went on to say that, in his opinion, scavenge fires were a reproach to the engine designer and to the engine operator, but more especially to the engine designer. Scavenge fires were a challenge; they constituted a problem to be watched for and to be overcome. The shipowner was then asked why he raised the matter. His reply was to the effect that he had a number of M.A.N. engines in his fleet. These engines were only a few years old and, far too frequently, they were in the throes of scavenge fires. He said that the equivalent man to the present speaker in the M.A.N. organization—whoever he might be—had advised him that scavenge fires in single-acting, two-stroke, pressure-charged engines must be accepted as part of the routine of life.

Apart from the fact that scavenge fires, sometimes, could be terrifying, and apart from the extent to which they could reduce the average speed of a ship over a long voyage, there was always the possibility of crankcase explosions resulting from them. For this reason Norske Veritas, for example, required a double top construction on the crankcase.

It would be instructive to learn what the authors had to say on this matter.

In the latest of his company's single-acting, two-stroke, opposed-piston engines such a double top was part of the design.

Of the various things that could induce proneness to scavenge fires, there was one that the operating engineer could not normally control. He could ensure that the pistons were tight, by the replacement of worn rings and cylinder liners; he could ensure that glands and stuffing boxes were in good order, and so on; but what he could not control was the counter-flow of hot exhaust gases through the scavenge ports into the scavenge spaces.

In the ships known to the present speaker that had been bedevilled by scavenge belt fires, the cause had always been traced to the pressure in the exhaust system being higher than the engine design could afford. The worst offending types had been the double-acting, two-stroke, opposed-piston, non-pressure-charged designs. One ship, when two or three years old, was so bad that, on a voyage between New Zealand and Britain, the average speed had been reduced by three-quarters of a knot. This came about because of the aggregation of delays caused by the frequency of heavy scavenge fires. By patient persistence the root-cause was traced to the exhaust gas pressure being abnormally high for the engine design. In due course a substantial reduction was effected by changes in the exhaust silencer and other components, and the scavenge fires disappeared.

Scavenge fires were sometimes the result of a late fuel cam setting. By advancing the cam and thus ensuring that combustion terminated in good time, the exhaust pressure could be lowered to a satisfactory level. But there was a limit to what was practicable.

The next point concerned the interchangeability of engine parts. This was a matter to which shipowners had repeatedly referred in recent years. The present speaker's attitude was simple. Almost anything was possible, if clients were prepared to pay the price. To make proprietary engine components interchangeable amongst say, ten or twelve licensees, was asking a great deal. It meant, of course, the provision of a master set of all the necessary jigs, tools and apparatus which affected interchangeability, and the distribution of sets to all licensees. The resulting cost could be expected to be so high that it would be doubtful if a commercially-minded shipowner would be prepared to pay the price.

The present speaker understood that a well known proprietary engine maker had evolved a system whereby licensees in Britain and elsewhere would be able to supply parts that were interchangeable. If this implied that, amongst the licensees, the engine parts were so divided that one firm made, say, the pistons, another the cylinders, another the cylinder covers, and so on, then this was not interchangeability as he understood the matter.

Perhaps the authors would indicate their approach to this subject.

Mr. P. Jackson, M.Sc. (Member of Council) said that he had found the paper most instructive and informative. The comments and arguments were very fair; they were those of an engineer. In particular, Professor Sörensen had talked about all designs being a compromise; there were advantages and disadvantages in all designs. He would make one observation on this; there must not be any one great disadvantage, otherwise the engine could not succeed. But in other respects, even if an engine had one very great advantage, it did not follow that it would rule the roost.

Professor Sörensen and Dr. Schmidt had dealt with the development of their engine from casting to welding and with the development of turbocharging and the abolition of the exhaust control valves. They had also (much in the same way as he had previously described for his company's engines) reduced their dimensions and weights to the extent shown in Fig. 1, and with more than double the power over the period of thirty years.

With regard to the method of scavenging, he fully agreed with what Professor Sörensen had said—that all methods were applicable. On the other hand he could not quite agree that the same mean pressure could be obtained from all types with equal efficiency for if that were so the simple type would win. There had to be advantages in other directions to compensate for any complication and he believed that there were such advantages.

Professor Sörensen had discussed ratings—a subject which at present was one of his (the speaker's) hobby horses. He agreed with Professor Sörensen that the crankshaft was the principal limitation to increased power output. As mean pressures were increased above about 130lb./sq. in. the rules of the classification societies rightly required that the crankshaft had to be bigger and thus the cylinder centres increased. He noticed that on the latest engine of the M.A.N. designs shown in Table I, the cylinder centres had been increased somewhat relative to the possible minimum. He had himself designed to the minimum of cylinder centres and had then obtained the highest rating considered reliable and had considered that the most economical engine. On the other hand one company in Holland was forcing ratings up to a degree which he considered would give trouble, particularly (as Professor Sörensen had mentioned) in regard to piston rings. It was amazing that a little thing like a piston ring should be the limitation to the development of an engine, but it was so.

He had been very interested to read the section on piston cooling and the statement that the M.A.N. company had been employing cooling by water for the past twenty years with an unchanged design.

As was known, the company which Mr. Jackson represented had considerable experience in this respect. At one time they had water cooling of the pistons through swinging links, and that had been quite satisfactory so long as Diesel fuel was used, but when boiler fuel came into use there was sludge in the crankcase and any water leaking from the swinging links mixed with it and caused corrosion. This problem was overcome by the fitting of a diaphragm and the adoption of oil cooling of the pistons. There had not been a single case of carbonization of the oil in the piston cooling spaces, but he agreed with the statements in the paper that water was a better cooling medium than oil, provided that leakage could be avoided.

With regard to bigger engines, one particular maker on the Continent was saying that oil cooling was no good and that they must be water-cooled. Now this company was having considerable problems, firstly with oil in water and, secondly, with water in the oil. In view of that, had designers been wise to adopt water cooling on big engines?

He hoped the authors would give more details of the design of their piston cooling gear in respect of the arrangement of the glands and the packing.

He had been very interested in many other aspects of the paper, particularly in the curves given for the higher harmonics of the tangential effort with increased power. He had compared these with values from his own company's engines and they agreed pretty well.

With regard to the detuning flywheel shown in Fig. 12, his company used a detuner on the forward end of the engine, which had springs of variable stiffness. The forward end of the engine was, he suggested, a more favourable position for a detuner since it would exert greater influence.

With regard to axial vibration, he agreed that it was excited by torsional vibration or by the propeller. In many cases (though not in every case) it had been found that the thrust collar was practically stationary, and that the natural frequency of axial vibration could be worked out on this assumption. He had never experienced transverse vibration but he supposed that was a problem which would be met some day. He thought there was a simple remedy; to tie the engine to the ship. It was always nice to have a simple remedy. Axial vibration was also much more easily dealt with than was torsional, and an axial damping diaphragm was easy to evolve.

As might be expected, he was very interested in the authors' experiments on the common rail injection system. The authors had gone further than his own company with respect to safety devices, such as the limitation of the quantity of fuel which could be injected at a time. However since adopting the timing valve injection system his company had not had a single valve stick, nor any case of excessive injection.

He questioned the statement in the paper that with the common rail system the fuel pump was delivering fuel for 180 deg. of crank angle. Surely with port control, that could not be the case; it must be no more than 130 deg. Similarly, there was a statement that with pulse injection the fuel was injected over 5-8 deg. of crank angle which seemed to be very very quick injection. He would have thought that 18 or 20 deg. was the very minimum.

The curve shown in Fig. 26 for the Kockum compression pump required further examination, but he had noticed how high the pressure became before the timing valve relative to the actual pressure of injection. This meant that the pump was delivering at the end of its downstroke under the influence of the firing pressure and the fuel having already been injected, the pressure before the timing valve increased to much higher than the injection pressure, which surely must cause trouble with pipes and joints.

He agreed with Mr. Pounder that scavenge fires were a reflection on the designer. In order to eliminate scavenge fires, firstly, there must be good combustion so that the products from the heavy oil did not accumulate round the scavenge ports, and, secondly, there must be a sufficiently early opening of the exhaust valves or ports so that the gases were evacuated from the cylinder before the scavenge ports opened. Then there would be no flame licking past the ports to set any sludge alight. If those two conditions were met he did not think there would be any scavenge fires.

Mr. T. W. Bunyan, B.Sc. (Member) said that there was one confused question he wished to raise, and the authors must forgive him if they themselves considered the matter to be clear and unconfused, which could very well be the case. It was possibly a matter that might have become somewhat invidious largely as a result of commercial pressure in a competitive market. The issue was the rating of large Diesel engines. There appeared to be no standard or uniform approach to this matter, as was the case with auxiliary engines. The "maximum continuous service rating" would appear to most engineers, to be quite a clear and concise definition of the rating of an engine, but with certain interpretations, serious trouble would in fact be encountered if this rating were taken literally and the engine were run day-in-day-out at this power. The "actual maximum continuous service rating" could, in fact, be some 10 per cent less than the "maximum continuous service rating". This might sound crazy but many engineers were already aware of this unsatisfactory situation. The sad part about this ambiguity was that the full implications did not catch up on the operator for four or five years after a ship had entered service, by which time the necessity for running day-in-day-out at the "maximum continuous service rating" was a fact, as the power margins for hull fouling, propeller roughening, etc., allowed for in the power estimate for the ship, had been used up and it was not now possible to maintain the ship's service schedule in deep draught. To make matters a little more difficult, the engine had developed wear in its liners, greater in some than others, and would never again in all its life be placed in a "new condition".

How did the authors deal with this problem of engine rating? It would be interesting to know whether a 20 per cent reduction in rating from the six hours overload rating could be generally accepted as an "actual maximum continuous service rating". Would this automatically take care of the condition of the engine in, say, eight to ten years' time—assuming the engine builders' recommendations were complied with as to maintenance?

He was intrigued that the M.A.N. engine still had the jerk lubricating oil pump on the crossheads. This must have tipped the balance unfavourably with some owners. The engine was actually a robust, rugged affair with remarkable simplicity of design and proven reliability, yet this complication was apparently necessary. He was at a loss to understand this.

The high cylinder oil consumption was well explained in the text. Did not this heavy consumption, however, increase the hazards of fires and make for a dirty engine?

A word of caution should perhaps be said concerning the very popular trend of the adoption of hydraulic tensioning methods, for dynamically stressed bolts. The hydraulic methods generally employed used an extension of the threaded portion of the bolt, to which was screwed the hydraulic straining piston and cylinder. When the bolt was stretched, usually by an amount determined by the hydraulic pressure, the nut was rotated down the thread of the bolt by means of a tommy bar operating through a slot in the skirted extension of the hydraulic cylinder. By gradually removing the hydraulic pressure, the load was transferred to the nut. It would be remembered that this technique had originally been used, for many years with great success, for tightening the long through bolts of large engines. This was an ideal application. Some 200 thousandths of an inch extension was involved and if ten thousandths of this was lost in compressing the nut as it took up the load, this did not matter; there were still 190 thousandths in hand. It could be seen that as the length of the bolt was reduced, the percentage of the tightening strain lost to the nut increased directly. With short stiff dynamically stressed bolts, the technique was positively dangerous, particularly as one never knew what strain was left in the bolt. There could be a tendency to over-strain short bolts in an abortive attempt to allow for this reduction in the residual bolt strain, which could, of course, start a crack of the first thread (without achieving the required residual strain in the bolt). Information recently to hand showed that the tightening strain in a certain hydraulically-tensioned assembly was 20 thousandths, the residual strain specially measured, was found to be only ten thousandths in one test and nine thousandths in another (this, incidentally, was after carrying out a couple of loading cycles to plastically deform the threads before commencing the test which must of course be done in every case).

Mr. L. Collin commented on the many interesting points of view and data given in the paper, especially the conclusions drawn from tests on different supercharging systems. These attracted his attention as they confirmed the views of his organization, which made them decide to use the constant pressure system from the very beginning. In his opinion future development would stress those points still more.

When comparing different large-bore engines, the authors had produced Figs. 5 and 6, showing specific air amount, exhaust gas temperature, scavenging pressure and maximum pressure. In the first two instances especially, the Götaverken engine data differed from that of the authors, for a reason which should be explained. With the Götaverken principle, where use was made of scavenging pumps working in series, it was possible to fix a relationship between stroke volume and scavenging air volume, which was not severely affected by the rate of compression given by the scavenging pumps. The scavenging pump stroke volume was chosen in accordance with Fig. 4 of the paper, in other words, with a limited volumetric surplus, suitable to a uniflow scavenging engine. Because of the automatic compensation in pressure rise given by the pumps if the turboblower efficiency should change, it was not necessary to match the engines with an extra air surplus to avoid insufficient amounts of air after a certain service time. In that way the specific air amount was given by the scavenging pump stroke volume and the scavenging air density required to give a sufficient trapped weight of air during the stroke. The turbocharger efficiency only guided the work done by the scavenging pumps and defined the point at which the pumps would be unloaded. This explained the temperatures in the diagram, where the Götaverken engine had the highest turbine inlet temperature, which increased the turbine expansion work accordingly. There might anyhow be an additional factor, which could exaggerate the difference. On a constant pressure turbine, neither the inlet temperature nor the pressures and velocities varied. On a pulse engine, the thermometer might feel the temperatures when applied immediately after the exhaust valve. Usually a 25-75 deg. C. (77-167 deg. F.) lower temperature was measured after the exhaust valve than before the turbine on his company's large-bore engines.

In the paper most of the factors governing the future possibilities were discussed, but the conclusion drawn was not in favour of a further increase in mean pressures. He did not agree with this entirely. To his knowledge, the turboblowers now available were able to deliver the amounts of air required at pressure ratios necessary for an increased mean pressure and still maintain high efficiencies. Air coolers at suitable efficiencies could also be installed. In his opinion the dominating problem in the future would be the thermal load on the hot parts in the engine and how to avoid that load being unduly increased by normal variations in the running conditions of the engine on board ship due to loss of efficiency as time passed by, different fuel quantities, tropical conditions, etc.

This could be assisted by increasing the specific trapped weight of air to obtain a lower maximum temperature, in directing and timing the fuel spray to minimize the heat transfer coefficient in the combustion chamber, and in avoiding high maximum pressures to diminish the forces and the necessary quantities of material involved in the structural design. It would mean a decrease in thermal efficiency, but if the present trend, with the maximum pressure proportional to the indicated mean pressure, according to the authors, gave unfavourable rises in engine weight as the output increased, new combinations might be studied. This solution might not be necessary, but should perhaps not be disregarded.

His company were of the opinion that the peak of the development had not yet been reached. They offered at present mean indicated pressures of 10·4 kg./sq. cm. (or $\sim$ 150lb./sq. in.) for continuous service. If the two-stroke development were compared with the four-stroke trends also discussed in the paper, it did not seem likely that the pressure rise had stopped yet. Perhaps the authors would comment on this.

MR. J. H. MILTON (Member of Council) said that this was a most interesting paper which, along with others recently presented on main propulsion Diesel engines, was a fund of information, especially, he imagined, to anyone making a choice of machinery. His reading of the paper prompted him to ask the authors for their comments on the following aspects.

Firstly, Fig. 5 on page 201 showed the M.A.N. and B. and W. engines to have very similar exhaust gas temperature/m.e.p. characteristics, one being a loop scavenge and the other a poppet valve engine. In view of the vastly different scavenging systems, how was it that the two curves in Fig. 5 approximated so closely over the whole power range?

Secondly, on page 198 it was stated that "the pistons of all engines were water-cooled through telescopic pipes arranged in the recesses of the columns, with the design almost unchanged during 25 years". Two cases of trouble came to mind, both of which originated through air from a small compressor on the end of the main engine crankshaft being injected into the piston cooling water to obviate hammering. In the first case, oil from the compressor found its way into the piston cooling water, and in the second case, so much air was being injected that serious overheating took place. It would be interesting to know if this system was still used.

Thirdly, one of the criticisms levelled at ported engines was breakage of piston rings. No mention was made in the paper of piston rings in connexion with this design of engine. Piston rings in loop scavenge engines could and did give trouble through breakage. Perhaps the authors would state whether they favoured composite rings; also if they pinned their rings to prevent rotation and whether troubles were frequent through pieces of rings entering turboblowers.

Fourthly, flame rings were apparently found necessary to protect the top of the liner in another design of loop scavenge engine. Why was this vitally necessary in one design and not in another?

Finally, the fact that the pistons, in their designs, were skirted, should enable liner wear to progress further than would be the case with skirtless designs. Perhaps the authors would care to express permissible figures for liner wear, actually worked to in practice, and to quote their experience with chromium-plated liners.

MR. S. ARCHER, M.Sc. (Member) welcomed the paper since, so far as he was aware, there were relatively few technical papers in English on the M.A.N. engine, certainly in this country, produced in recent years.

Piston cooling had already been mentioned. It was not quite clear to him, from the very small illustration, whether the telescopic pipes were actually inside the crankcase. If they were, would it not be a wise precaution to transfer them outside the crankcase?

Secondly, with regard to Fig. 6, this was a very interesting comparison of the current large Diesel designs, and the authors' conclusions here were borne out by the curves, particularly by the Sulzer maximum pressure line. He had calculated what the firing load would be at 1,100lb./sq. in. maximum pressure for that engine, and it came to 500 tons gas load. These were quite formidable forces. The authors' comments with regard to crankshaft scantlings were very much to the point.

With regard to Fig. 11, dealing with torsional harmonic coefficients, he agreed that with the higher pressure charging these values had increased, but for the most part, in his experience, only the lower orders after about the sixth were affected; little difference had been found with the higher orders.

Incidentally, turning to firing with natural gas, which the authors mentioned, surely this might have quite an effect on the shape of the indicator card and that would, presumably, have a corresponding effect on the harmonics.

Dealing with the measurement of axial crank stiffness (page 205, left-hand column) it would be interesting if the authors could say a little more about this, because quite obviously it was not an unique value for any given design, but would depend vitally on the relative crank angle of the adjacent cranks, since obviously the stiffness of a crank with the adjacent one at 180 deg. was less than if that crank were at 90 deg.

With regard to the formula for nominal bending stress in the crankpin due to axial vibration, this had been used at Lloyds' Register of Shipping on several occasions recently, and brief details might be of interest. In the first case, which was a relatively small medium speed engine of 170 r.p.m. the 12th order occurred at round about 80 per cent of the running speed, with a stress on this formula of $\pm$4,000lb./sq. in. in the crankpin fillet, and the 9th order, which was just above running speed, was about $\pm$6,000lb./sq. in. These were actual stresses in the fillet, which corresponded to a stress concentration factor of just over 4. The second case was a larger engine running at 115 r.p.m. where a 4th order stress of $\pm$4,800lb./sq. in. was obtained by the formula. Incidentally, in the first case it was clear that the mode of vibration was the two-node mode in which one of the nodes was at the thrust block seating and the other was in the shafting. In the second case, it was the one-node in which the shafting and propeller were vibrating in the same phase as the crankshaft, with only a single node at the thrust block seating. In general, these one-node frequencies seemed to vary from about 450 to 600 vibrations per minute and were therefore amenable to propeller excitation. For the two-node modes the figures varied from about 800 to 1,600 vibrations per minute. It would be interesting to know the authors' experience on the value of these frequencies.

In a recent paper to this Institute (*loc. cit.*)* he had attempted to compare the relative importance of axial vibration and torsional vibration and, for the same value of vibratory stress, had found that the axial stress had less than half as much effect on the factor of safety of the crankshaft as the torsional stress, so he would agree with the authors generally that its influence was of much less consequence.

With regard to crankshafts, he noted that the authors did not go into design in any detail in the paper. It would be interesting to know whether they felt that the semi-built cast crankshaft was preferable to the semi-built forged. The Society's record showed double the rate of failure of cast semi-built crankshafts compared with forged semi-built crankshafts, for cracked or broken shafts. He wondered whether the lower price (about 30 per cent cheaper for the cast crankshaft) was really worth while.

With regard to the gas pressure fuel pump, it would be interesting if the authors would indicate what sort of financial saving, in a large engine, would arise due to the elimination of the large case-hardened and ground timing gears. This must amount to quite a substantial figure. With this particular type of fuel injection might it not be that poor combustion in a cylinder could lead to reduced combustion pressure, which again could lead to poor injection, which would lead to still worse combustion? This was just a thought which had occurred to him.

With regard to transverse engine vibration and the tying of the engine to the ship's structure, all marine engineers would be loth to do this if it could be avoided, since obviously any damage to the ship's structure might adversely affect the engine. Presumably the ties used were fitted to some shear-bolt device which would protect the engine in the case of such damage, and also presumably the ties were not taken to the ship's frames but perhaps to the engine casing. Could the authors enlarge on that?

Finally, would the authors indicate the value of brake thermal efficiency they were getting on the large two-stroke engine?

MR. B. BLOMSTERGREN said that he belonged to the M.A.N. family. He was from a Swedish shipyard which had been building M.A.N. engines under licence for 40 years now. They were building mostly big tankers, many of them motor tankers, and had always had the problem of getting, as quickly as possible, the biggest engines M.A.N. had developed. They had had, in many cases, to build ten and in some cases 12-cylinder engines at a very early stage. It was the same with the Japanese. This was the explanation why their large-bore engine now did not have the same dimensions as the M.A.N. engine. The M.A.N. three-cylinder test engine had dimensions of 84/160, but when M.A.N. obtained the first order they altered the diameter to get the power the customer wanted and changed over to 86/160. At that time it was too late for his company to change, and that was why they and the Japanese were now building the 84/160 engine and all the others the 86/160. Now the new type, the 93, was coming, and they were glad that this time it had already changed from 900 to 930, so that there would not be again the disadvantage of two dimensions. However, his company's engine, the 84/160, had not remained static in the development of m.e.p. as might appear from Table II. Here the m.e.p. was 7·9, but in the meantime they had followed the 86/160, and the 84/160 now gave 1,900 to 2,000 b.h.p. per cylinder, corresponding to an m.e.p. of 8·4 to 8·8, depending on the number of cylinders, so that the 12-cylinder engine was producing 24,000 b.h.p.

With regard to supercharging, for some years now, M.A.N. had employed the series parallel system. It had been found that, as the degree of supercharging increased, of course more air was needed to keep the exhaust gas temperatures down and prevent them rising too high at full load. As the number of parallel undersides increased, the remaining series undersides decreased accordingly. This meant that the point in the r.p.m. scale from zero to full r.p.m. where the series undersides could do any work was getting lower down, because they could only work in the range where they took over the full air capacity of the blowers. The problem was that the series undersides had to take over before the surging range had been reached. This problem was growing with the higher degree of supercharging.

There was another possible solution, to blow the parallel air into an injector after the turbochargers on the air side. At a very low speed it sucked air through the turboblowers and, as the r.p.m. increased, the pressure before the injector grew higher, and it was not economical to stay with the injector too far up. At a certain point, well above the range of surging, an automatic valve opened with a direct distribution of the parallel air into the scavenging air receiver.

This meant that the turbocharger could be close to the surging curve at normal engine speed and still avoid the risk of surging at lower speed. With this solution there were no series undersides. This meant that there were fewer scavenging valves and a simpler scavenging receiver, because it was not divided, and there were no non-return valves. The efficiency of the blower at full load could be higher and they were getting down a few degrees in exhaust temperatures. When this idea first arose and tests were made on the injector it was discovered that it had been patented many years ago, by Brown Boveri, so that this restricted them a little as to choice of blowers.

Then there was gas pressure fuel injection. As some of those present might know, his company had worked continuously for over 30 years with this type of injection. They had, in fact, had orders for re-building old air injection engines, continuously for years. It was only two weeks since they received an order concerning a Greek ship with an old B. and W. engine. There were very few now, but there was still a small market for them. During the last five years they had concentrated on achieving a gas pressure system suitable for new engines. They had tried many systems for regulating, which was the new aspect here. A new engine needed a pump with the regulating system. The old rebuilt engines already had pumps to regulate the fuel quantity. A lengthy test had been carried out with the new regulating system on one engine in a ship sailing all over the world, and it had been very satisfactory.

In the paper, the authors had made a critical analysis of the system, but he did not agree fully with the three points mentioned there. The regulating system in the pump should not have given any trouble; no problems had been experienced by his company. M.A.N. had gone over to another system, with a small camshaft and a quantity-regulating valve, and he did not think that their experience, with the regulating system in the pump, was very great. His company's system had also been tested on an 84/160 engine at sea and had worked satisfactorily, although this test was not very lengthy.

A question had been raised concerning the savings achieved by using this system. The saving in cost, for an 18,000 h.p. engine, was in the order of 200,000 crowns per engine, or a little more.

According to the classification rules there had to be a spare gear, and one such gear cost 70,000 crowns, and then there was the camshaft, the driving and the reversing mechanisms. It was not necessary to have any reversing mechanism with this compression system because it worked in both directions.

With regard to the automatic control of these engines, an electronic system had been developed, together with another Swedish company, and only the previous week, a 57,000-ton tanker had been put into service, with an 18.000 h.p. engine. The system was tested and it worked very well. Having seen it, his company was quite convinced that there would be a future for some kind of automatic manœuvring of these big engines. Manœuvring could be made much safer and faster than with the usual engineer in charge.

MR. J. F. ALCOCK said that the paper was an extraordinarily well-balanced assessment of the problems facing the designer who wished to work up his engine further in size or

* Archer, S. 1964. "Some Factors Influencing the Life of Marine Crankshafts". Trans.I.Mar.E., Vol. 76, p. 73.

in rating. Of the main problems mentioned, mechanical and thermal loadings, the latter were, in his view, likely to be much more serious in the future. The reason for this was that progress was made both by improvements in materials and improvements in design. As far as mechanical loadings were concerned, in the next five to ten years a good deal of help could be expected from the metallurgists; and he included in that term not only the people who peered down microscopes but those who shaped steel to the form required—by forging, casting and welding. He would not be at all surprised if, in five or ten years, material improvements were such that classification societies would allow both stresses and bearing loadings to be raised considerably to cope with higher powers.

However, with thermal loading, the scope was very much more limited. Unfortunately, there was very little that could be done to improve the thermal properties of the material in one way which did not make it worse in another. Matters could easily be made worse. It was very difficult to find a reasonably cheap material which was more resistant to thermal stress than the ordinary low-alloy steel or cast iron. The criterion was (strength × conductivity)/(expansion ratio × Young's Modulus). In steels, one could not do much about Young's Modulus and if the strength were increased by alloying, down came the conductivity, and the thermal stresses went up about as much as the strength. The same applied to spheroiding of cast iron. Spheroiding made the iron stronger but it also dropped the conductivity, and the last stage was apt to be worse than the first. With regard to thermal problems, then, the buck was passed back from the metallurgist to the designer.

He thought the right basic principle was that of the Doxford liner. The liner proper, through which the heat flowed, was made thin. This reduced the temperature difference which determined the thermal stress. To take the pressure loads, the liner was reinforced by a strongback, with struts or ribs in between. That basic "Doxford principle" would, if one were forced to go higher either in size or rating of engine, have to be applied to other parts of the engine. The theoretically ideal piston crown, for example would be a layer of foil supported on a wire brush, with water oozing through the bristles, so that there was negligible thickness and thermal stress, and lots of little struts to support the foil against the pressure stress.

Coming down to practicability, the struts, between liner and strongback must be a reasonable distance apart, for reasons of manufacture, freedom from blockage, etc., and the minimum liner thickness was then set by local bending stress between supports. The strongback was then designed to provide the additional strength needed to take the pressure stress.

In some circumstances an over stiff strongback could increase liner stress, and one had to calculate the right value. To do this one had to know the liner temperature, inside and out, and this in turn needed knowledge of the heat flux from the hot gases and of the thermal resistance at the metal-water interface. Knowledge on points such as these was being accumulated, but there were still gaps; an important one was in the effect of scavenge air flow on heat flux. It was known that increased air flow reduced the heat flux, but he had seen no quantitative data on this point. Perhaps the authors could give some figures as to this.

He had noticed with considerable interest the successful supercharging of a double-acting engine mentioned on page 199. A year or two back he had seen an old Harland and Wolff double-acting Diesel stripped, after it had done no less than 12,000 hours since the last strip. He had a look at the piston rod gland, which was generally supposed to be the weakest point of double-acting engines, and was amazed to find every ring free, although it had been running on 3,000 second oil. He imagined that this was due to the modern alkaline lubricant which had been used. It rather made him wonder whether the double-acting engine was not worth re-considering, now that alkaline oils extended the maintenance period to a point where its inaccessibility might not be an intolerable drawback.

Mr. W. L. Coventry (Associate Member) said that he wished to question the suitability of the use of Professor Kjaer's formula for the purpose envisaged by the authors, for this reason, that it was the first time, to his knowledge anyway, that this expression had been published in Britain. It had been in use for many years in determining the ratio of crankshaft deflexions to stresses in the crankpin and crankwebs, but it had certain limitations. One was that the formula was essentially for a forged crankshaft with webs and pins of standard proportion and without overlap of the pin and journal. The second was that the stress in the web was 1·4 times that in the journal. This was important, because just as many crankshafts failed through the web as through the journal, and the fact that the stress was so must not be overlooked. In the use that the authors had made of this expression it had been with hand loading of the crankshaft, which had only created a bending moment in the pin and the web by virtue of their shape. There had been no bending moment applied to the shaft; the bending was purely produced locally by end forces. Although the expression was, in his opinion, reasonably good when the crankshaft was 180, when they were approaching say between + or −45, the stresses could be considerably higher, and he suggested that a nearer formula, following on the same build-up, would be: $\dfrac{500{,}000\ d.f}{R(L + 0 - 4R)}$

Mr. A. J. S. Baker said that the paper had provided something for almost all sections of the community. His own interest was connected with lubrication. In particular, he was interested to see the reference to the use of the gas pressure operated injection, and he had one or two queries in relation to this.

There did not appear to be any provision for unloading the high pressure line between the cycles of operation of the injection. Had the authors found this desirable in order to eliminate secondary injection and after-burning? Had they considered the case, as Mr. Archer postulated, when the piston crown had become somewhat eroded and the maximum pressure in the cylinder was reduced? It seemed to him that some compromise might be required to provide a safeguard against excess pressures in the high pressure fuel lines of brand new engines in first class condition, and to ensure adequate injection at part load and low speed in a very worn engine.

He was pleased to see the diagram which appeared during Professor Sörensen's introduction. Had he any needle lift diagrams taken at a corresponding time? These would be very valuable in augmenting the information, particularly if they did not only concern the full load operation but the part load and low speed condition, which he believed to be of considerable importance in preventing cylinder wear, if the low speed injection could be maintained in a very compact diagram.

The cylinder oil distribution was also of considerable interest in regard to lubricating the top end of the cylinder and, in particular, he noticed that the authors had now included the non-return valve. This did not appear on the versions of the KV type engine so far produced by the M.A.N. company, and he suggested that with the very high pressure and high temperature in the four-stroke engine it was even more important to ensure that the oil did not become injected violently during the period immediately after the fall in cylinder pressure.

Mr. Alcock's remarks prompted him to make a point he had not intended to make in connexion with the use of high alkalinity oils. Had he noted that in the application of these oils the oil was actually applied via the sealing rings of the stuffing boxes? This corresponded, in cylinder lubrication, to feeding the lubricant via the piston rings.

He had considered various means of applying the lubricant in the main engine cylinders to the piston rings rather than the periphery of the liner. This offered considerable advantages, particularly in the four-stroke engine, where there was a reversal of the piston ring from one side of the groove to the other at the top of the stroke. This would be useful in providing a sort of timed injection without the complication of the high pressure jerk pump which had been postulated for this application.

He had been very pleased to see some mention of methods facilitating repair work and inspections. This was a very encouraging point, but there were two areas where some ingenuity was still required. First of all, the overturning or inverting of the very large pistons now being used in the 900 mm. range of engines appeared to call for quite a lot of ingenuity in the confined spaces of the engine room, and sometimes resulted in certain bruising in the vital gas sealing areas. Designs aimed at facilitating this to eliminate the bruising of these areas would be very advantageous in reducing wear and in promoting good sealing.

In this connexion also the usual method of removing the ridge at the top of the cylinder was not perhaps very elegant; it usually consisted of free hand removal with air-grinders and these left paths through which a high pressure gas might pass the top ring during the initial stages of combustion.

Any alternative methods the engine builders could offer their customers in providing a smoother and more accurate method of removing the ridge during overhaul periods would be of great advantage in extending the useful life of cylinder liners.

Ing. E. R. Gröschel said that as an hydraulics engineer and fuel injection equipment designer he would confine himself to fuel injection matters.

He had only had a brief time to study the paper, but was somewhat surprised that M.A.N. considered the use of compression pressure operated injection pumps on highly rated engines. Professor Archaouloff's compression pressure operated fuel injection pump was used extensively on the loop scavenged two-stroke engines of the Krupp-Germania Werft and their licensees, but these were very moderately rated engines and he knew of one instance where the gas cylinders of the fuel pumps of a six-cylinder engine on a vessel engaged on the regular service between England and Sweden had to be cleaned and decarbonized after each return trip. However, the chief engineer did not seem to mind.

The compression pressure operated fuel injection pump had to use dirty and hot gas, as operating medium, precautions had to be taken with the gas transfer pipe which no doubt would become very hot at high brake mean effective pressures, heat had to be prevented from flowing into the fuel pump plunger in order to prevent seizure, all this leading to endless complications.

As an hydraulics engineer he was biased, but he would have said that the hydraulic accumulator injection system would score heavily against a gas pressure operated system. He knew he would incur displeasure with the M.A.N. Swedish licensee who, together with a British company, had pioneered the gas pressure operated fuel injection system, but nevertheless, since gas temperatures were going up and heavy fuel was being burnt, he would have said one would be more apt to stake all on the elegant method of the easily controllable hydraulics system dealing with comparatively cool, clean fuel, especially as the trend seemed to coincide with the recent emergence of an hydraulics servo injection system (B.I.C.E.R.A.),

Would the Herr Professor say on which side of the scale he would come down finally—pneumatics or hydraulics?

Mr. V. H. F. Hopkins said that he would confine his remarks to a few questions on items in the paper with special regard to four-stroke cycle engines. He had only one disappointment in the paper, namely that the references by the authors to their excellent four-stroke developments had been so very brief, in favour of the more extensive treatment of the two-stroke engines. There was one point on these latter he wished to raise.

On page 208, the periods between piston overhaul were given as between 5,500 and 7,500 hours *even* when operating on heavy fuels. He took it that the implication that these engines had run on lighter fuels was an accident of translation. On such fuels, bearing in mind the experience today with the medium size four-stroke engine, he would expect to see the piston overhaul periods two to three times longer than those mentioned. He would be glad if the authors would comment on this point.

Turning to the four-stroke engines—the VV24/30 type, he thought, was that originally mentioned by Professor Sörensen at the 1955 C.I.M.A.C. Since then it was learnt that 40 engines had been put into service and he asked if any of these had been employed in the arduous duty of rail traction and if so whether there were any data available on their hours or miles of operation, maintenance, etc.

The V12V26,5 type four-stroke was a most interesting engine and he wondered whether the authors could enlarge a little on their very brief references, e.g. the b.m.e.p. which he translated to be 270lb./sq. in.; was this considered to be a rating or a maximum which had been achieved and what maximum cylinder pressure was associated with it? A previous speaker had referred to the considerable difference between a performance "rating" and other greater output.

The piston speed for this engine was high for its size; was success with this attributable to the crankshaft bearing design or to a break-through on piston friction, lubrication, etc.? For engines which were not tied to relatively low speeds by propeller considerations, etc., there was great advantage in substantial increase in piston speeds and, therefore, crankshaft r.p.m., so as to permit the use of smaller electrical equipment for land generation purposes and better power/weight figures for other applications, both having a bearing on the capital cost aspect.

Bearing in mind the improved fuel injection developments carried out for the larger engines of two-stroke type, he asked whether the performance of the four-stroke engine in question was obtained with injection equipment of orthodox jerk pump and needle injector type.

The turbocharging system employed, he would assume was of pulse type, and about 3 : 1 pressure ratio with single stage machines.

He would appreciate having the authors' answers to these points.

Mr. W. H. Menzies (Member) said that he agreed with the authors' remarks about fuel at the top of the second column on page 208, and had held the same opinions as the authors for some years. There was certainly a vast difference between fuel cracked off at the refinery, at say 1,500 sec. Red. No. 1 and the *usual* method of mixing some good Diesel fuel with the dregs of the refinery, generally known as C.3. Many times, as chief engineer, he had received bunkers from a tank barge in which C.3 and Diesel fuels had been dumped and supposedly mixed by a few puffs from an air hose. Sometimes the two fuels were led in separate lines to a mixing pump which delivered to the ship. Often, when taking C.3 as boiler fuel only, and the hose had been disconnected, he had seen a splash of C.3 fall into river water and promptly sink.

Most seagoing engineers would appreciate the authors' remarks on the advantages of simplicity in an engine or system. Simplicity paid in the long run.

The foregoing remarks about fuel and simplicity reminded him that, a few days before the outbreak of the war in 1939, his tanker was allowed only sufficient bunkers to go from London to the U.S. Gulf. Unfortunately, his second engineer put most of the C.3 boiler fuel into the bunker which contained all the Diesel fuel. The ship had to leave in the first convoy from London and it burned this mixed fuel in one of the simplest systems, the air injection Sulzer. With only the addition of two extra atomizing rings to each fuel valve and very little heat, this heavy fuel burned much better and gave better indicator cards than with some present day solid injections. The engine was put back on Diesel fuel on arrival in U.S.A. This happened a decade before the popular use of heavy fuel for Diesel engines.

He did not care for the non-return valves of the cylinder lubricators being so near the inside face of the liner; his experience of such an installation was that the layer of lubricating oil under the valve became carbonized and sealed up the valve. From Fig. 18, it appeared that to clear the valve or passage the cylinder cover would have to be removed, which was a lot of work for a small job.

With reference to Figs. 5 and 6, he thought that the comparisons would be more complete if curves from the Sulzer engine were included in Fig. 5.

Referring to the top of page 199, would the authors' give their reasons why they considered that the double-acting M.A.N. engine was not suitable for burning heavy fuel. He asked this question about the change from double-acting to single-acting, because about two years ago he was "Chief" of the 18,000-d.w.t. *North Prince*, with a six-cylinder 720 mm. bore M.A.N. double-acting engine (not supercharged). Just after passing the Bahamas, bound for Genoa, during a severe gale and high following seas, a connecting rod fractured at the fork end. The piston pushed out the bottom gland and the piston cooling gear was broken. The whirling connecting rod punched three large holes in the crankcase bottom before the engine was stopped. As the reference marks of the crank were still in line, the broken rod and bearings were taken out, sheet steel electric-welded over the crankcase holes, and the cylinder ports (scavenge and exhaust in centre of liner) blanked off by packing spare piston rings from the top ring of the piston up to the cover. There was no spare connecting rod. After 34 hours stoppage, the ship proceeded on five cylinders and reached Genoa for the loss of only 1¾ knots for the same fuel control.

Subsequent investigation revealed a small flaw where the top of the lubricating oil hole, running through the rod, was plugged, but many engineers put the blame for failure on the reversal of stresses in a double-acting engine and he had recently heard that the foregoing was by no means the first failure of a double-acting connecting rod. Did the authors consider that ever increasing supercharging would so increase the range of reversed stresses that it would be difficult to design a strong enough rod for double-acting?

On page 197 (second column) the dimensions were described, and it was stated: "At any rate, the increase in output by 105 per cent, 120 per cent and 145 per cent and the reduction of the power/weight ratio from 82·5 kg./h.p. originally to 39·9 kg./h.p. and 36·8 kg./ h.p. and 33·1 kg./h.p. are the typical features of the aforementioned development." He had wondered how the German authors came to use the words "at any rate". London might have said "at all events" and Oxford "nevertheless" or "notwithstanding", but "at any rate" came from the North East Coast. It was very peculiar. Perhaps this partly explained the excellence of the paper; he assumed it had been translated by a North East Coast man.

Correspondence

Mr. F. G. van Asperen (Member) wrote that he had found the paper a clear survey of the present state and results obtained nowadays in the field of the large marine Diesel engine and the highly pressure-charged four-stroke engine.

Being connected, in a manner on the inside, but no longer with direct responsibility, with the present performance of both Stork and Sulzer large-bore engines, he restricted his comments to the items of scavenging and the pressure charging system, by agreeing with the authors' opinion that both uniflow-longitudinal and loop or cross-scavenging systems each had their particular advantages and, in consequence, only needed to be developed to optimal efficiency to be equally or comparably successful. The same could be said of the different pressure charging systems, pulse or constant pressure or what could be regarded as the combination of both, according to the actual shape of the gas-pressure diagrams taken in the exhaust lines before the turbines of engines, designed for either principle. The rule, adopted by M.A.N., to build all engines with 6, 9 and 12 cylinders with the pressure-pulse system, and all engines with other cylinder numbers with the constant-pressure system, appeared to be the outcome of a very logical conclusion and would not cause special production difficulties for their engines, as the M.A.N. design already comprised so many flexible features in respect to piston-underside pumps for series or parallel operation. For other, less flexible, engine designs, it was normally possible to correct similar deviations in performance between engines of different cylinder numbers, by varying the specification of the relevant turbochargers or exhaust pipe arrangements, which, to a lesser extent, affected the production of standard engine parts. Although these corrective measures would not perhaps attain quite equal performance results at the maximum attainable output on the test bed, they would be practically, satisfactory for the actual service-output on board.

As to the maximum combustion pressures, shown in Fig. 6 it was evident that, probably for the reasons mentioned by the authors, the M.A.N. engines had a tendency to a flattened top-pressure curve at a lower level than the other engines compared especially at higher loads. It was remarkable that this did not seem to affect the specific fuel consumption, as shown in Fig. 7, which was related to a smaller cylinder bore, but showed the same flat curve at an even lower level, with accordingly relative low compression pressures.

Regarding the cylinder lubrication, he asked the authors if they had ever tried to make use of lubricating points, placed in the cylinder liner at a level between the top ring and second piston ring in the bottom piston position, therefore under the ports, in combination with an exactly timed lubricating oil supply.

This system was successfully applied in earlier (uniflow) two-cycle engines of his company's manufacture, working on heavy fuel, comparable in size to the authors' engine type C from Table II, where only three supply points in the low temperature circumference part were used and a very satisfactory surface distribution was obtained.

As to the matter of remote control, Mr. van Asperen remarked that in most European vessels, apparently the complement of qualified engine room staff had already been so far reduced, that the expense of centralized control and consequent mechanization and automation did not outweigh the remaining possible reduction of assistant personnel. The aims with the undeniable trend in these developments were therefore, in the first place, directed more towards other inherent advantages, like saving time for upkeep and maintenance at sea, realization of still higher standards of reliability and availability of the whole engine room plant, regularity of engine performance and further rationalization of the available manpower. Perhaps later, after having gained more experience in actual large vessels with gradually extended reliable automation and fully centralized controls, which might also justify remote bridge-control, further staff and crew reductions might be allowed by the Government navigation authorities, resulting in a reduction of crew-quarter space (and, therefore, of the accommodation and stewards-service costs) and in a gain in cargo space,

These views represented, at any rate, the general opinion of interested parties from engine builders, navigation companies and shipbuilders in the speaker's country, where ship-automation was probably the most discussed technical and economical subject in the appropriate circles.

MR. W. J. L. FOREMAN (Member) wrote that it was pleasing to note that the critical question of cylinder lubrication was given a reasonable amount of prominence and it was upon this aspect of the paper that he wished to make one or two comments and ask one or two questions.

It had been suggested, at the bottom of page 207, that "cylinder lubrication was an operational and economic problem, rather than a technical problem of design". This seemed to place an unfair responsibility on the operator, as, once the ship was in service, practically the only control he had over the cylinder liner wear was the quality and quantity of the lubricant. It surely must be the responsibility of the designer to determine the peripheral and longitudinal disposition of the injection points in the cylinder, aiming not only at minimum liner wear but endeavouring to ensure a more even pattern of wear along the length of the liner. Furthermore, cylinder and piston ring materials, the number and design of rings in relation to the prevailing mean effective pressure, must also have some effect on ultimate liner wear. The authors' views on these points would be welcome.

At the top of page 208, the authors itemized four factors, which they considered to be of great importance in cylinder lubrication. It was probably fair to say that any consideration of the efficacy of a cylinder lubricant must start on the basis that the total wear in a cylinder was divided into two components—abrasive/mechanical wear and corrosive wear. The former would depend upon the maintenance of an oil film between the rings and the cylinder walls (a most difficult condition to obtain with reciprocating motion), the latter, on the ability of the additives in the lubricating oil to counteract the corrosive effects of the products of combustion.

Before the advent of hyper-basic cylinder oils it was estimated that something like 70 per cent of total cylinder wear derived from corrosion. This was in engines burning high viscosity fuels and lubricated with the only material then available.

Attempts to reduce this wear, by increasing the quantity of cylinder oil delivered, were unsuccessful and it was only when hyper-basic cylinder oils became available that corrosion showed signs of being overcome. Quite clearly, then, quantity of oil must be related to the quality and, from the viewpoint of corrosive wear, quality, in turn, would be related to the ability of the cylinder lubricating oil to neutralize corrosive products of combustion. It followed, and it had been proved in practice, that the adoption of a cylinder oil with high Total Base Number would give better wear results than a cylinder oil of a lower T.B.N.

It might be possible in specific cases, by consideration of the combustion conditions in the engine, the quality of the fuel, and the load and temperature conditions, to estimate the alkalinity required to neutralize completely any acids formed within the cylinder. This alkalinity could be provided, either by a reduced quantity of oil with a high T.B.N. or an increased quantity of oil with a low T.B.N. However, the latter conditions might bring attendant troubles, in the shape of piston fouling. Could the authors report any experience of these two conditions?

The paper mentioned a method whereby the quantity of cylinder oil could be adjusted, presumably decreased, during manœuvring, but it was felt that this approach should be treated with caution, for, although the quantity of fuel being burnt during manœuvring conditions was much less than that at service power, the temperature prevailing in the cylinder was lower and, hence, conducive to corrosive wear. There was a certain amount of danger in trying to deal with this problem on a purely academic basis and a prudent operator would probably decide that it was better to err a little on the side of higher consumption than to risk the possibility of an increase in cylinder wear.

The fitting of non-return valves, as close as possible to the point of injection in the cylinder, was to be commended, not only for the reasons given by the authors, but, also, to overcome "dribbling" of the cylinder oil during shut-down periods, as it was known that this "dribbling" would result in marking or grooving of the liner longitudinally from the lubricator points.

Much more was being done, at the moment, on the timing of cylinder oil injection, presumably with the object of improving general conditions within the cylinders and, in particular, cylinder liner wear. The authors had simply indicated that precise control was difficult, extremely difficult, and the matter was thus dismissed rather briefly. Could it be that they had decided that, with their own particular engines and with their existing method of cylinder lubrication, economic and operational conditions were reasonably satisfied?

Some field results, independently obtained from the KZ78 turbocharged engines, indicated that, over periods of 24,000 hours, maximum wear rates of from 0·05 to 0·1 mm./1,000 hr. were obtained. These were on cylinder oil consumptions of 0·5 to 0·45 g./b.h.p.-hr.

Thus, in a dry cargo vessel, operating say 4,500 hr./year, the maximum wear at the end of fifteen years would still be well below the figure at which it was generally considered advisable to replace the liner. It had also been noticed, in practice, that, where good liner wear conditions existed, piston ring and piston groove wear was most satisfactory.

In general, the authors views on cylinder lubrication were as helpful as any that had yet been published.

MR. S. H. HENSHALL, B.Sc. (Member) remarked, in a written contribution, that one of the most attractive features of the loop scavenge two-stroke cycle engine was the simple design of cylinder head which resulted in easy accessibility. The engine was not an easy one to develop, however, and the authors were to be complimented on the progress they had made.

If the second stage manifold shown in Fig. 9 was individual to each cylinder then there would be a rise in pressure due to the pumping effect of the undersides at a useful time just before the air ports were opened. This rise in pressure would, however, be different for different cylinders if the series parallel system was used. Had the authors any comments to make on this in view of the fact that the M.A.N. engine had this section of the manifold common to all cylinders?

The authors' comments on the constant pressure system versus the pulse system of turbocharging were of great interest. Looking at Fig. 7 it was seen that the scavenge pressure was roughly about the same for both systems, though rising slightly for the constant pressure system at the higher loads. The compression pressure for this constant pressure system was, however, lower than that for the pulse system throughout the speed range. There would seem to be some higher pressure of trapped charge when the pulse system was used and as this was used on six-cylinder engines where three cylinders were grouped together it could well be the result of a higher exhaust pressure due to a pulse in the manifold at port closing. In Fig. 8 the five-cylinder engine had scavenge pressures and compression pressures well below those of the six-cylinder pulse system charged engine, but the relationship was such that a similar suggestion might be made about this comparison also.

The development of a cylinder lubrication system which delivered a quantity more in accordance with the engine's needs, as speed varied, was a most interesting development. Such regulation could be of advantage on naturally aspirated engines, although the load range covered might not be so high as on turbocharged engines. After all, the cycle temperatures were about the same and the top ring temperature was no different whether turbocharged or naturally aspirated. However, for a given cylinder size the turbocharged engines certainly burned more fuel in a given time and this undoubtedly resulted in greater amounts of residues which the oil had to wash away.

The spreading of oil round a large cylinder was almost as big a problem as supplying the right quantity. It would be interesting to know how many cylinder feeds of the type shown in Fig. 18 were used on each line by the M.A.N. engine.

MR. J. H. HOARE (Associate Member) wrote that the paper was especially valuable because of the frankness with which the more important factors influencing engine designs were discussed.

Whilst a very good case was made out for the loop scavenge system, the overall advantages of uniflow scavenging could not be overlooked, especially as the four-exhaust-valve cylinder head had given good results in one engine version, not shown in the comparisons given in Fig. 6. Latest techniques of piston examination, which did not involve disturbing the cylinder head or its associated exhaust valves, made the uniflow design worthy of a closer look. The servicing of exhaust valves had been appreciably simplified by special facing and grinding plant which could be accommodated in a small space. Exhaust valves were capable of running for 7,000-10,000 hours on heavy fuel before overhaul.

The simple valveless cylinder cover was an advantage in that there were no serious thermal stresses. However, at the other end of the cylinder, there were non-return air valves which were sensitive to the presence of dirt or sludge, which could not be entirely excluded, especially with underside of piston compression.

Although the air compressed by the piston undersides could alone maintain engine performance at partial power, ideally, it should be possible to bypass a faulty turbocharger by a simple rearrangement of gas and air ducting. The M.A.N. series-parallel system appeared to have such a number of passages that there might be difficulty in providing circuits to the available turbochargers in such an emergency.

The uniflow system, using turbocharging only, seemed to have a more simple air and exhaust duct configuration which should be more adaptable in this respect.

The problem of achieving proportionately greater scavenging efficiency with less work absorbed by mechanisms and at lower cost, was a most difficult one for the designer, who had little positive evidence to follow. Performance could depend on often quite slight adjustments to the shape of ports and proportions of other parts of the scavenge system. Such changes could be expensive as a complete engine had to be built and run on the test bed before full results of the modifications could be obtained.

Similarly, to the shipowner, the choice of an engine was very difficult as each of the designs offered was made up of respective advantages and disadvantages. Generalizing, it seemed that scavenge performance had reached a practical limit as an improvement in one direction induced an equal and opposite limitation in another.

Concerning component details, the provision of a non-return valve for cylinder lubrication as explained on page 207 and shown in Fig. 18, was no doubt desirable in preventing the entry of hot exhaust gas into the lubricating oil bores. However, such valves would require maintenance. This design required the removal of the cylinder head before the valve could be reached. A modification based on a quill inserted through the cylinder wall might be feasible and so overcome this disadvantage.

Concerning the reduction of the overall length of the engine as development proceeded, this must now depend, surely, on the use of shorter main bearings. Crankshafts were increasing in size due to higher supercharge levels. These two factors might be resolved by use of improved anti-friction materials or lubrication methods. However, such desirable dimensional changes introduced the further difficulty of excessive friction torque on starting. The authors' views on this aspect of development would be appreciated.

Glancing at the engine shown in Fig. 24, which was provided with roller main bearings, perhaps the authors could advise whether any problems of hardness change of bearing components had been detected as a result of cooling in an oil laden atmosphere after sustained periods at full load crankcase and lubricating oil temperatures.

The paper made stimulating reading as it reflected the thorough approach of the manufacturers towards achieving an efficient compact and favourably priced propulsion unit suited for any likely power requirements.

Authors' Reply

In their reply, the authors said that, with regard to Dr. Davis's remarks they had to admit that the technical man must, after all, deal with prophecies to a certain extent much as he might dislike doing this. With this in mind, they were prepared to predict that the trend towards higher mean effective pressure would continue. A discussion of the limits and obstacles that appeared now showed, however, that progress in the near future would not continue as fast as before, especially if one considered the economic problems inherent in the present unfavourable business situation.

The problem of the real scavenging efficiency was, in their view, clearly dealt with by Fig. 3. The ordinate of the diagram gave the scavenging efficiency, i.e. the ratio of the fresh air volume actually trapped in the cylinder as compared with the swept volume. This factor only depended on the ratio of the total volume of scavenge air delivered as compared with the swept volume. The air weight available would then be calculated from the charge air pressure and the temperature.

The highest temperature at the upper edge of the exhaust ports was of the order of 170-180 deg. C. (338-356 deg. F.). At the scavenge ports the temperature was only 80-100 deg. C. (176-212 deg. F.).

A calculation revealed that with increased output, assuming the same engine and turbocharger design, the specific air volume must decrease. Conditions here differed completely from those of Professor Sörensen's premises where the following conditions were assumed for an increase in power: diagrams similar in their geometry, the same exhaust temperatures, proportionately higher charge pressure. In short, if it was required to keep the same specific air quantity while increasing output K times, turbine output would have to rise by K^3. This, however, was not possible without effecting some basic changes of the turbocharger since it was only the exhaust temperature which rose approximately K times.

Whether high-powered four-stroke engines were to be used for cargo vessels and tankers was first and foremost up to the shipowner. The advantages of such geared installations were the following: flexibility, reduction in weight, low overall height. The drawbacks were: a great number of cylinders, and greater sensitivity with regard to low-grade heavy fuels. The aggregate price of such an installation differed only negligibly from that of a slow-running large engine. At all events, technical literature showed that the number of high-powered geared installations was noticeably on the increase.

With regard to Mr. Pounder's remarks concerning scavenge belt fires the authors first wished to state that they were by no means inevitable. They would be most surprised if any expert be he from M.A.N., its licensees or any other engine manufacturer, had evinced an opinion to the contrary.

Until a few years ago such cases were hardly known but recently they had occurred more frequently, strangely enough, however, only in certain plants. It was also recognized that they occurred in the engines of other firms too, i.e. the engines of entirely different design so that it could not be considered a fault that was attached only to certain engine types. Concerning their extremely thorough investigations into the reasons for these fires and the countermeasures that suggested themselves only a few important points could be mentioned here.

Fuel grades were deteriorating all the time. Whereas formerly heavy fuels were supplied mostly as distillation residues, the majority were now mixed with residues from cracking processes, and had a higher percentage of asphalt and vanadium. If complete combustion were not achieved or if droplets of unburnt fuel reached the cylinder walls, the reaction upon the cylinder lubricating oil resulted in a smeary paste on the piston skirt. This was primarily responsible for the formation of coke in the scavenge and, partially, also in the exhaust ports, and this highly viscous sludge was eventually deposited in the scavenge air belt because it was difficult to drain. If the draining-off was insufficient or if no check-ups and cleaning were effected, the sludge might be set alight. These fires, however, were only dangerous if a great quantity of combustible material had accumulated. From observation through windows in the exhaust and scavenge air lines it was known that even in engines which were in optimum condition the penetration of sparks into the scavenge belt could not be avoided.

These remarks then brought them to the following countermeasures:

1) Close observation of the injection system with a view to ensuring complete combustion.
2) Selection of a suitable cylinder oil, avoidance of excessive lubrication.
3) Amply dimensioned drains in the scavenge air belt, covers for inspection and cleaning.
4) Alarm features so that fires can be detected in good time.
5) Possibly CO_2 connexions for effective and rapid fire fighting; their large-bore engines, incidentally, feature the double partition between crankcase and scavenge belt recommended by Norske Veritas.

Interchangeability of spare parts between licence and original engines had been the authors' concern for years. It was, however, understandable that owing to the long separation during the war, the licensees went their own ways. Gradually, however, these faults were being eliminated, and it could be said that today approximately 60-80 per cent of the spare parts for licence engines matched those of the Augsburg Works exactly. The method envisaged by Mr. Pounder whereby the different licensees each manufactured certain parts, would not seem to be practicable.

The authors welcomed Mr. Jackson's complementary remarks concerning the merits and demerits of the various scavenging systems. Some owners would, of course, accept a slightly lower m.e.p. if, in exchange, they received a particularly simple engine of easy maintenance.

The remarks about great difficulties encountered with the glands of the telescopic pipes for the piston cooling system surprised them; complaints about the design which had acquitted itself well for many years, were extremely rare, and it then always turned out that they were due to some mistake such as, for instance, faulty alignment of the telescopic pipes, moulding sand in the system, complete absence of lubrication when starting up, etc.

With regard to Mr. Archer's first question they would like to use this opportunity to explain that the telescopic pipes ran in lateral pockets of the columns whose only communication with the crankcase was a small gap for the telescopic pipe bracket. Separation was therefore, extremely good.

With regard to the arrangement of a detuner it was perfectly correct to say that the latter should normally be fitted on the forward end. If, however, the mode of vibration was of a type where the vibrations to be dampened had their node more or less in the centre of the engine, a detuner installed in the barring wheel would be just as effective.

Concerning the authors' tests of the common rail system, timing of fuel injection was, of course also 18-20 deg., the 5-8 deg. referred only to the peak pressure in the pump.

With regard to the supplementary diagram (Fig. 26) showing the pressure curves of the regulated Archaouloff injection system they wished to make the following remarks.

The pressure rise before the timing valve when injection ended was due to the sudden stoppage of the fuel column between pump and valve.

Mr. Bunyan had raised the extremely difficult question of correct ratings for marine Diesel engines. Attempts to get all the manufacturers to agree on a clear definition had been going on for years. It should, however, be clear to anybody that under the prevailing circumstances where there was a dearth of orders, this was difficult. The authors found that maximum continuous output stated was the output at which the engine could be run continuously provided it was in perfect operating condition or which they could demonstrate on the test bench under guarantee for any length of time. For practical considerations the shipowner would, however, place the continuous service output lower making allowance for a variety of basic operational requirements and the qualification of his engine room personnel. If they were asked, their advice was to utilize 85-90 per cent of the rated output.

The authors could not quite agree with Mr. Bunyan's remarks concerning the jerk lubricating oil pumps on the crossheads. These pumps were, no doubt, very simple elements and had, in more than 10,000 items, given a reliable and trouble-free service. Failures had been extremely rare and were in most cases due to the fact that the pumps had been neglected completely or the lubricating oil delivered to the engine had a high air-content. They had, in several instances, operated engines without these pumps with the result that the bearings became rather unreliable. It was consequently quite obvious why they resolved to retain the pumps.

They were pleased to note that the endurance and reliability of their crosshead bearings in supercharged engines, even with high boost rates, had been recognized.

They also agreed with Mr. Bunyan that with short bolts hydraulic tensioning implied certain risks. However, the bolts to be tensioned in large engines were all extremely long.

The authors were in complete agreement with Mr. Collin's remarks concerning the air amount and the exhaust gas temperatures. They had, in this respect, made some remarks concerning the Fiat engine which was turbocharged on the same principle as the Götaverken engines. In their assessment of the possibilities of a further increase in mean effective pressures, the authors merely wished to point out the limits, resultant increases in weight and, possibly, a deterioration in thermal efficiency, etc. that were bound to counteract this trend. They too, did not doubt that development in this direction would continue, but they were certain that its pace would slow down.

With regard to Mr. Milton's query concerning the exhaust temperatures shown in Fig. 5, it was pointed out that they depended primarily on the amount of air as against the amount of fuel burnt and maximum firing pressure. Since both figures were about the same for the M.A.N. and B. and W. engines, this should also hold good for the exhaust temperatures. There might be, of course, certain discrepancies such as point and method of measuring the exhaust temperatures.

For injecting air into the piston cooling water, the authors still used a small compressor. By keeping the lubricating system of the compressor drive completely separate from the piston space, lubricating the latter as little as possible (only once a day) the oil could be prevented completely from finding its way into the cooling water. It could never happen that the compressor delivered too much air. There was, however, for emergencies, a connexion to the vessel's main air system. If this was not used properly it might, of course, cause trouble.

The statistics of the company concerning the breakage of piston rings in large two-cycle engines were extremely satisfactory, their number being uncommonly small. They employed a very ordinary type of ring with oblique gap, no wear rings, but hardened ring grooves. The rings were *not* pinned to prevent rotation but their ends were carefully chamfered so that they could not stick in the ports. The turbine was protected by a close-meshed grid in case pieces of rings should ever get into the exhaust pipe. There were, however, a few cases where such pieces had been thrown back and forth in the exhaust line for many months until they were finally broken up completely. Very small pieces were then able to work their way through the grid causing damage to the turbine blades which, however, was of a very minor nature. Flame rings to protect the top of the cylinder liners were not required in their case since the separating joint between cover and liner was located rather low and covered over by the piston crown during combustion. Liner wear was not only perfectly normal but it could even be maintained that it was extremely low. In heavy oil operation, it amounted to between 0·05 and 0·15 mm. per 1,000 hours. Perhaps the long piston skirt accounted for the fact that wear in the longitudinal direction was often slightly more than in the transverse direction. The authors, like almost all their customers, were of the view that chromium-plated liners might lead to wear rates that were still somewhat lower. Good utilization was, however, only possible if the chromium layer was comparatively thick, which caused a rather high price for chromium-plating, and made it doubtful whether it would still be economical.

In reply to Mr. Archer the authors said that stiffness of the crank throws in the longitudinal direction was ascertained experimentally in its entirely free condition as well as with the shaft supported in the main bearings. The measured values were practically the same from which it was concluded that the position of an adjacent throw could not have a considerable influence on longitudinal stiffness. As for an assessment of longitudinal vibrations, it would seem at first quite improbable that such a natural vibration with two nodes could occur. The authors failed to comprehend why there should be a natural vibration with a node in the vicinity of the thrust bearing, both halves of the system oscillating in the same phase. In their view, both cases involved a forced longitudinal vibration which was only caused by a torsional critical of the system situated at that particular point. This assumption was substantiated by the fact that the frequencies indicated were between 440 and 600/min. But for a larger engine of about 115 r.p.m. the natural frequency of axial vibrations was, in their experience, situated at about 1,000/min.

The authors' company had still not lost its faith in cast-steel throws for large semi-built crankshafts. It must be admitted that some of the steel firms had committed manufacturing errors which gravely affected reliability. After adoption of the most stringent requirements and inspections, it was concluded that cast throws could be produced in the same quality as forged ones but were considerably cheaper.

Mr. Blomstergren had already commented on the question of savings if the camshaft gear drive was dispensed with.

They also were of the opinion that lateral support of the engine should only be adopted if it proved necessary. So far, they had never encountered any difficulties with engines featuring lateral supports and this even applied to those rare instances where fractures occurred in the supports themselves.

The effective thermal efficiency of the large M.A.N. engines was of the order of 41-42 per cent.

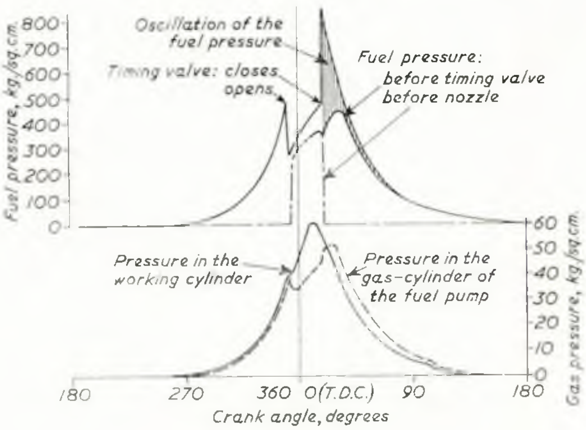
Mr. Alcock's remarks concerning thermal stresses and the influence of strength and thermal conductivity were extremely interesting and touched the heart of the matter. Unfortunately, no tests had yet been undertaken concerning the decrease in heat flow if the air amount increased. This, however, was bound to occur since wall temperatures went down.

The glands of the double-acting two strokes were and would remain a sensitive part even if they were lubricated with high alkalinity oils. The most difficult problem, however, was due to the fact that the double acting cylinder had no possibilities of self-cleaning. The importance of this drawback was shown in the comments on Mr. Pounder's remarks.

The authors agreed with Mr. Coventry that the formula of Professor Kjaer for calculating the stress in a crankshaft deformed in the longitudinal direction afforded only a rough approximation which, however, until now, was sufficient to gain a general idea. The nearer formula given was of great interest to them.

With regard to Mr. Baker's question the authors would like to point out that unloading of the fuel line between timing valve and fuel valve was unnecessary provided this pipe was very short. Secondary opening of the needle need, therefore, not be anticipated. It was correct that injection conditions grew less favourable at smaller loads and if the general condition of the engine deteriorated. Mr. Blomstergren's remarks showed, however, that this phenomenon did not have any unfavourable effect, nevertheless it must be watched closely.

The diagram showing the pressures in the gas pressure injection system was given in Fig. 26. It might be noted from



n = 115 r.p.m.
m.e.p. = 8·46 kg./sq. cm.

FIG. 26—*Gas pressure injection K3Z90/160*

the pressure curve before the fuel valve that needle lift at full load was exact and faultless. The same pressure diagrams were also taken down to ¼ load and resulted in the findings given in the paper. No difficulties were ever experienced with the cylinder lubrication of highly turbocharged four-cycle engines. There was consequently no necessity to fit additional non-return valves into the lubricating passages.

A device for overturning large pistons was at present being evolved by M.A.N. in co-operation with their service engineers. Since in all their engine designs the top piston ring stayed above the edge of a recess in the cylinder liner by approximately 3 mm., it was hardly ever necessary to reface any edges. This only happened if the top ring had failed for some length of time so that the second ring must take over main load.

With regard to Mr. Gröschel's query the authors wished to point out that they had studied the hydraulic drive of large fuel pumps for some time but that it proved too complex and expensive so that they no longer considered it.

Mr. Hopkins was quite right in saying that good quality Diesel oils and high alkalinity lubricating oils would result in a very long running life of the pistons.

The engines of type V8V24/30 (Fig. 23) were supplied for the navy and some of them had been in service now for over three years with very satisfactory results. It went without saying that naval operation never accomplished the same number of operating hours as were encountered in merchant marine duty. Nevertheless, the first plant had already run 3,500 operating hours.

The V12V26,5/30 engine shown in Fig. 25 was also destined for special duties. The mean effective pressure stated, i.e. 19 kg./sq. cm. (270lb./sq. in.) corresponded to a maximum output which, of course, was not used as a continuous rating.

The authors agreed with Mr. Menzies that the classical air injection system was far less sensitive to low-grade fuels than was the case with mechanical injection. For economical reasons, however, air injection was no longer acceptable.

Apart from the aforementioned reasons which precluded a resurrection of the double-acting engine, the severe alternating stresses in a highly turbocharged engine must be taken into account. The connecting rod fracture which occurred in a double-acting M.A.N. engine was primarily due to a forging defect in the fork which had escaped attention despite all manufacturing controls.

In reply to Mr. Hoare, the uniflow-scavenged engine with the four-valve cylinder head showed the most favourable flow areas. On the other hand, however, it involved considerable design complexity with regard to arrangement and drive of the four valves.

The authors doubted whether examination and withdrawal of the piston in a downward direction would in fact constitute any appreciable advantage just as they ventured to doubt the 7,000-10,000 hours indicated as service life of the exhaust valves. All the practical results that had come to their knowledge—even information from the manufacturers themselves—pointed to considerably shorter times between overhauls.

Naturally, the valves in the cylinder undersides of the M.A.N. engines had to be cleaned from time to time, this, in particular, applied to the delivery side. Since periods between overhauls were long and since each vessel, in accordance with rules and regulations, carried a sufficient quantity of replacements, this work had never met with any criticism.

In the event of a faulty turbocharger the latter could be put out of commission by locking the rotor. Since failure of these turbochargers was extremely rare, the authors had, so far, never found it necessary to fit a special bypass line which would be proposed for the pulse operation (three-cylinder group). Apart from this, the turbine casing could be used as a through passage by removing the rotor and fitting two covers. In constant pressure operation a faulty blower could be blanked off completely at both exhaust and air side or removed altogether. With a corresponding reduction in output the complete exhaust volume was then passed through the intact blower or blowers, respectively. Experience showed that the engine would still be able to develop approximately 55 per cent of its rated full load output.

The non-return valves fitted at the end of the lubricating oil bores had performed quite well in several engines belonging to one shipowner, barring one case where the bores and valves were clogged owing to the disintegration of the special-grade cylinder oil. Because of earlier very unfavourable experience the authors dismissed the idea of admitting the cylinder lubricating oil via lateral quills through the cylinder block and the cooling spaces.

They did not think that by using special anti-friction alloys appreciably smaller bearing areas could be achieved, without at the same time somehow affecting the reliability of the bearings. Quite apart from this, further shortening of main and crank bearings was hardly possible, be it only in view of bedplate and connecting rod dimensioning.

Throughout the extended test runs the roller bearings of the engine shown in Fig. 24 had performed excellently so that there was no reason to assume any decrease of hardness.

Replying to Mr. Henshall, the authors said that in the M.A.N. engines, the second-stage manifold was common to all cylinders, i.e. without any partition wall between the individual cylinders. With regard to compression pressures it should be noted that these did, of course, depend to a great extent on the dimensioning of the turbine nozzle sectional areas, in constant pressure operation even more so than in the pulse operation of three-cylinder groups.

The authors entirely agreed with Mr. van Asperen's remarks concerning the various scavenging systems and their relative flexibility. In their two-cycle engines they had, so far, not tried to place the lubricating points very low so that they were located at the bottom dead centre of the piston stroke between the first and second piston ring.

Many owners had voiced the same opinion concerning the possibilities of automation and of staff and crew reduction. The authors also shared these views.

The authors were, of course, also of the same opinion as Mr. Foreman that the first essential was faultless design of the cylinder lubricating parts. This, however, seemed to them less difficult than determining the proper lubricating oil quantity and grade in practical operation.

The lubricating oils used in heavy fuel operation were, nowadays, mostly of the high-alkalinity type (Total Base Number about 35-70). They had never noted that, with the lubricating oil quantities used by them, the alkalinity was not sufficient to neutralize the acids deposited on the cylinder wall. Their experience had not led them to the conclusion that the recently introduced automatic quantity regulation resulted in too scanty lubrication during manœuvring. They were of the opinion that timed injection of cylinder lubrication by means of jerk pumps afforded only negligible advantages so that the considerable design effort involved did not pay off. So far no clear-cut experience existed with regard to the life expectancy of cylinder liners over and above a ten-year period. Allowance must, of course, be made for the fact that heavy operation had not yet been with us for a sufficient length of time and that fuel grades were deteriorating from year to year. Nevertheless, it had been possible to ascertain running times of liners between six to seven years in cargo vessels that had been operating 6,000-7,000 hours per year.

In summing up, the authors wished to point out once more that for years the underlying philosophy of the M.A.N. engine design had been the principle of simplicity. At the same time, however, they were also very aware that in engineering no principle could be followed through uncompromisingly, thus, for instance, there were two answers to the problem of piston design, viz. the long-skirt piston as it was being used in the M.A.N. engine and the short-type piston. The latter, however, involved a valve on the exhaust side. From the technical point of view both solutions were feasible and the final question then was which design the owner preferred: the somewhat higher and heavier engine with long-skirt pistons, or the somewhat lower and lighter version with short pistons, but with the added rotary valve. Obviously the solution finally adopted did not depend on straightforward technical consideration but more or less on the personal views of the shipowner.

The supercharging method as such would not decide the superiority of a Diesel engine. Every scavenging method permitted of a further increase in m.e.p. and output per cylinder over and above the present degree. The limitation was not inherent in the scavenging method, but was due to other factors. In this context the crankshaft of which the measurements were subject to specification by the classification societies should be mentioned first. Any further increase of m.e.p. in the cylinder was useless if it failed to afford a reduction of size and weight in relation to power output. A further limiting factor for increasing m.e.p. was the thermal stress of thick-walled elements and mechanical stress due to peak pressures. Even if the difficulties arising therefrom could be overcome, economical aspects of the design as a whole, i.e. questions of price, might set the final limit.

INSTITUTE ACTIVITIES

Minutes of Proceedings of the Ordinary Meeting Held at the Memorial Building on Tuesday, 28th January 1964

An Ordinary Meeting was held by the Institute on Tuesday, 28th January 1964, when a paper entitled "Recent Development of the M.A.N. Diesel Engine" by Professor Dr. Ing. E. Sörensen and Dr. Ing. F. Schmidt, was presented by the authors and discussed.

Commander F. M. Paskins, O.B.E., R.D., R.N.R. (Chairman of Council) was in the Chair and one hundred and seventy-five members and visitors were present.

In the discussion which followed fourteen speakers took part.

A vote of thanks to the authors was proposed by the Chairman and received prolonged and enthusiastic acclaim.

The meeting ended at 8.00 p.m.

Summer Golf Meeting at the New Zealand Golf Club

The Summer Golf Meeting was held at the New Zealand Golf Club, Weybridge, on Wednesday, 3rd June 1964, and attended by forty-five members.

A Medal Competition for the Institute of Marine Engineers Silver Cup was held in the morning and three scores of 70 net tied for first place. Mr. A. Bartholomew (20) won the competition with the best net score over the last nine holes.

In the afternoon a Bogey Greensome Competition was won by Messrs. H. Armstrong and J. F. G. Arman with a score of 5 up after a tie with Captain R. D. Fielder and Mr. M. MacDermott. The leading scores for both competitions were as follows:

Medal Competition

A. Bartholomew (20) 70	1st Prize
J. F. G. Arman (18) 70	2nd "
E. F. J. Baugh (9) 70	3rd "
H. Armstrong (14) 71	
J. White (13) 71	
R. D. Fielder (14) 72	
W. J. S. Glass (12) 73	
D. Lyon (12) 74	
T. L. Kendall (17) 74	
R. G. Wood (16) 74	
R. M. Hewlett (24) 75	
M. MacDermott (22) 75	
J. F. Watson (2) 75	
H. P. Jones (13) 75	
C. A. Larking (7) 76	
J. M. Mees (18) 76	
E. C. Cowper (18) 77	
R. K. Craig (24) 77	
D. G. Welton (10) 78	
C. Winyard (18) 78	
F. C. Bown (24) 78	

Greensome Bogey Competition

H. Armstrong and J. F. G. Arman	5 up
R. D. Fielder and M. MacDermott	5 up
P. S. Rosseter and A. D. Timpson	4 up
E. F. J. Baugh and R. K. Craig	3 up
J. White and D. G. Welton	3 up
C. Winyard and J. M. Mees	2 up
L. E. Smith and J. F. Watson	2 up
T. L. Kendall and D. Lyon	2 up
C. A. Larking and J. Henderson	1 up
R. D. French and W. J. S. Glass	1 up

Mr. S. Hogg, O.B.E., Honorary Vice-President and Chairman of the Social Events Committee, presented the prizes and thanked the committee of the New Zealand Golf Club for the use of the club and course and members who had donated to the prize fund.

It was announced that the Autumn Meeting would be held at the Grim's Dyke Golf Club, Hatch End, on Tuesday, 29th September 1964, and that the subsequent meeting would be held at the Burhill Golf Club on Thursday, 20th May 1965.

Section Meetings

Colombo

The Third Annual General Meeting of the Colombo Section was held on Friday, 5th June 1964, at the Volunteer Naval Force Headquarters, Kochchikade, Colombo, 5.15 p.m.

The office bearers and Committee for 1964 were elected as follows:

Local Vice-President: C. W. V. Ferdinands
Committee: S. M. B. Dolapihilla
A. L. S. Fernando
Lieut. (E) D. A. G. Fernando, R.Cy.N.
D. C. R. Goonewardene
A. L. Goonewardene
A. J. Hill
B. H. F. Jacotine
Inst. Lt. Cdr. M. G. S. Perera, R.Cy.N.
Honorary Secretary: Cdr. (E) E. L. Matthysz, R.Cy.N.
Honorary Treasurer: L. A. W. Fernando

The membership of the Colombo Section now stands at thirty-four.

Vancouver

Annual General Meeting

The Annual General Meeting and Spring Dinner of the Section was held on Thursday, 20th February 1964, in the Sands Motor Motel, 1755 Davie St., Vancouver, B.C., at 6.15 p.m.

Forty-one members and guests were present.

The report of the Honorary Treasurer was read and accepted.

The following were elected to serve on the Committee during 1964:

Local Vice-President: R. Rennie, M.B.E.
Committee: S. F. Corbett
W. Dey
D. M. Laing
F. C. Salter
Honorary Secretary: R. W. Brown
Honorary Treasurer: J. A. Forsyth

Following the Annual General Meeting a paper entitled "High Output Modern Diesel Engines" was presented by Mr. A. L. Plint (Associate). The speaker who was introduced by

Mr. J. A. Stewart (Member), gave a very interesting paper in which he emphasized the importance of following the manufacturers recommendations as to the maintenance in modern Diesels and resultant consequences if this was not followed.

A lively question period ensued in which many interesting points were brought forward.

Mr. J. S. Logie, B.Sc. (Member) thanked Mr. Plint and expressed his gratification at so much interest shown by the members.

West of England

On Thursday, 21st May 1964, members of the Section together with their ladies, travelled by coach to Coventry where the members visited the gas turbine division of Bristol Siddeley Engines Limited, whilst the ladies toured the Cathedral and the City Council Chambers.

Members were met by the heads of departments of the gas turbine division. Lunch was then served after which two films were shown, giving an account of the various gas turbines in production and their application in air, marine and industrial fields. Details of some of the research which had gone into the development of these projects were also shown.

After the showing of the films a tour of the turbine assembly shops was undertaken where component parts of different classes of gas turbines were closely inspected.

Three different sizes of these turbines were seen during the afternoon, starting with the Gnome, which weighed only 30 cwt., and was capable of developing 1,050 b.h.p. at 19,000 r.p.m., to the Olympus which was capable of generating $17\frac{1}{2}$ MVA when used in that capacity. The better known Proteus was also seen on the assembly line, this unit being particularly suitable for both air and marine application.

It was noted that the Olympus gas turbine was to be used extensively by the Central Electricity Generating Board in multiples of two or four units giving 35 and 70 MVA respectively. These sets were to be installed in remote parts of the countryside for the purpose of helping the grid system in times of peak period electricity loading, and were to be remotely controlled from divisional electricity headquarters.

During luncheon Mr. Allen, Chief Engineer of the gas turbine division, made a speech of welcome to the members, and on behalf of the Section, a vote of thanks to the Company was given by Mr. J. P. Vickery (Vice-Chairman of the Section).

The visit was a great success and the party returned to Bath and Bristol later the same evening.

Institute Awards

Members are reminded that the following awards are now made:

The Denny Gold Medal for the best paper read by a member during the session.

The Institute Silver Medal for the best paper read by a non-member during the session.

The Junior Silver Medal and Premium of £5 for the best paper by a Graduate or Student read before the Junior Section during a session.

The W. W. Marriner Memorial Prize, value £5, given annually to the candidate who submits the Engineering Knowledge paper (Steam or Motor) of the highest merit in the Ministry of Transport examinations for the Second Class Certificate of Competency.

The Extra First Class Engineers' Certificate Examination—Institute Award of a silver medal for the candidate obtaining the highest marks in the Ministry of Transport examination.

The Herbert Akroyd Stuart Award, value £50, available biennially, open to members of all grades and non-members for the best paper read at the Institute on "The Origin and Development of Heavy Oil Engines".

The Yorkshire Award, value £40, available biennially for the writer of an essay or the author of a paper read before the Institute dealing with any development related to any aspect of marine engineering or a product applicable to marine engineering.

A cash prize of £25 awarded annually from the interest on the John I. Jacobs, W. Murdoch, D. F. Robertson and A. Girdwood funds for the best essay on the technical advantages to be gained by taking the Extra First Class Engineers' Certificate course—available to engineers taking such a course at a technical college.

Awards, value £2 2s., are given annually to students of technical colleges in marine centres for the best year's work in the study of heat engines.

Prizes for students taking the Ordinary National Diploma Course under the alternative scheme for the training of seagoing engineers. Two prizes are given each year to each technical college operating the scheme, a prize of two guineas being awarded to the best first year student and a prize of three guineas to the best second year student.

The Frank Roberts Award of books or instruments to the value of £7 10s., given annually to the Student or Probationer Student member of the Institute gaining the highest aggregate marks in the courses and examinations in Phase III of the alternative scheme for the training of seagoing engineers.

Administered by the Institute

The William Theodore Barker Award—£100 annually to the candidate who gains the highest marks in the Ministry of Transport examinations for the First Class Certificate of Competency, provided that such candidate takes the course for the Extra First Class Engineers' Certificate at a technical college.

Election of Members

Elected on 15th June 1964

MEMBERS

Andrew Caie Baxter
Edmund Porter Crowdy, V.R.D., M.A. (Cantab.)
Stanley Crozier
Frank Dawson, A.R.C.S.
Eugene Jerome Fink
Walter Shorey Hayden
Edwin Jones
James Kirkham
Eric Edward Lane
Robert Joseph Morgan
Richard Neil Peter Nielsen
Ernest Wilton Pilling
Stanley George Pratt
Anthony Patrick Rabbit, Lt. Cdr., R.N.
Arthur Grainger Raitt
Charles Edward Percy Simpson

ASSOCIATE MEMBERS

Maurice Amit
Sultan Mahmud Asad
George Martin Baker, Eng. Lieut., R.N.
Rabindra Kumar Biswas
John Giles Carver
Ronald Crossley
Michael William Drew
Edward Royston Evans
Donald Fallow
John Bustard Formby
John Harold Forsyth
Arthur Stanley Hall
Alexander Gideon Henderson
Hugh David McGeorge
Jangbir Bahadur Mehta
Alan Myatt
Brian Maurice Conroy Newbury
Kenneth Nolan
Harry Noel Walton Pickering
Robert Stewart Pryde
Lawrence Patrick Steven Quinn
Agaram Kandadai Ramanujan
Harold Royston Rees
Amitkumar Sengupta
Eric Stephenson

John Syme
Peter Horace Tyson
Prasada Rao Nishtala Venkata

ASSOCIATE
Edward C. B. Barron

GRADUATES
Sajjad Ahmad
Phiroze Framrose, B.Sc. (Dunelm)
Surendra Nath Jha, Lieut. (E), I.N.
Adam Lee
Jonathan Bing Chung Poon
John Charles Price
John Frank Shaw

STUDENTS
William Barry Clegg
Thomas Samuel Leeper
John James Russell Logan
Paul Richard Oliver
Keith John Sparshott
John Woods

PROBATIONER STUDENTS
Alan Cook
David Harold Douglas Owen
James Stirling Sandison

TRANSFERRED FROM ASSOCIATE MEMBER TO MEMBER
Tirloki Nath Bhargava, B.Sc. (Durham)
Derek James Edwin Brooker
Richard Brown
Alexander Clapham
William Alfred Corp
Raymond Leonard Floode
Arthur Louis Henry
Khalid Saifuddin Hyder
Ronald Alfred John Kinsey
George Kirk
George Charles Lusztig
Kenneth Meikle MacKay
Robert McKechnie Nicholson
Ronald Pearson
Philip Simpson Pratt
Charles Leslie Robertson
Keith Ivan Short, Cdr., O.B.E., D.S.C., R.N.
Raymond Frank Surplus

TRANSFERRED FROM ASSOCIATE TO MEMBER
James McGrail Cochrane
Donald Charles Flamank
Frank Henry Walter Johnson
Robert William McConnell
Kenneth Ross
Bernard John Worden

TRANSFERRED FROM ASSOCIATE TO ASSOCIATE MEMBER
Syed Mohed Jamadar, Lieut. (SD) (ME), I.N.
James William Wealleans, B.Sc. (Manchester)

TRANSFERRED FROM GRADUATE TO ASSOCIATE MEMBER
John Addison
Satyapal Bhardwaj, Lieut., I.N.
Michael Ralfs Casey
Allan Dale Craig
Harry Flottiland de Vos
Kraft Lazarus D'Souza
George Ernest Dumbell
Roger Edward Goddard
David Howe
Ashok Babubhai Merchant
David Morter
Edward Arthur Popple
Gude Prabhakara Rao
Swaraj Kumar Sabherwal, Lieut., I.N.
Harpreet Singh, Lieut. (E), I.N., B.Sc. (Hons.) (Delhi)
Colin Taylor
Robert William Wray

TRANSFERRED FROM STUDENT TO ASSOCIATE MEMBER
Andrew Yule

TRANSFERRED FROM STUDENT TO GRADUATE
Michael Patrick Barden

TRANSFERRED FROM PROBATIONER STUDENT TO GRADUATE
Terance Cottrill Procter

TRANSFERRED FROM PROBATIONER STUDENT TO STUDENT
Sean Seamus Hogan
Alistair Kennedy Joynes
David Graham Sharp

OBITUARY

ALEXANDER ROBERTSON EDMISTON (Member 11630) was born in Liverpool on 5th January 1880 and was educated at Merchant Taylors School. He served an apprenticeship in a shipyard at West Hartlepool, followed by some years at sea during which he obtained a First Class Board of Trade Certificate.

In 1907 or thereabouts, he took up an executive position at Smith's Dockyard on Tyneside.

During the First World War, he was precluded from serving in the armed forces for reasons of his health and therefore joined the Ministry of Munitions, where he remained throughout the conflict.

After the war, he returned to Merseyside, joining a marine engineering and consulting firm founded by his late father, James Brown Edmiston, who had been one of the early members of the Institute of Marine Engineers.

In 1924, Mr. Edmiston left Merseyside to become the first works manager of the then newly constructed Palmer's Dry Dock at Swansea. He remained there to become dry dock manager when the company was taken over by the Prince of Wales Dry Dock Co. Ltd. in 1936. During the Second World War, Palmer's dry dock and ship repair yard were heavily engaged in the construction of L.C.M.'s and L.C.T.'s, as well as handling a large volume of dry dock and floating repairs. For his part in this work, Mr. Edmiston was awarded an M.B.E. in the New Year Honours List for 1947.

Following his retirement, he continued his association with marine engineering, first being retained for some time by R. S. Hayes Ltd., Pembroke Dock, during a re-organization of that firm, and later representing Welin-MacLaughlin Davits and Swinney Bros. Pumps, in the Bristol Channel area.

Throughout his career, Mr. Edmiston was widely considered to be particularly successful in the field of industrial relations.

Mr. Edmiston, whose death occurred on 10th January 1964, was elected a Member of the Institute on 5th January 1948 and Chairman of the Swansea Local Section in December 1951. He was also a Member of the Royal Institution of Naval Architects.

FREDERICK EDWARD HALL (Member 8863) died on 18th February 1964, aged seventy-one. He served his apprenticeship with Palmers Shipbuilding and Iron Co. Ltd., from 1908-1913, and joined the P. & O.S.N. Co. in September 1915. He remained with this company until 1954, when he retired.

Mr. Hall was chief engineer of the *Mirzapore* in April 1938, on the Indian coast, and also served as chief engineer of the *Lahore* and the *Kidderpore,* in that trade. In 1941, he was in the *Lahore* when she was torpedoed, and was serving in the *Narkunda* when she was sunk by enemy action in 1942. He subsequently served as chief engineer of the troopship *Empire Fowey* and of the passenger liner *Stratheden,* his last ship.

Mr. Hall was elected a Member of the Institute on 27th March 1939. Predeceased by his wife who died two years ago, he leaves an only daughter.

DOUGLAS HORSBURGH (Member 8345) died on 8th February 1964, in his seventy-eighth year.

He served his apprenticeship from 1903 to 1908, with Harland and Wolff Ltd., after which he went to sea. In 1919, he joined the White Star Line and, in 1937, was serving as chief engineer in m.v. *Georgic.* He was in *Georgic* when she was holed and set on fire by enemy action at Suez in July 1941 and remained with her whilst extensive repairs were carried out. He also served in *Mauretania* and *Caronia.* In May 1950, he retired, after 42 years' service.

Mr. Horsburgh was elected a Member of the Institute on 4th January 1937 and was also a Member of the Liverpool Engineering Society. He leaves a widow.

CYRIL WILLIAM GRIERSON MARTIN (Member 15434) died on 19th March 1964, aged sixty-one. He was educated at Stratford Grammar School and the University of London, and also underwent training in the Chemical Department and laboratories of the Asiatic (Shell) Petroleum Co. Ltd.

He joined the Oil Department of Asiatic Petroleum on 17th August 1920 and, in 1923, was transferred to the chemical side. He then went to Shell-Mex Central Laboratories for a three-year period, returning, in 1929, to Fuel Oil General. In 1954, he became head of Quality Division in fuel and light oils, also Technical Adviser to the management. He was a Division Head in Oil Products Development before his retirement in 1961.

In the forty years of his service, Mr. Martin spent thirty years in dealing with the technical problems of fuel oil and was also a qualified chemist. He represented the company on the Admiralty Fuels Committee from the time it was first constituted in 1948, for the ensuing seven or eight years. He had presented papers to this Institute (*The Influence of Modern Refinery Techniques on Marine Fuel Oil Quality,* published in 1954) also the World Petroleum Congress, and was responsible for the compilation and editorship of the Shell Handbook on Fuel Oil, published in 1946.

Mr. Martin was elected a Member of the Institute on 7th December 1954.

WILLIAM LEYSON STEVENSON MOORE (Member 7807) was born at Peel, in the Isle of Man, on 21st August 1884. He served his engineering apprenticeship with Dunsmuir and Jackson of Govan, and his seagoing time was spent with T. and J. Harrison, between 1910 and 1917. He held a First Class Board of Trade Certificate.

On leaving the sea in 1917, he joined Charles Howson, ship repairers of Liverpool, and remained with that company until 1930. In that year he became superintendent engineer to Manchester Liners Ltd. holding that appointment until 1955.

Mr. Moore was well known and greatly respected in the marine engineering world and he was one of the first to fit Bennis Stokers to Scotch boilers, in a ship owned by Manchester Liners; when the vessel sailed a regular run and could obtain standard quality coal, the Stokers proved most useful.

Mr. Moore was elected a Member of this Institute on 4th February 1935 and had also been a member of the general committee of the Liverpool Marine Engineers and Naval Architects Guild. His death occurred on 5th March 1964.