50-322 OL I-SC-70
9/12/84

NUCLEAR REGULATORY COMMISSION

Docket No. 50-322-OL Official Exh. No. Suffolk County Deisel-70

In the matter of Long Island Lighting Co.

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TRANSACTIONS

JANUARY 1966

Vol. 78 No. 1

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The Development of a Highly-rated Medium-speed Diesel Engine of 7,000—9,000 Horsepower for Marine Propulsion

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The design considerations and development tests are described which have resulted in the production of the Mirrlees-National K Major engine, which has a current commercial rating of 3,000 to 7,500 b.h.p. in 6 to 18-cylinder units, and a projected future rating of 9,000 b.h.p. in 18 cylinders.

The Mirrlees K engine has been well established for over 12 years, some 980 engines now being in service for power generation and marine propulsion. Of these engines, 250 are operating on heavy fuels with viscosities ranging from 200 to 4,600 seconds Redwood I, representing over 650,000 horsepower. The objective in the design and development of the K Major engine has been to increase the specific power output by 50 per cent and at the same time to maintain or to increase the safety factors possessed by the original K engine. These factors, which determine the ability of the engine to operate on residual fuels with low maintenance and high availability, are discussed and the achievement of the objective is illustrated.

Component parts of the engine are described in turn, with details of the methods of measurement of pressure and temperature levels, air flow and wear rates in test rigs and in a prototype three-cylinder engine which was equipped with special features, such as a camshaft with variable timing, to facilitate development work.

The test results obtained on the first 12-cylinder KV Major engine are shown to confirm the performance expected from the rig and prototype engine tests.

INTRODUCTION

In general, the requirements of a marine propulsion engine are:

a) reliability;
b) low fuel consumption;
c) the ability to burn heavy fuels obtained in any part of the world;
d) low lubricating oil consumption;
e) low maintenance requirements;
f) minimum space and weight in keeping with a) and e).

These requirements are obvious but can only be achieved if certain basic principles in design are followed. The paper is divided into sections, each dealing with one aspect of design which affects these overall qualities.

However, before detailing these definite sections, some observations must be made on the application for which the engine is to be used and its suitability for that application. An engine developing 7,000 to 9,000 b.h.p. in 18 cylinders would be ideal for medium-speed marine propulsion since the power range available would be from 3,000 h.p., in a single six-cylinder engine, to 18,000 h.p. with twin 18-cylinder engines. This power range covers a large section of the marine market, as illustrated in Fig. 1, so that if conditions a) to f) can be achieved a worthwhile market should exist for such an engine.

The initial design study showed that the dimensions of the Mirrlees-National K engine (15-in. bore × 18-in. stroke) would fit this power range very well, if the new design embodied the modern features resulting from research and development which would enable high specific outputs to be obtained whilst retaining economy and reliability. At 500 r.p.m., and 200 lb./sq. in., b.m.e.p., this size of engine would give 402 h.p./cylinder, while

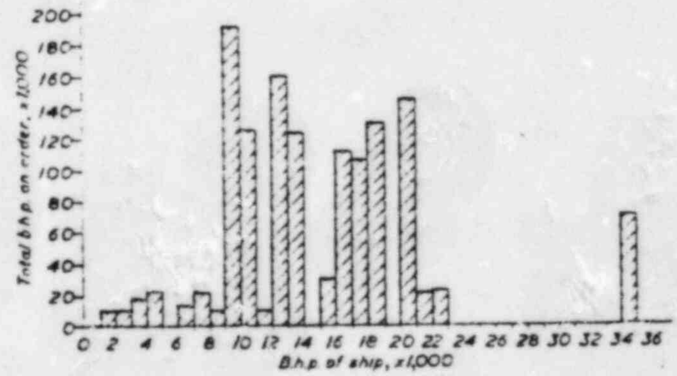

FIG. 1—*Distribution of horsepower for ships over 2,000 d.w.t. on order in Great Britain in March 1965*

at 525 r.p.m., and 220 lb./sq. in., b.m.e.p., 465 h.p./cylinder would be developed. The design of the K Major engine was based on a continuous rating of 250 lb./sq. in., b.m.e.p., at 525 r.p.m., giving 528 h.p./cylinder, and the development programme was planned to achieve this rating, using heavy fuel, in the three stages mentioned.

At the present time, the K Major is released for the commercial market at a rating of 200 lb./sq. in., b.m.e.p., at 500 r.p.m., and development testing for the second stage of 220 lb./sq. in., b.m.e.p., at 525 r.p.m. is well advanced.

A cross-section of the engine, showing its general construction, is shown in Fig. 2 and the details of the design will be dealt with in the following sections of the paper under the headings a) to f) already given.

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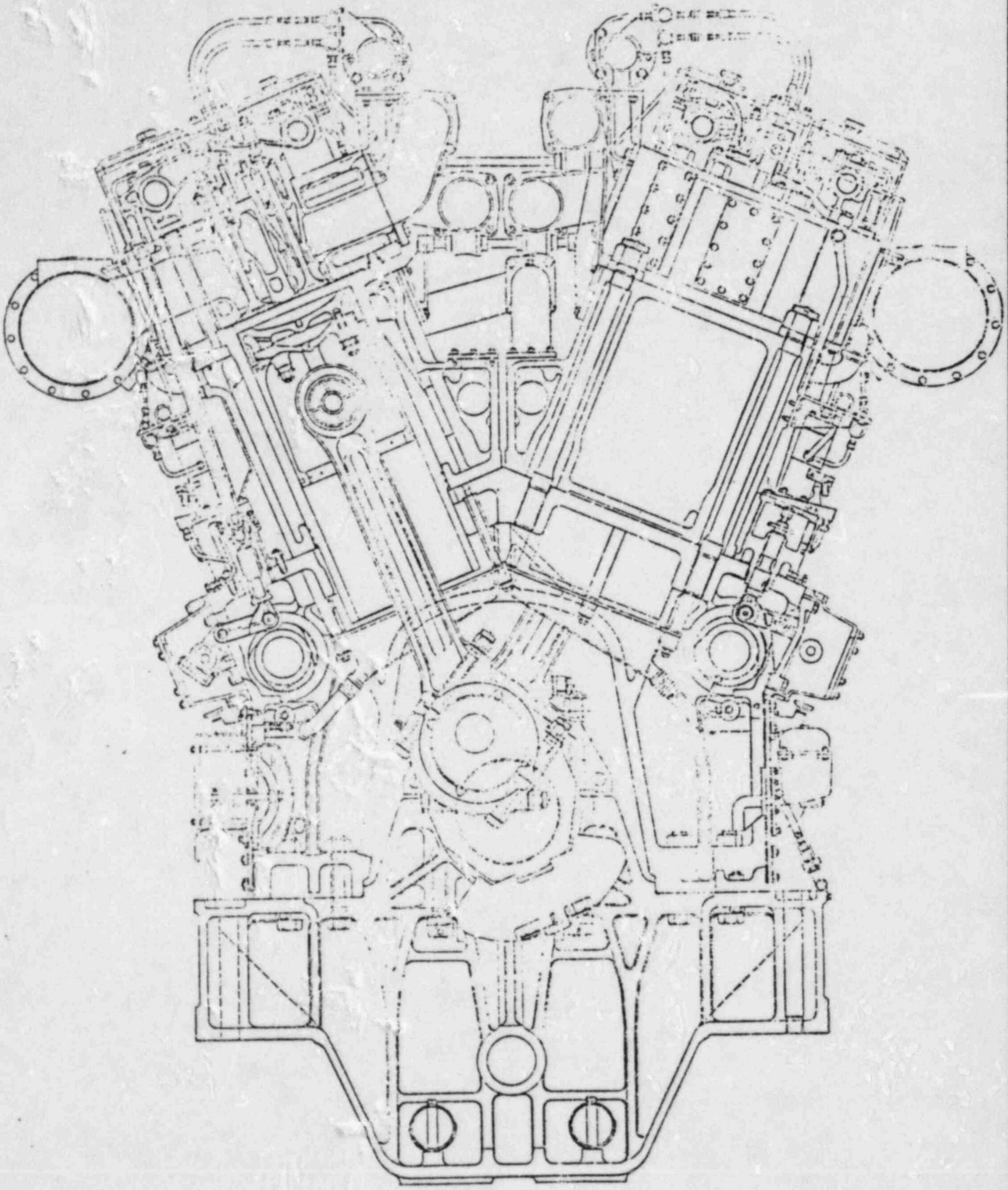


FIG. 2—*KVD Major engine—Cross-section*

a) RELIABILITY

General Considerations

Experience in engineering has shown that one of the surest methods of producing intrinsic reliability in a complex piece of machinery, such as a Diesel engine, is to proceed by a process of logical evolution from one successful design to the next, taking care that the critical parameters, proved in the original design, are maintained in the new design. From the authors' experience of continuous-duty Diesel engines, the critical parameters to be carefully watched are:

TABLE I

Parameter	Mirrlees KV12 engine	Mirrlees KV12 Major engine		
	154 lb./sq. in. b.m.e.p. 450 r.p.m.	200 lb./sq. in. b.m.e.p. 500 r.p.m.	220 lb./sq. in. b.m.e.p. 525 r.p.m.	250 lb./sq. in. b.m.e.p. 525 r.p.m.
Moment of inertia of bedplate (in.[4])	18,650	31,430	31,430	31,430
Maximum internal couple, tons. ft.	210	264	290	290
Ratio, maximum couple ÷ moment of inertia	0·0113	0·0084	0·0092	0·0092
Relative stress in crankshaft	1	0·82	0·91	0·96
Maximum cylinder pressure, lb./sq. in.	1,080	1,350	1,350	1,400*
Cylinder head stud stress, tons/sq. in.	8·3	10	10·1	10·4*
Fatigue strength of threads, tons/sq. in.	17	27	27	27
Ratio, stress ÷ fatigue strength	0·49	0·37	0·375	0·385*
Hertzian stress in fuel cam, lb./sq. in.	262,000	205,000	215,000	225,000*
Main bearing load, lb./sq. in.	814	1,234	1,275	1,330*
Maximum permissible bearing load, lb./sq. in.	1,500	2,500	2,500	2,500
Ratio load/permissible load	0·54	0·49	0·51	0·53*
Large end bearing load, lb./sq. in.	2,400	2,800	2,900	3,050*
Maximum permissible bearing load, lb./sq. in.	2,700	5,000	5,000	5,000
Ratio load/permissible load	0·89	0·56	0·58	0·61*
Maximum stress in piston ÷ U.T.S. of material	0·44	0·33	0·35	0·36*
Top piston ring groove temperature, deg. C.	220	165	185	205*
Exhaust temperature at cylinders, deg. F.	800	810	850	890*
Air flow, lb./b.h.p.-hr.	13·3	13·8	13·9	13·7*
Exhaust valve seat temperature, deg. C.	540	460	490	520*
Specific fuel consumption, lb./b.h.p.-hr.	0·336	0·335	0·336	0·340*
Lubricating oil consumption, lb./b.h.p.-hr. at full load	0·0030	0·0020	0·0020	0·0018*
Weight of engine, lb./b.h.p.	44	30	26	23
Thrust area of piston, sq. in.	138	162	162	162
Maximum thrust pressure on piston, lb./sq. in.	35·8	33·5	34·0	34·8*
Depth of cylinder head, in.	11·75	13·5	13·5	13·5
Injector nozzle tip temperature (fuel at 200 deg.F.), deg. C.	177	127	130	136*
Inlet valve seat wear factor	192	156	156	162*

*Extrapolated values

1) exhaust temperature after the valves should not exceed 820 deg. F. (440 deg. C.) with uncooled valve seats and 930 deg. F. (500 deg. C.) with cooled seats;
2) top piston ring groove temperature should not exceed 430 deg. F. (230 deg. C.);
3) injector nozzle tip temperature should not exceed 350 deg. F. (180 deg. C.);
4) exhaust valve seat temperature should not exceed 1,020 deg. F. (550 deg. C.);
5) lubricating oil consumption should not exceed one per cent of fuel consumption at full load and if possible should approach 0·5 per cent of full load fuel consumption;
6) all bearings should be well within their load-carrying capacity;
7) the stressing of all components, both in fatigue and static loading conditions, should be such that an adequate factor of safety exists.

Some of the more important design and performance characteristics of the K Major engine are compared in Table I with those of the earlier and successful K engine, the comparison showing that safe values of the critical parameters have been maintained and, in many cases, improved.

Piston Design

The control of top piston ring groove temperatures by cooling the underside of the crown of the conventional single-piece cast iron piston, used in the K engine, is acceptable up to a rating of about 180 lb./sq. in., b.m.e.p., using a cast iron having a U.T.S. in the ring belt of 17 tons/sq. in., but, above this load, high tensile thermal stresses are produced on the inside wall of the piston behind the ring grooves[1]. For the K Major engine, a two-piece construction has been developed, as illustrated in Fig. 3, which has a high-tensile steel crown and a "Meehanite" skirt. This design incorporates an inner load-carrying boss, so that no pressure load is taken on the outer wall which carries the rings, and the latter may be quite thin, thus reducing the heat-flow path to the piston rings and giving efficient oil cooling of the ring belt, as well as ensuring that the roots of the piston

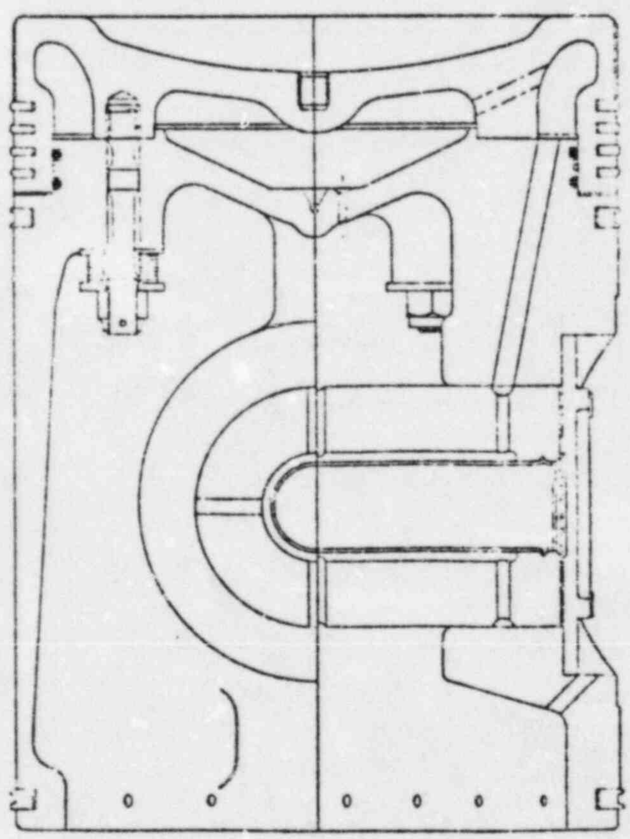


FIG. 3—*Assembly of two-piece oil-cooled piston*

ring grooves are stress-free. The piston crown is retained by four high-tensile studs which have rolled threads to give maximum fatigue strength, and heat-resisting "Helicoil" inserts are used to carry the studs, thus further improving the fatigue strength of the assembly and also acting as a heat barrier for the studs. Disc springs are fitted under the castle nuts to increase the resilience of the assembly and to provide an accurate method of checking the correct pre-load of the studs, this being achieved by measurement of the gap between the two retaining plates for the springs. Lubricating oil is fed, via a drilling in the connecting rod, through the piston pin and to the annulus chamber behind the ring grooves, through which it circulates at high velocity before meeting a transfer drilling to the inner chamber below the piston crown, from where it finally passes down an integral drain drilling in the piston skirt. The returning oil is collected in a cast aluminium tray, supported from the engine column, and is fed through a flexible connexion to a sight-flow and temperature indicator mounted adjacent to the crankcase door.

The thermally-induced and pressure-induced stresses have been thoroughly investigated in test rigs prior to tests in the prototype engine. Fig. 4 is a diagram of the thermal stress rig which is used to simulate the heat flow through the piston which occurs in the engine, heat being supplied by electric immersion heaters using solder as the medium for transferring the heat to the piston crown. Heat transfer through the piston rings is achieved by water-cooling the standard engine liner and oil-cooling of the piston internally is arranged in the same way as in the engine. Thermal stresses are measured by Budd self-temperature compensated strain gauges, having an overall size of $\frac{1}{8}$in. × $\frac{1}{4}$in., so that the effect of the gauges on the heat transfer conditions is

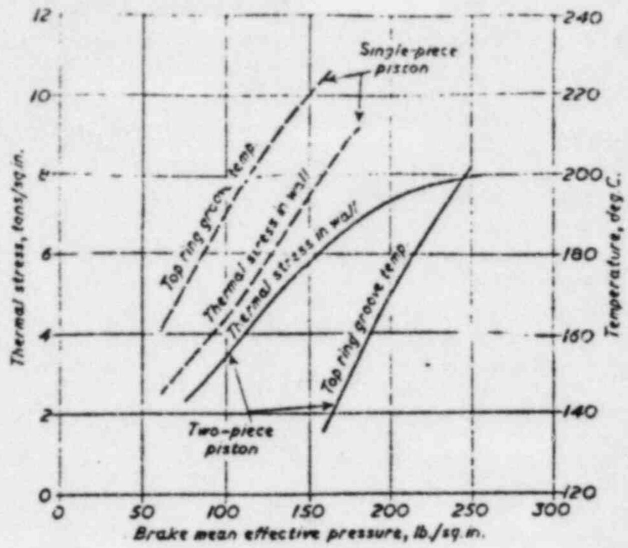


FIG. 5—*Comparison of temperatures and stresses in single-piece and two-piece piston designs*

extremely small. Fig. 5 shows the variation in thermal stress i the piston wall and also the temperature in the region of the to ring groove as a function of brake mean effective pressure fo both the original single-piece piston and for the K Major tw piece piston; Fig. 6 illustrates the temperature and stress distr

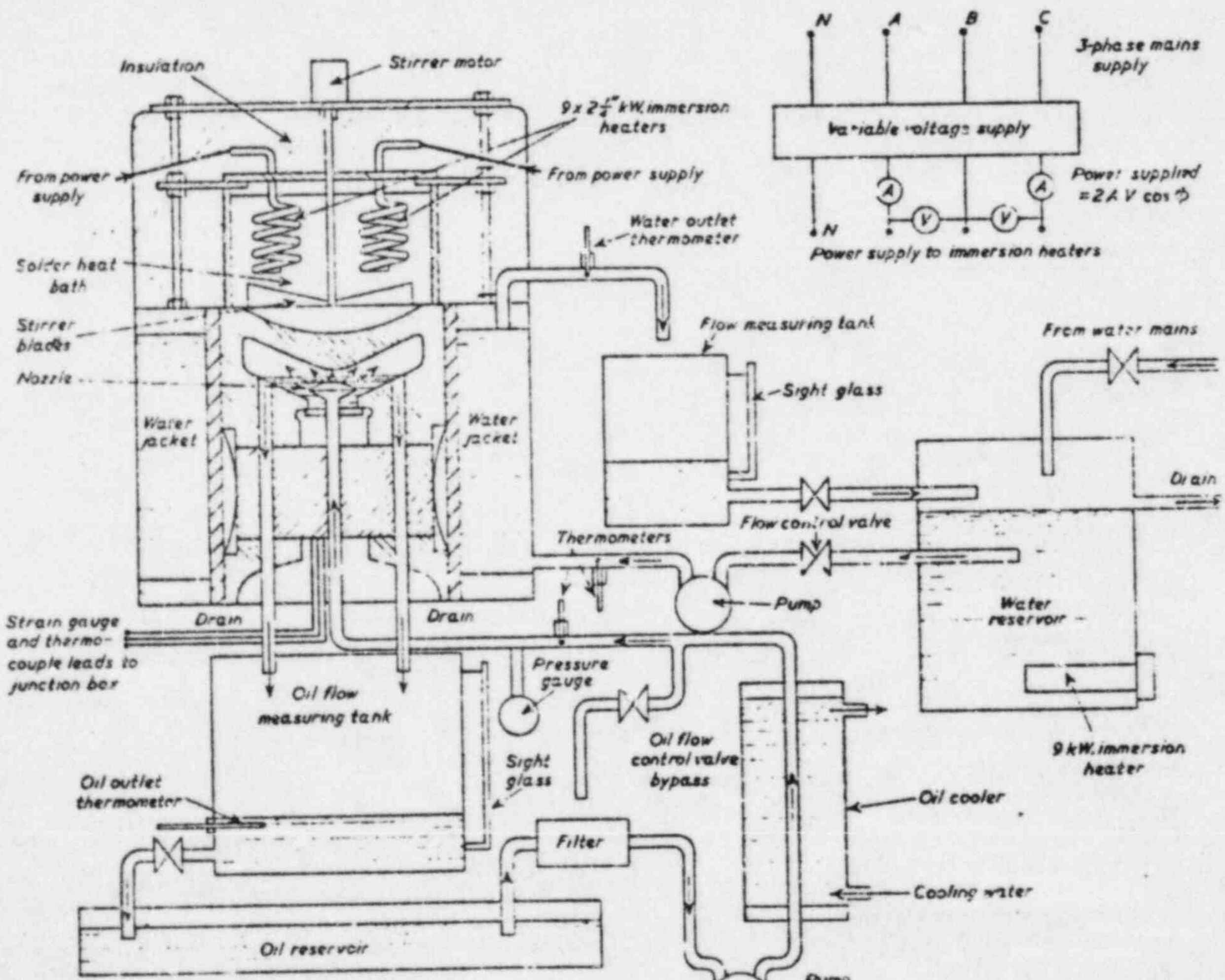


FIG. 4—*Thermal test rig for pistons*

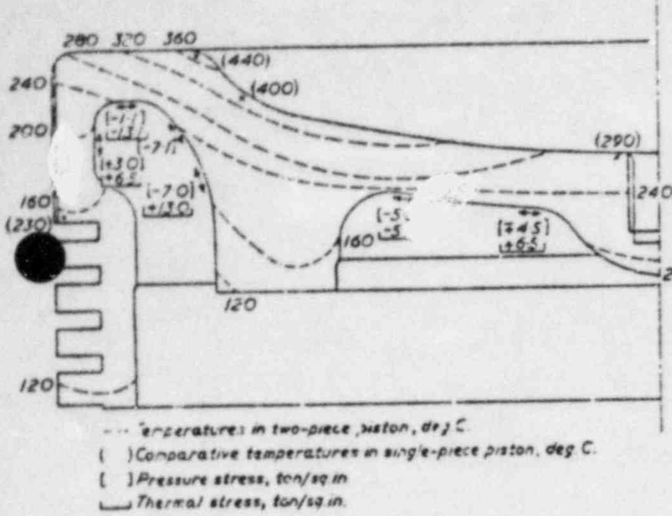
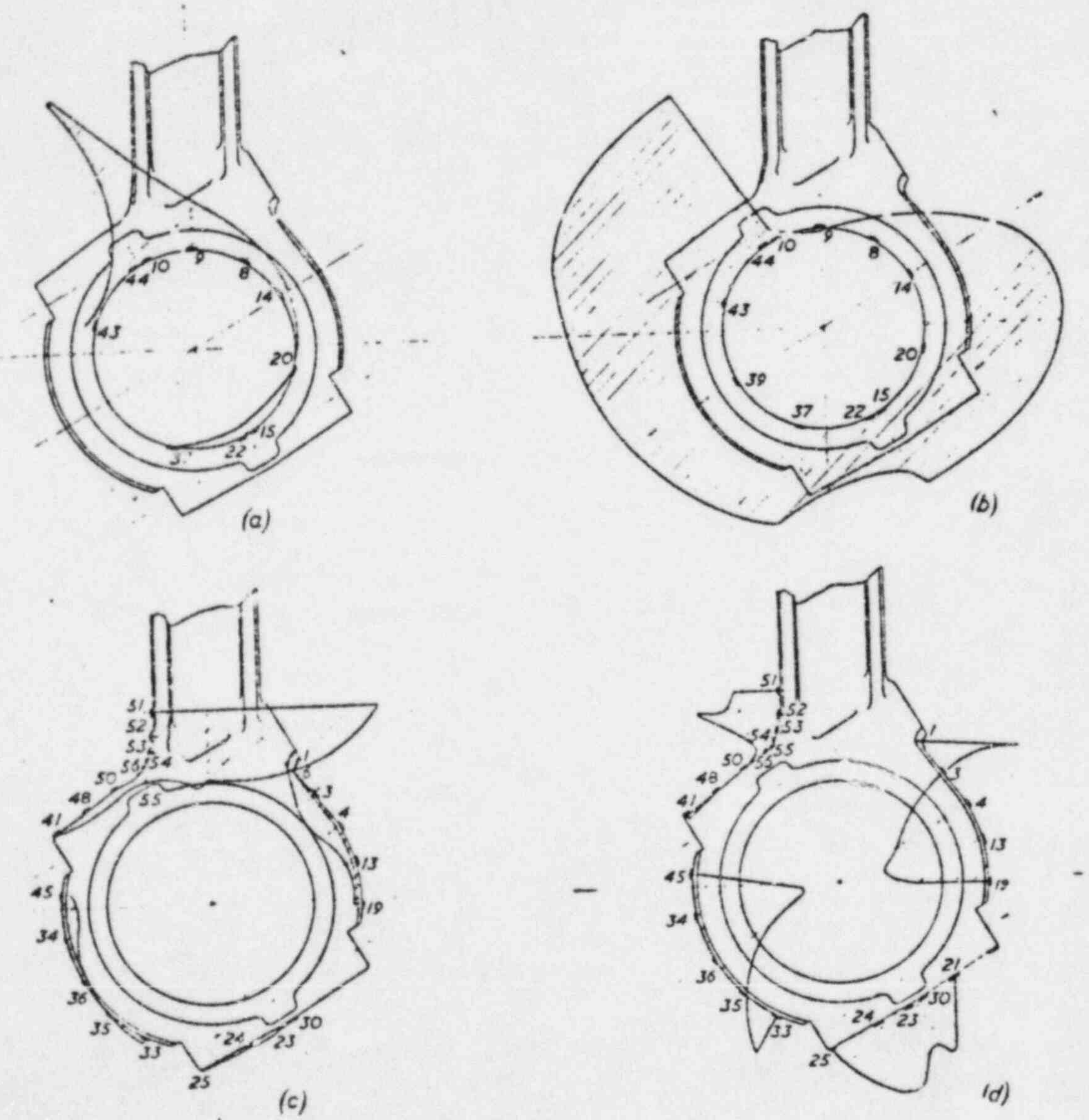


Fig. 6—*Two-piece piston crown temperatures and stresses*

bution in the crown of the two-piece piston. The reduction in top ring groove temperature by some 126 deg. F. (70 deg. C.), achieved by the new design, has made available a wide potential for increase in rating in the future, before any limitation due to lubricating oil break-down is reached. The thermal and pressure stresses are quite modest for the steel crown, which has a U.T.S. of 60 tons/sq. in., at room temperature, so that the factors of safety are much increased over the original single-piece cast iron design.

Connecting Rod

The connecting rods are one-piece stampings with the large-end bearing housing obliquely split at 30 degrees to the rod axis, and carry thin-wall tin-aluminium bearings. This construction permits a crank pin of maximum diameter, consistent with the withdrawal of the connecting rod through the cylinder bore. The optimization of the connecting rod proportions has been assisted by rig tests in a full scale static rig, in which gas loads and inertia loads are simulated by hydraulic pressure and the resulting stresses measured by strain gauges attached to the connecting rod. It was thus possible to reduce the weight of the connecting rod by 15 per cent from that of the original K rod so that, even at



a) Bore stress (gas load)
b) Bore stress (inertia load)
c) Backbone stress (gas load)
d) Backbone stress (inertia load)

Fig. 7—*Connecting rod large-end stress distribution*

the increased speed and load, the connecting rod stresses are lower than in the original design. Fig. 7 shows the stresses in the large-end of the connecting rod under firing pressure and inertia loading, and, in Table II, the safety factors at the most highly stressed points have been listed. In determining these values, allowance has been made for factors which would affect the fatigue strength of the material, such as specimen size effect and surface decarburization where it exists, so that the resulting values indicate the worst conditions and show that the rod design has a large margin of safety.

TABLE II

Gauge No.	Position	Factor of safety
1	Bolt platform radius	3·8
19	Supporting rib	6·8
33	Supporting rib	5·7
51	Base of shank	3·4
55	Shank radius	3·7
59	Bolt platform radius	4·4

Bearings

Main and large-end bearings are thin-wall steel shells lined with tin-aluminium, the increase in bearing loads from the K to the K Major being more than compensated by the improvement in fatigue strength of the bearing material. The actual and permissible bearing loads given in Table I illustrate the increased factor of safety in the new engine, the figures given being the conventional pressures obtained by dividing the maximum bearing load by the projected area of the bearing so that a simple comparison can be made. In the design of the K Major the more accurate methods of calculation, which have been made possible by the use of computers, have been used to assess oil film thickness over the range of speeds and loads so that the true factor of safety is even higher than the simple comparison suggests.

A positive displacement lubricating oil pump is driven from the free end of the engine by a flexible drive and delivers oil through a 15-micron full-flow filter to the main oil gallery cast in the bedplate. An oil-pressure regulating valve is fitted at the engine gallery to ensure that engine oil pressure remains constant, regardless of the degree of contamination of the filter, and a pressure-safety valve at the pump delivery protects the pump in the event of a complete blockage of the system. In addition to the full-flow filter, about five per cent of the flow is bypassed and filtered by small centrifuges mounted at the engine. This dual filtration ensures that carbon and water particles are removed from the lubricating oil and prevents the formation of sludge in the main filters. Tests have shown that a considerable increase in filter life is achieved by this system.

The quantity of lubricating oil circulated through the engine has been determined after thorough development tests to investigate the distribution of oil to main bearings, large-end bearings, piston cooling and other requirements, and the oil quantity has been chosen not only to lubricate but also to cool the main and large-end bearings, thus ensuring that the fatigue strength of the bearing material is maintained at its maximum value.

b) LOW FUEL CONSUMPTION

The importance of adequate air flow in a high-powered Diesel engine cannot be over-emphasized, the air delivered by the turbocharger having to perform the duties of scavenging the cylinder from the products of combustion and of cooling the components in the combustion space region, as well as providing a high mass of trapped air for the combustion process. In recent years the efforts of specialist turbocharger manufacturers to improve turbine and compressor efficiencies have made a substantial contribution to the success of the highly-rated Diesel engine, and the engine manufacturer can play his part by ensuring the maximum utilization of exhaust gas energy and by minimizing flow losses in the porting and ducting.

Air Ports

Air flow tests on the K cylinder head showed that the pressure drop in the inlet passages was made up as follows:

Inlet passage up to valve	11 per cent
Velocity change round valve seat	38 per cent
Loss of velocity head at outlet	35 per cent
Interaction between valves and cylinder wall	11 per cent

The large percentage loss around the valve seat indicate that optimization of the valve head profile and inlet passage shape in this region would be worth while, and the tests also showed that a greater effective flow area could be made available by increasing the valve lift beyond the value of a quarter of valve diameter at which the minimum geometric area becomes constant. Fig. 8 shows the increase in coefficient of discharge beyond the normal L/D ratio of 0·25 and the K Major valve lift was chosen

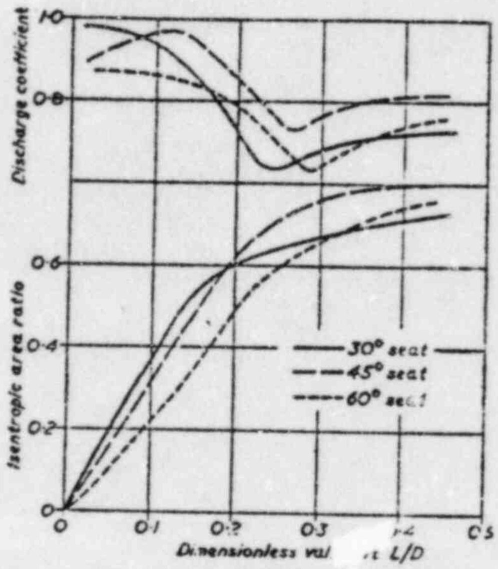


FIG. 8—*Flow characteristics of valves with 30-degree, 45-degree and 60-degree seats*

to be 0·3 of the valve diameter, giving an increase in maximum effective area of just over five per cent. This improvement is quite significant when it is remembered that it is effective over a large valve opening period.

The effect of varying valve seat angle on flow characteristics was also examined and Fig. 8 shows the characteristics of valves of the same throat area with seats at 30, 45 and 60 degrees to the face of the valve. The 60-degree seat valve is clearly inferior to the other two, and the 30-degree seat is the best at small valve openings, whereas the 45-degree seat is best at large valve openings, while the advantage to be gained by increasing the valve lift beyond $L/D = 0{\cdot}25$ is valid for all values of seat angle.

There are other factors to be considered in choosing seat angle which determine the relative merits of 30 and 45-degree seats, of which the most important is that of useful seat life in service. Here, the conditions for inlet and exhaust valves are quite different, and will be considered separately.

The inlet valve operates in a relatively unlubricated condition at the seat so that seat wear, due to the relative movement of the valve seating face against the face in the cylinder head, as a result of the gas pressure, may be quite appreciable. A "wear factor" was derived theoretically and its validity confirmed by rig and engine tests from which the K Major inlet valve head profile was determined to give the minimum practicable relative movement and hence minimum wear(2). The "wear factor" is defined as:

$$F_W = \frac{P_m^2 . N . u . D^4}{E . B . t . v^3 . b . \cos\theta}$$

where μ = coefficient of friction;
P_m = maximum cylinder pressure;
N = engine speed;
D = valve disc diameter;
θ = seat angle;
E = Young's modulus;
B = wear resistance factor (hardness number);
b = seat width;
t = distance from valve disc face to top of seat;
v = height of valve disc cone.

It can be seen that a decrease in θ, or increase in t and v have the effect of reducing the "wear factor" and the K Major inlet valve head profile was designed from these considerations with a 30-degree seat angle and a stiff valve head. From experience on other engines a wear factor of above 250 gives unsatisfactory life in service and a value of 200 is satisfactory. It will be seen from Table I that the original K engine has a satisfactory value, which is confirmed by service experience, and the K Major has an even bigger safety margin.

The criteria for the seat of the exhaust valve are quite different and will be discussed later in the paper under the heading of "Heavy Fuel Operation".

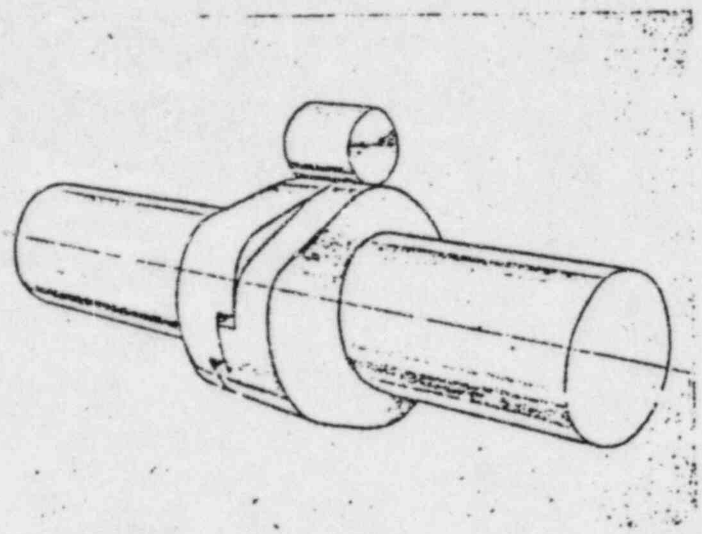


Fig. 9—*Construction of variable timing cam*

Valve Timing

The influence of valve timing on the exhaust, scavenge and charging processes has been examined experimentally on a three-cylinder engine, which was fitted with a special camshaft, in which the timing of both opening and closing of the air and exhaust valves, and of fuel injection were widely variable. Fig. 9 is a pictorial sketch of one of the variable timing cams showing the method by which the valve period is adjusted. Each cam is made in two pieces which are able to rotate relative to each other when hydraulic pressure is applied between the cams and the shaft from a hand pump. Release of the pressure then shrinks the cam on to the shaft to give an interference fit and the two parts of the cam are interlocked to form a bridge over which the cam follower roller can run without any discontinuity of profile. This method of hydraulic mounting allows the whole composite cam to be rotated to any desired position, as well as permitting the opening and closing flanks to be rotated relative to each other. Engine tests have been carried out over a wide range of valve timings, recording overall engine performance and pressure diagrams in the air inlet passages, engine cylinder and exhaust passages, from which optimum cam timings can be determined for any engine speed and load condition.

It will be appreciated that the optimization of valve timing is a complex operation and for a given set of timings it is necessary to match the injection equipment and the turbocharger performance for the engine speed and load range being considered. Some results have been selected from the range of tests carried out on the three-cylinder engine to illustrate the way in which changes in timing can affect the power range over which minimum fuel consumption is achieved.

In Fig. 10, the performance of the three-cylinder engine is shown with all valve timings held constant except the point of exhaust valve opening, the turbocharger match being changed to give the same total air flow. The valve timings were:

	E.V.O. before B.D.C.	E.V.C. after T.D.C.	A.V.O. before T.D.C.	A.V.C. after B.D.C.
Timing A	43 degrees	62 degrees	73 degrees	32 degrees
Timing B	65 degrees	62 degrees	73 degrees	32 degrees
Timing C	75 degrees	62 degrees	73 degrees	32 degrees

The left-hand curves of Fig. 10 show the performance at 450 r.p.m., and since the engine did not have the improved air flow already described in the previous sub-section, the optimum fuel consumption occurs close to the original K rating of 150 lb./sq. in., b.m.e.p. As the exhaust valve opening point is advanced, the position of minimum consumption moves further up the b.m.e.p. scale. This point is more strikingly illustrated in the right-hand curves of Fig. 10 where fuel consumption is plotted against exhaust valve opening point. At the lower rating of 140

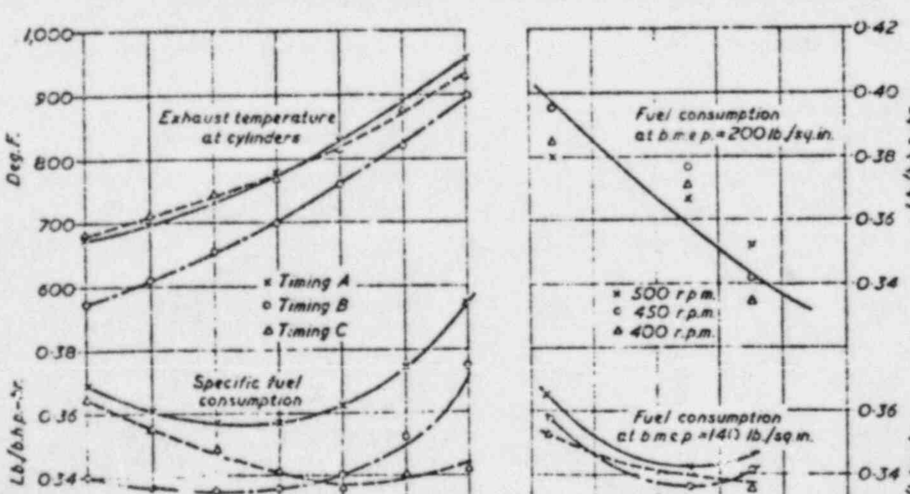


Fig. 10—*Effect of advanced exhaust valve opening point on performance*

lb./sq. in., b.m.e.p., the change in fuel consumption is small but, at 200 lb./sq. in., b.m.e.p., there is a marked reduction in fuel consumption as exhaust valve opening is advanced. It must be emphasized that this illustration is intended to be indicative only of the beneficial effects of early exhaust valve openings. It is important that, at high b.m.e.p. ratings, good thermal efficiency is maintained and in the 12-cylinder K Major engine the minimum fuel consumption occurs at about 200 lb./sq. in., b.m.e.p., as illustrated later in Fig. 14, this result being achieved by improvement in air flow and fuel injection. Fig. 11 shows a low-pressure cylinder and manifold diagram for the 12-cylinder engine at 240 lb./sq. in., b.m.e.p., and 500 r.p.m., and demonstrates the good scavenging and adequate charging of the cylinder which has been obtained. As the development of the engine continues to even higher ratings it will be necessary to move the specific fuel consumption loop still further and the indications from the three-cylinder engine tests are that the advantages of earlier exhaust valve opening will be realized at this stage.

Cam Design

The increase in speed and loading, accompanied by faster opening and closing rates of the air and exhaust valves and the increased lift already described, would be expected to make much

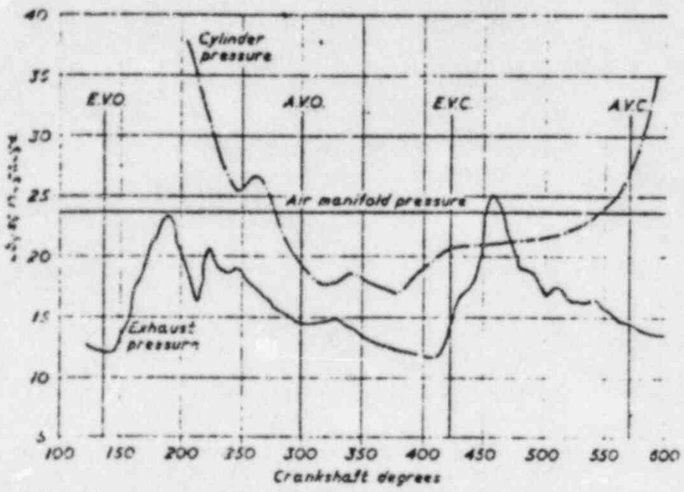


FIG. 11—*Cylinder and manifold low pressure diagrams at 240lb./sq. in., b.m.e.p., and 500 r.p.m.*

greater demands on the air and exhaust cams and follower gear. However, the design of cam profile, to optimize on rates of opening without exceeding established acceleration levels, has been considerably facilitated by the use of computer calculation techniques. The K Major air and exhaust cams are of polynomial profile, the mathematical analysis of the profile by computer calculations making selection of the most desirable curve a relatively simple procedure. The behaviour of the valve gear mechanism under running conditions, to determine the degree and frequency of vibration, has also been programmed and vibration calculations confirmed by a very simple technique attaching strain gauges to the engine push rods. Fig. 12 is typical push-rod strain trace which clearly indicates the natur frequency of the valve gear system and confirms that there is tendency for separation of the valve train to occur.

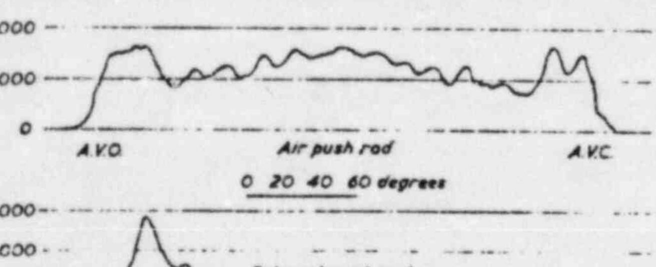


FIG. 12—*Push-rod strain at 200lb/sq. in., b.m.e.p., and 500 r.p.m.*

Fuel Injection

To obtain a maximum rate of injection, the fuel cams a of a profile which gives a constant plunger velocity during t injection period, and the correct matching of the injection equi ment was facilitated by the use of test rigs which enabled t injection characteristics to be determined and the design of t injection equipment to be very nearly finalized before engine tes were started, only the confirmation of nozzle spray angle and t number and diameter of nozzle holes of a predetermined ar remaining for final decision from the performance of the engin

These test rigs enable a large number of permutations of fu cam, pump plunger diameter, delivery valve design, nozz design, etc., to be tested quickly and cheaply, using convention methods of electronic indication of needle lift, fuel line pressu and nozzle sac pressure. The latter has proved to be of conside able importance in ensuring long life of injector nozzles by e plaining the reason for over-rapid deterioration of nozzles in t K engine under certain service conditions. This phenomen was a difficult one to explain until, as a result of calculatio and rig tests carried out by the fuel injection manufactur it was realized that a particular combination of load and spe resulted in a hydrodynamic system in which there was a sudd reduction in fuel pressure in the nozzle sac just before t needle closed, the time interval between the two events bei of the order of a quarter of a millisecond. This resulted in penetration of gas from the cylinder into the nozzle sac duri the combustion process, the hot gases impinging on the botto of the needle and eventually impairing its performance. In F 13 this condition can be seen at (a) on the left, where the pre sure in the nozzle sac has fallen down to a low level at a poi 16½ degrees after spill closure and there is a period of o degree during which the needle is still off its seat and gas c blow past it into the sac. The rig tests now ensure that t seating of the needle occurs before the sac pressure falls, illustrated at (b) on the right of Fig. 13. The value of th preliminary rig work was confirmed by the performance pr duced in the 12-cylinder engine at a very early stage in development running, many hours of "cut and try" tests optimize injection equipment being saved. Fig. 14 show

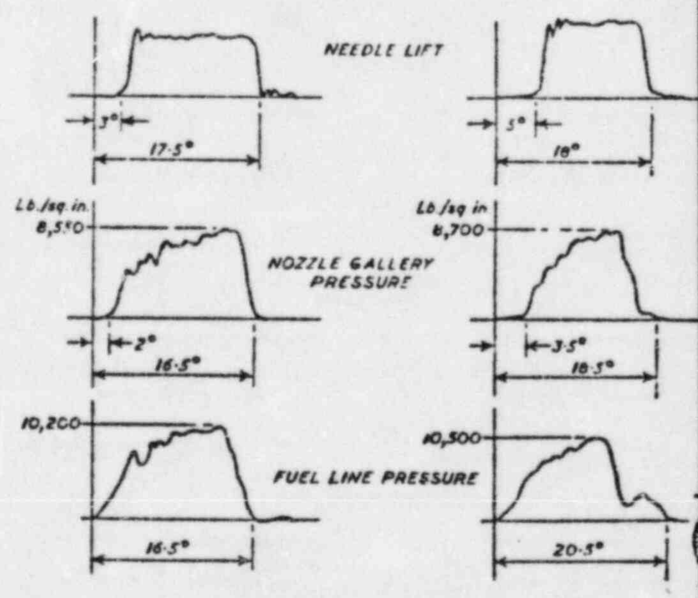


FIG. 13—*Injector needle lift, nozzle gallery pressure and fuel line pressure diagrams*

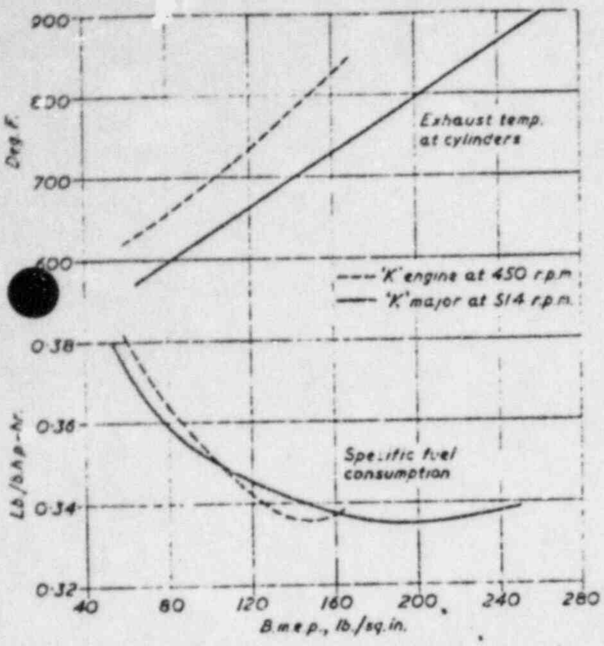


FIG. 14—*K and K Major performance comparison*

the performance of the engine as compared with that of the original K engine, from which it can be seen that the specific fuel consumption of the K Major engine is below 0·34 lb./b.h.p.-hr. over a very wide range of power, i.e., from 140 lb./sq. in. to 250 lb./sq. in., b.m.e.p. The curve also shows that, in spite of the increase in speed, from 450 to 514 r.p.m., and an increase in brake mean effective pressure, from 150 lb./sq. in. to 200 lb./sq. in. (i.e., a power increase of 50 per cent) the same exhaust temperature as in the K engine has been maintained.

c) HEAVY FUEL OPERATION

In the operation of a Diesel engine on heavy fuel, the two items which normally deteriorate most rapidly are the injector nozzles and the exhaust valves, and the frequency of servicing of these two items is of predominating importance. In both cases there is a "threshold" of temperature of the critical parts of the components so that, as ratings increase, the design of the component must be improved to maintain safe operating temperature levels.

Exhaust Valves

Exhaust valve life with residual fuels is usually limited by the formation of deposits on the valve seat, resulting from the incombustible constituents of the fuel and largely from the combination of the sodium and vanadium salts present. As the seat deposits build up, they prevent the valve from making full contact on its seat, thus reducing the degree of heat transfer and eventually allowing tracking across the seat between the gaps in the deposits. The left-hand picture of Fig. 15 shows such a condition for an uncooled valve after 600 hours operation at 180 lb./sq. in., b.m.e.p., on a blended fuel of 300 seconds Redwood 1 viscosity with a three per cent sulphur content, 85 p.p.m. sodium and 100 p.p.m. vanadium. The beginning of erosion across the seat face between the deposits can be clearly seen[3].

Although the chemistry of the formation of these deposits is a most complex study, and is beyond the scope of this paper, field experience and engine tests have shown quite clearly that the presence of sodium and vanadium is of great significance and a practical assessment of the temperature range in which deposits are likely to adhere to the seating face of the valve can be made. Table III gives the melting point of possible deposit constituents which are in the temperature range which may appertain in the seat region of an exhaust valve, as is shown in Fig. 16, where the left-hand valve is of the normal uncooled design corresponding to the left-hand illustration of Fig. 15.

TABLE III

Compound	Melting point (deg. C.)
Nickel vanadate $NiO.V_2O_5$	900
Sodium sulphate Na_2SO_4	880
Sodium orthovanadate Na_3VO_4	850
Vanadium pentoxide V_2O_5	675
Sodium pyrovanadate $2Na_2O.V_2O_5$	640
Sodium metavanadate $NaVO_3$	630
Sodium vanadyl vanadate (1.1.5) $Na_2O.V_2O_4.5V_2O_5$	625
Sodium vanadyl vanadate (5.1.11) $5Na_2O.V_2O_4.11V_2O_5$	535

From the Diesel engine designer's point of view, it is sufficient to accept that if the valve seat temperature can be kept below about 1,020 deg. F. (550 deg. C.), adhesion of any of these components will not occur to any appreciable extent so that rapid build-up of the deposits will not be possible. The problem is thus quite different to that of the gas turbine engineer, who has to consider the corrosive effect which occurs at higher temperatures. However, the achievement of low valve seat temperatures at high outputs is not easy and calls for careful attention to details of design and patient development engine testing to achieve the desired result. The three-cylinder prototype engine, shown in Fig. 17, has been used for continuous testing on heavy fuel in the research laboratory for the past three years and more

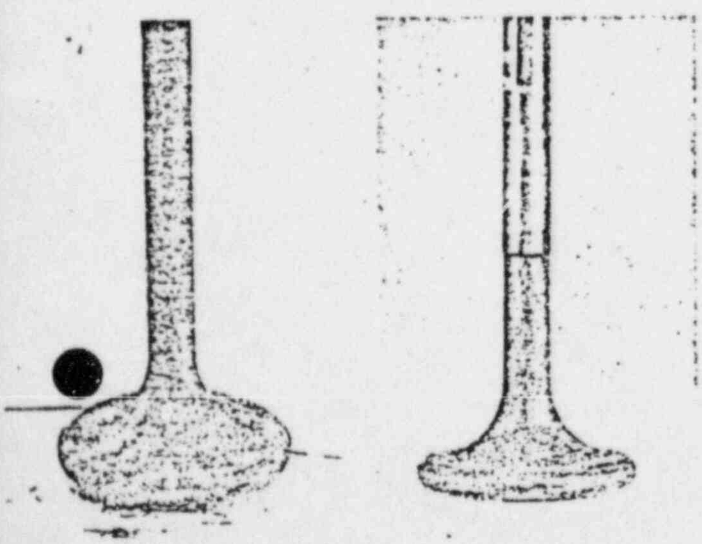


FIG. 15—*Comparison of exhaust valve condition after operation on heavy fuel*

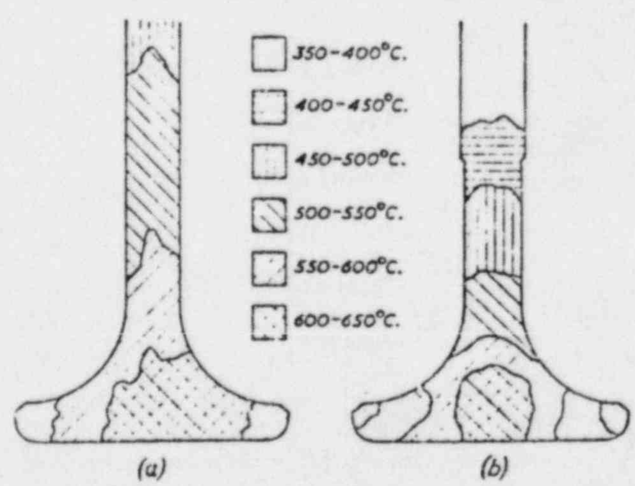


FIG. 16—*Temperature distribution in exhaust valves with uncooled and cooled cages*

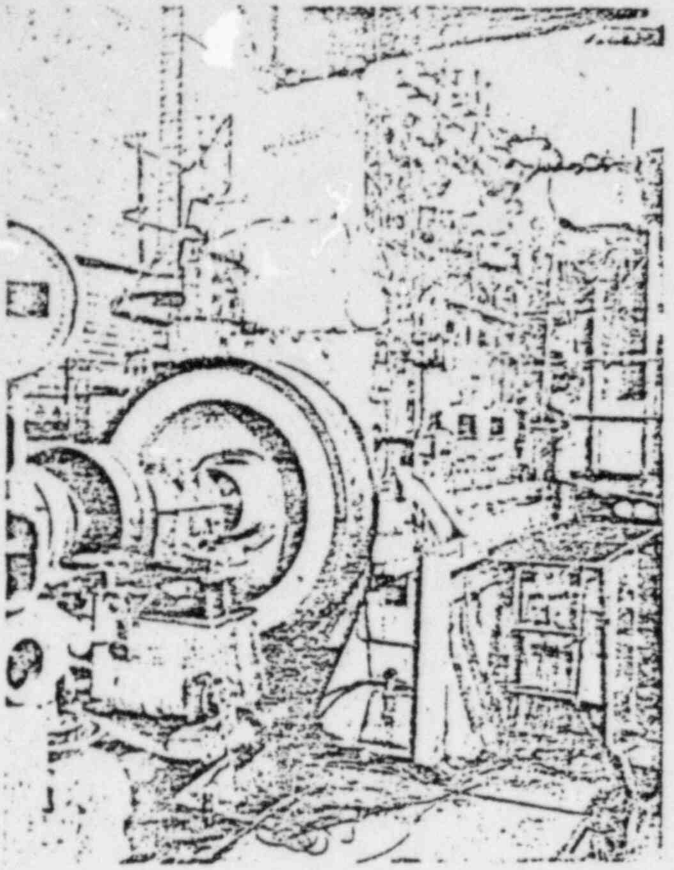


FIG. 17—*Prototype three-cylinder development engine*

than 60 exhaust valve and cage design combinations have been tested, of duration between 300 and 1,300 hours each to determine the effect of different factors in the design. A basic test duration of 500 hours at 200 lb./sq. in., b.m.e.p., loading was chosen and valve seat condition as the main parameter, together with other features such as valve guide wear, was compared with a reference design which was maintained throughout. In many cases the test was prematurely stopped at about the halfway stage where the valve condition was not satisfactory and, in the later stages, valve assemblies were replaced in the engine without re-grinding for second and third runs. Space does not permit a detailed report of the individual tests, but the resulting K Major exhaust valve and cage design will be described in detail to illustrate the factors which were found to be important.

Fig. 18 shows the valve and cage and it can be seen that a cooling passage is provided in the cage close to the valve seat. The seat is made of a single piece of Stellite 6, grooved to form the lower part of the cooling passage, and the upper portion is machined in the valve cage which is a three per cent Cr-Mo steel casting, the two being welded together by electron beam welding. The Stellite portion thus provides the facility for simple rebuilding of the seat by oxy-gas deposition of Stellite after a long period of time in service. The valve stem is of increased diameter and the valve of high-conductivity Cr-Ni-Si steel, to assist heat transfer from the head of the valve through the stem, and the valve guide is also surrounded by a water-cooled space, the cooling water passing along a drilled passage from the top of the cage direct to the annulus around the seat, then through a drilling to the space around the guide and via another drilled hole to the outlet at the top of the cage. The heat transfer from the valve to the cooled guide is assisted by the close fit between the valve stem and guide, the previously mentioned development tests having shown that a diametral clearance of 0·012in. resulted in an uncooled guide temperature of 450 deg. F. (232 deg. C.) in its middle position, with rapid deterioration of the lubricant and the formation of hard carbon. Halving the clearance reduced the temperature to 380 deg. F. (193 deg. C.) and, halving it again together with stem lubrication and guide cooling, brought the temperature down to 170 deg. F. (77 deg. C.), i.e., about 10 deg. F. (6 deg. C.) higher than the cooling water temperature, with no deterioration of lubricant and a guide wear rate of less than 0·0005in. in the first thousand hours.

The valve is fitted with a rotator in the top spring carrier which helps mechanically to prevent build-up of seat deposits, but its most important function is to ensure an even temperature distribution around the valve so that there is no local high-temperature region. The flow lubrication of the valve guide takes advantage of this rotation by using the valve itself as a timing device. Two flats are provided on the valve spindle which periodically line up with oil inlet and outlet drillings on the guide as the valve rotates. The linear positioning of the slots only allows the oil to pass while the valve is open and thus the oil space around the valve is only pressurized when the exhaust pulse pressure is present in the valve cage gas passage, the oil acting as a seal against gas penetration up the stem and the exhaust pressure preventing leakage of oil from the guide. It was at first feared that a continuous oil supply to the guide might result in excessive leakage of oil from the bottom of the guide, but this has not proved to be the case and, in fact, the tendency is for the leakage to be upwards as the retardation when the valve meets its seat is greater than the acceleration during opening and the inertia of the oil carries it upwards. This intermittent pressure lubrication of the valve stem makes it possible to use a very small stem/guide bore clearance without any risk of valve sticking, and this helps the heat transfer from the valve to the water-cooled guide. In addition, the danger of stem or guide bore corrosion at low load running conditions is avoided.

Since the stem to guide clearance is important in the heat transfer process, the reduction of guide wear helps to maintain low valve seat temperatures over a long period in service, and many of the development tests were concerned with valve guide material and valve rocker lever geometry to this end. The long guide and the small overhang of the valve head beyond the guide will be noticed in the illustration and were found to be important factors in reducing guide wear, as was the composition of the special "Mechanite" iron which was finally used for the guide material.

Sodium-cooled and water-cooled valves were tested among the many combinations but were found to offer no advantage over the design finally adopted, mainly, it is thought, because of the difficulty, with an internally-cooled valve, of providing cool-

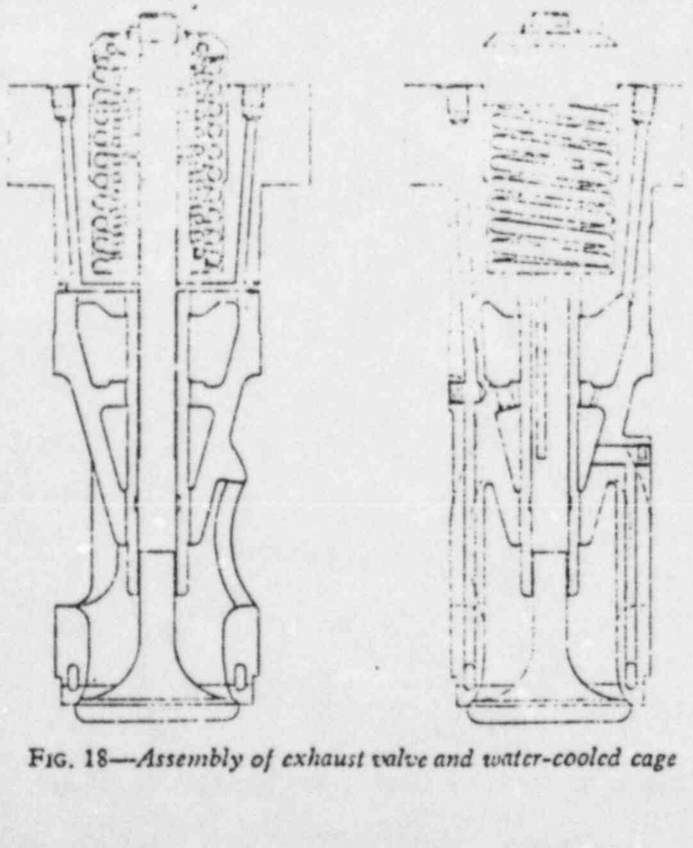


FIG. 18—*Assembly of exhaust valve and water-cooled cage*

ing passages close enough to the actual seat of the valve. The usual methods of drilling down the centre of the valve stem, although successfully cooling the centre of the head, still leave a fairly high temperature at the seat, and in the case of the internally water-cooled valve, the water connexions to the valve are a difficult problem.

The right-hand valve of Fig. 15 shows the results of this development, the valve having run for 900 hours at 200 lb./sq. in., b.m.e.p., on the same type of fuel as before. The good condition of the seating face shows that no re-grinding is necessary and the valve can operate for a much longer period without attention. The corresponding temperature distribution in the valve head is shown in the right-hand illustration of Fig. 16, and the effect of rating on exhaust valve seat temperature is given in Table I.

The valve development tests also included investigations into the effect of fuel treatment on exhaust valve life and while one fuel additive showed promise, in that the nature of the valve seat deposits was altered, it was not effective enough to justify its adoption. The principle of this additive was that other chemicals were added to the fuel so that the compounds, which were formed during combustion, would have higher melting points than those listed in Table III. It seems likely that, with further development work by the additive manufacturers, there may be some advantage to be gained in the future from this type of additive. Water washing of the fuel, to remove the sodium content, was found to be quite effective and the sodium could be reduced from 90 p.p.m. to about half of this value without difficulty, engine tests showing that the washing had quite an appreciable beneficial effect on the exhaust valve seat condition. As can be quickly calculated from Table III the critical sodium/vanadium ratios in the important temperature zone range from 1:0·74 to 1:13·3, the lower melting point compounds being associated with the latter end of the range, so that a reduction in sodium content may tend to produce the compounds with the lower melting points and, with particular fuel compositions, have an undesirable effect. Thus, with the wide variation in constituents in fuel from different parts of the world, it is difficult to make a clear case for water washing of the fuel.

Injectors

Fuel injection nozzles, when operated at high temperatures, tend to form carbon around the holes in the nozzles, known as "trumpeting", which may interfere with the injection spray pattern and reduce combustion efficiency, thus aggravating the temperature problem. For a time, the carbon formation develops until the "trumpets" become detached from the nozzle and a periodic rise and fall of exhaust temperatures can often be seen as this occurs. The general trend of temperature, however, is upwards and conditions eventually level out at the top end of the exhaust temperature cyclic range. In more extreme cases of high temperature, the needle seat may lose its hardness and the needle rapidly hammers its way into the seat. The temperature at the nozzle tip can be measured by thermocouple and a temperature of about 356 deg. F. (180 deg. C) is considered to be the limit

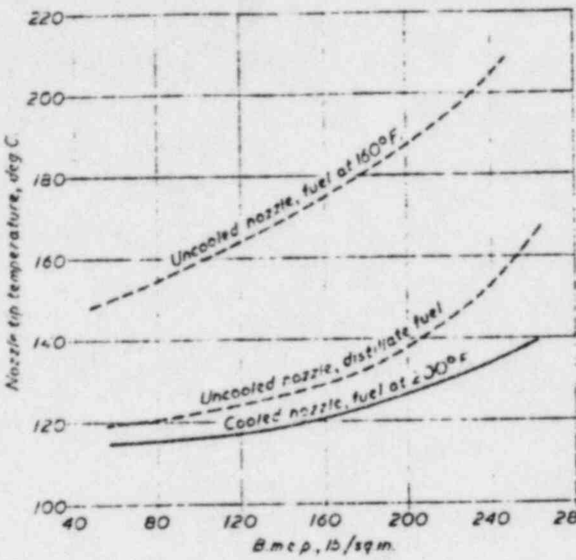


FIG. 19—*Injector nozzle tip temperatures with cooled and uncooled injectors*

for satisfactory operation. In Fig. 19, the middle curve shows the variation of nozzle tip temperature with load for the K Major engine, using an uncooled nozzle and distillate fuel, where the fuel itself has a considerable cooling effect, and there would be no difficulty in operating an uncooled nozzle on this type of fuel up to a load of about 280 lb./sq. in., b.m.e.p. In the upper curve, however, blended fuel of 300 seconds Redwood I viscosity was used, with a fuel temperature of 160 deg. F. (71 deg. C.) and it can be seen that the loss in cooling effect from the fuel has limited the acceptable load level to about 180 lb./sq. in., b.m.e.p., and with heavier, and hence hotter, fuels the load limit would be much lower. A water-cooled nozzle is therefore necessary for high ratings on heavy fuel, and the lower curve shows the tip temperature for a cooled nozzle using 1,000 seconds fuel at 200 deg. F. (93 deg. C.) with cooling water at 150 deg. F. (66 deg. C.). It is important that the nozzle should not be over-cooled as cold corrosion can occur at temperatures below 230 deg. F. (110 deg. C.), but this is controlled by the water-circulation system which is separate from that of the engine-cooling water. Fig. 20 shows the cooling system which is a closed circuit serving the injectors and water-cooled seat exhaust valve cages with a thermostatically controlled bypass around the heat exchanger and minimum volume in the system to ensure that correct operating temperatures are reached quickly.

d) LUBRICATING OIL CONSUMPTION

The consumption of lubricating oil in a Diesel engine is an important factor in maintenance costs and it is not always realized that, at a reasonable consumption rate of one per cent of

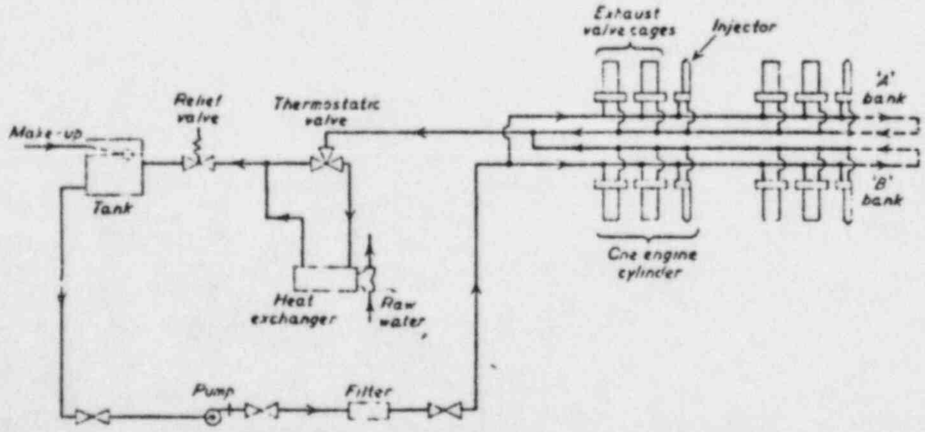


FIG. 20—*Arrangement of injector and valve cage cooling system*

fuel consumption, a 4,000 h.p. engine will burn a quantity of lubricating oil equivalent to its sump capacity in a period of the order of 300 hours. Emphasis is often laid on long periods between oil changes which are extended by an engine with a high oil consumption, whereas the relative importance of oil consumption to oil change period is around 50 to 1. The cost of modern high-duty detergent oils is quite appreciable, so that an oil consumption of one per cent of the fuel consumption represents something like ten per cent of the fuel bill. Not all of this could be saved, of course, but a reduction of 50 per cent in lubricating oil consumption is equivalent to a five per cent saving on the fuel bill, and would be well worth having from the point of view of running costs.

Piston Ring Design

To carry out lubricating oil consumption tests in the relatively short running periods of 500 hours or so in the research laboratory, it was necessary to develop an accurate method of measuring top-up rate and a system was devised, and has proved very successful, whereby consumption can be measured consistently over successive two hour periods and plotted consecutively. The running-in period and the levelling-out to a steady consumption can now be followed and it has been possible to obtain steady state results after a total test period of only 300 hours, which allows much more latitude for testing variations on a ring pack than was previously the case. There is a large number of detail points to be considered such as liner finish, roundness, drainage in the piston, etc., but the basic concept which has been established is to provide a parallel-faced chrome-plated top compression ring, three taper-faced plain compression rings, a relatively mild scraper ring below the gudgeon pin, and a more severe scraper ring above the pin. This ensures that adequate lubrication is available around the body of the piston but that the minimum of oil is allowed to pass up into the combustion space. The consistency of oil consumption measurement has enabled some interesting facts to emerge, and Fig. 21 illustrates one of these—the effect of the wall pressure of the scraper ring above the pin. The left-hand curve is from the three-cylinder, 15-in. bore prototype engine, and the right-hand curve from a

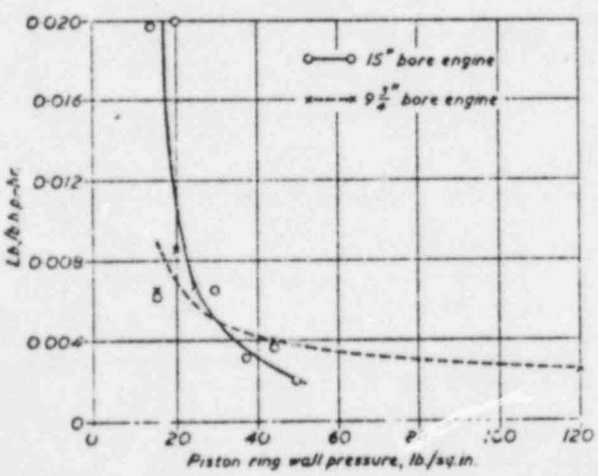


FIG. 21—*Variation of lubricating oil consumption with scraper ring wall pressure*

completely different high-speed engine of 9¾-in. bore, the points marked being the stable lubricating oil consumption achieved after running periods of about 300 hours in each case. Both curves show the same trend of reducing oil consumption with increased ring wall pressure and the tendency for the curves to level out at higher values of wall pressure. The value of wall pressure necessary to achieve a satisfactory consumption can be seen to be much higher for the smaller high-speed engine than for the K Major engine, and in the case of the smaller engine it was necessary to use a spring-loaded conformable scraper ring to achieve the desired consumption. A conventional type of slotted scraper ring was adequate to provide the 50 lb./sq. in. pressure needed for the K Major engine, the resulting consumption, of less than 0·002 lb./b.h.p.-hr., being confirmed in the 12-cylinder engine during development running.

Piston Ring Quality

Consistent oil consumption and low wear rates are largely dependent on the quality of the piston rings, from the point of view of metallurgical structure as well as accuracy of manufacture. Accuracy and good finish in manufacture can be assured by conventional inspection methods, and such methods can easily be extended to give some indication of material quality, such as by measuring the permanent set of the ring at a given load value above that required to close the gap. A simple sample checking method on metallurgical structure was devised in which a small piece of ring is clamped with its working face subjected to a given load and resting on the surface of a ring of liner iron. The ring is then rotated at a standard speed for a fixed time without lubrication and the weight loss of the piece of ring measured. Weight loss is used as a measure of the relative wear resistance of the material and, although "rough and ready", is found to co-relate well with the differences in microstructure of the ring material. Some typical results are given in Table IV, and illustrated in Fig. 22, and show that with the same Brinell hardness, increasing amounts of free ferrite give progressively worse results and these are not improved by increase in phosphorus content within the amounts to comply with mechanical strength requirements[4].

TABLE IV

Sample No.	Structure	Hardness, HB	Weight loss, gm.
A	Greatly undercooled graphite, considerable free ferrite (centricast) 3·15 per cent. T.C., 0·83 per cent P.	210	0·404
B	Some undercooled graphite, a little free ferrite (centricast) 3·20 per cent T.C., 0·40 per cent P.	210	0·185
C	Random uniform medium flake graphite, fully pearlitic (sand-cast) 3·45 per cent T.C., 0·55 per cent P.	210	0·017

e) MAINTENANCE

Engine running times between overhauls are dependent upon the load, duty and running conditions, and the preceding sections have indicated the attention that has been paid to the components which operate under the most arduous conditions. By reducing the critical temperatures of injector nozzles and exhaust valve seats, so that when operating on heavy fuels the temperatures are below the "threshold" values at which deterioration becomes rapid, it has been the aim to achieve periods of 2,000 to 3,000 hours before servicing of injectors or exhaust valves is necessary. Experience on the prototype engine has indicated that this ambition is by no means unreasonable but, of course, true confirmation of success will only come from the accumulation of service experience. Maintenance of other components would not be different from that established over many years, e.g., piston removal annually, complete overhaul every two years, the periods generally being dictated to suit the convenience of the operator rather than by the demands of the engine.

f) SPACE AND WEIGHT

In achieving high engine ratings reliably, the weight per horsepower, and space per horsepower, are naturally reduced and the emphasis on reliability for commercial marine work necessitates a different approach from that which would be appropriate for naval work where light-weight constructions become necessary but short life may be permitted. Sight should not be lost of the importance of low fuel consumption in the conside

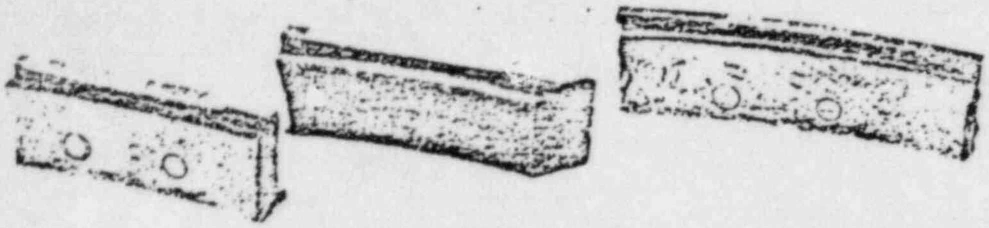


FIG. 22—*Comparison of piston rings after wear rig tests*

tion of weight. A ship refuelling every 3,000 miles, for example, at an average speed of 15 knots, and having engines weighing 34 lb./b.h.p., and a specific fuel consumption of 0·34 lb./b.h.p.-hr., would re-bunker an amount of fuel equivalent to twice the weight of the engines. Thus a five per cent reduction in fuel consumption would be equivalent to a ten per cent reduction in engine weight in addition to the saving in fuel cost.

In the design of the K Major engine, cast iron has been used as the main structural material and, in the authors' experience, has many advantages over fabricated steel designs. Few fabricated structures are able to avoid fillet welds in load-carrying regions and the fatigue strength of such a weld is as low as ± 1·2 tons/sq. in. Even butt welds must allow for discontinuity so that their fatigue strength is only ± 3·8 tons/sq. in., these values being for good quality welds, the strength of an imperfect weld being, of course, very low indeed. A good quality cast iron has a fatigue strength of over 5 tons/sq. in., and as well as freedom from the notch sensitivity, which so drastically reduces the fatigue strength of a steel structure, cast iron has good internal damping properties and also possesses the useful property of a diminishing E value with increased stress so that stress concentrations are considerably reduced and the material tends to relieve itself of any excessive stresses.

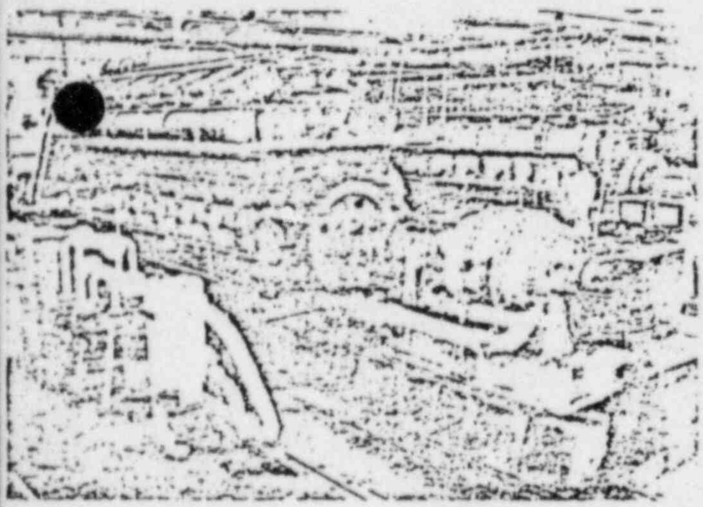


FIG. 23—*Prototype KV Major 12-cylinder engine*

In keeping with the philosophy of designing for maximum

TABLE V

Cylinder bore	15in.
Stroke	18in.
Compression ratio	11.35:1
Maximum r.p.m.	525
Minimum running r.p.m.	125
Continuous rated b.m.e.p.	200 lb./sq. in.
Maximum continuous b.h.p./cylinder	420 b.h.p. (426 cv)
Lubricating oil inlet temperature	150 deg. F. (65 deg. C.)
Lubricating oil outlet temperature	165 deg. F. (74 deg. C.)
Lubricating oil drain tank capacity	650 gal. (2,960 litres)
Fresh and salt water flow rates	5.5 gal./b.h.p.-hr at 50ft. head (25 litres/cv-hr.)
Engine cooling water inlet temperature	155 deg. F. (68 deg. C.)
Engine cooling water outlet temperature	170 deg. F. (77 deg. C.)
Exhaust temperature after turbocharger	800 deg. F. (427 deg. C.)
Starting air pressure	400 lb./sq. in.
Specific fuel consumption	0·335 lb./b.h.p.-hr. (l.c.v. of 18,400 B.t.u/lb.)
Thermal efficiency	42 per cent

TABLE VI—POWER RANGE

	B.m.e.p. lb./sq. in.	No. of cylinders					
		6	8	9	12	16	18
B.h.p. output at 250 r.p.m.	200	1,200	1,600	1,800	2,400	3,200	3,600
	250	1,500	2,000	2,250	3,000	4,000	4,500
B.h.p. output at 350 r.p.m.	200	1,680	2,240	2,520	3,360	4,480	5,040
	250	2,100	2,800	3,150	4,200	5,600	6,300
B.h.p. output at 450 r.p.m.	200	2,160	2,880	3,240	4,320	5,760	6,480
	250	2,700	3,600	4,050	5,400	7,200	8,100
B.h.p. output at 525 r.p.m.	200	2,520	3,360	3,780	5,040	6,720	7,560
	250	3,150	4,200	4,720	6,300	8,400	9,450
Overall length of engine		20ft. 0in.	24ft. 0in.	26ft. 0in.	24ft. 3in.	29ft. 10in.	32ft. 7in.
Overall width of engine		7ft. 6in.	7ft. 6in.	8ft. 2in.	11ft. 7in.	11ft. 11in.	11ft. 11in.
Overall height of engine		11ft. 6in.	11ft. 6in.	11ft. 6in.	11ft. 11in.	11ft. 11in.	11ft. 11in.
Height above crankshaft C.L.		8ft. 9in.	8ft. 9in.	8ft. 9in.	8ft. 2in.	8ft. 2in.	8ft. 2in.
Engine weight (dry) tons		38	41	48	65	85	95

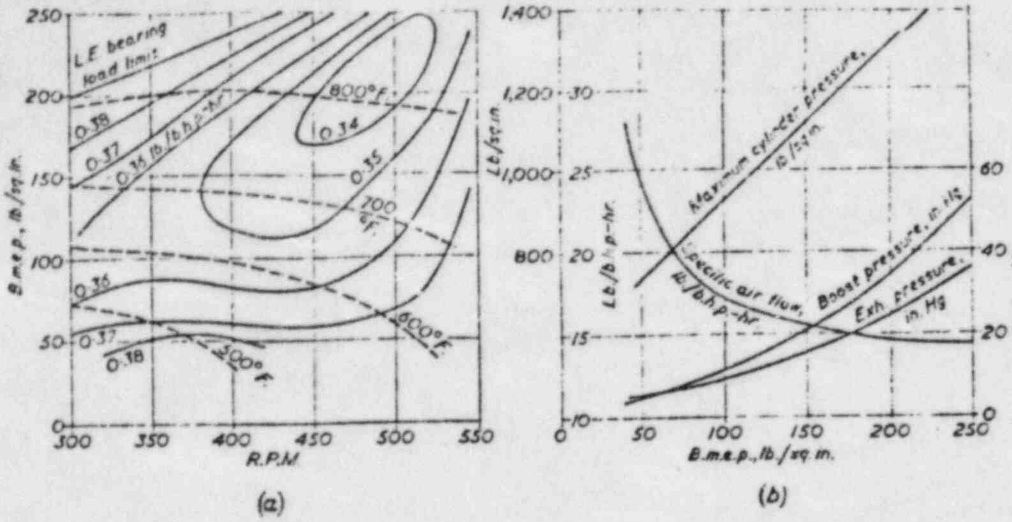


FIG. 24—*KV Major engine performance characteristics*

reliability and easy maintenance, the bedplate type of construction has been retained for the K Major engine and a useful facility has been added by the inclusion of a machined strip on the top surface of the bed so that alignment of the engine can be quickly and accurately checked, and crankshaft deflexion measurements more easily interpreted.

Fig. 23 shows the 12-cylinder KV Major on the test bed, from which the general construction and appearance of the engine can be seen, and Tables V and VI give the specification for the range of engines available at the current commercial rating of 200 lb./sq. in., b.m.e.p., and the future rating of 250 lb./sq. in., b.m.e.p., under normal temperature and pressure conditions with sea water up to 75 deg. F. (24 deg. C.) to the charge air cooler.

CONCLUSIONS

The design and development of a highly-rated medium-speed Diesel engine, to operate economically and reliably on heavy fuels, has been described and it has been shown that, for the K Major engine, the critical parts of the engine, which determine its reliability, have adequate safety margins for its current rating of 200 lb./sq. in., b.m.e.p., and have potential for a substantial increase in rating, to 250 lb./sq. in., b.m.e.p., in the future. The performance of the 12-cylinder prototype engine, beyond the current commercial rating, is illustrated in Fig. 24, curve (a) showing the performance at variable speed and curve (b) the performance at a constant speed of 514 r.p.m., and these, in conjunction with Fig. 14, show clearly the enormous strides which are being made in the Diesel engine industry towards higher specific outputs without exceeding the temperature and pressure levels which past experience has shown to give reliable and trouble-free operatio[n]. There is little doubt that in the marine propulsion field there [is] considerable interest in the use of medium-speed Diesel engin[es] for higher powers than have hitherto been possible, and that wit[hin] the next few years engines of this type will be available [to] cover almost the whole range of power demands of Briti[sh] shipping.

ACKNOWLEDGMENTS

The authors wish to express their thanks to the Board [of] Mirrlees National Limited for permission to publish the inform[a]tion contained in this paper and acknowledge the assistan[ce] given by Bryce Berger Ltd. and by their colleagues in t[he] compilation of data.

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Discussion

MR. R. COOK, M.Sc. (Member of Council) said that, at the present time, the manufacturers of medium-speed engines in Great Britain were making very strenuous efforts to extend their share of the marine market in propulsion machinery. The paper was, therefore, timely and few who had read it would have failed to be impressed by the manner in which the authors and their colleagues were applying the latest knowledge and research techniques to the solution of the problems which arose when such machinery was developed to operate at high ratings on residual fuels. One could hardly doubt that success would attend their efforts, although he suspected that the large direct-drive Diesel would be about for quite a few years to come.

The histogram shown in Fig. 1 was interesting. It would be noted that by far the largest horsepower on order at the present time was between 9,000 and 21,000 s.h.p. per ship. With the machinery described in the paper this implied the use of some 18 to 36 cylinders of 15 in. diameter, each with two exhaust valves, two inlet valves, together with the injection and starting equipment. He said that he could not help wondering whether the modern seagoing engineer, who was perhaps not quite so amenable to long and arduous hard work as his forebears, would take kindly to the never-ending task of top-overhauling such a formidable number of cylinders.

Another point for thought was the effect which such maintenance requirements would have upon the reduction in engine-room staff now being achieved with direct-drive machinery by the application of an increased degree of automatic control. He hoped that some superintendents would comment on these aspects later in the discussion.

He said that some years ago Dr. Pope had made a very thorough theoretical and experimental investigation for the British Shipbuilding Research Association into the causes of failure of pistons, liners and cylinder heads in marine oil engines, with results which had since been published in the Transactions of another Institution. He was not surprised, therefore, to see the attention which the authors had given to thermal and pressure-induced stresses in the design of the two-piece, oil-cooled piston. Presumably, the piston temperatures shown in Fig. 6 were measured on the actual engines and the rig used to check the thermal stresses calculated from these temperature measurements. If so, he wondered whether good correlation was achieved. It would be interesting to know how the temperature distribution in the stationary-rig piston correlated with that on the engine.

He said that he was interested to observe the use of rolled threads on the high-tensile steel studs used for securing the piston crown. Work by B.S.R.A. on the rolling of threads of large mild-steel bolts such as those used in the dynamically-loaded components of direct-drive Diesels, had shown very striking improvements in fatigue strength. Reference to a paper* appearing in the Transactions of the Institute four years ago would show that form-rolling increased the fatigue strength of large forged bolts made from mild steel some 2½ to 3 times when compared with cut-thread specimens. Rolling of thread roots gave almost as great an increase. The degree of rolling had been found to be not very critical, but his Association was at present investigating more fully the optimum degree for various sizes and pitches. He imagined that with the high-tensile steel material used by the authors, the gain in fatigue strength would not be so great as in the case of mild steel, but it would be interesting if the authors would quote some figures. Form-rolling could be a very cheap method of bolt production, particularly in small sizes.

The means adopted to ensure correct pre-loading of studs was to be commended since there was no doubt that the majority of failures of dynamically-loaded bolts were due to fatigue caused by inadequate tightening. It was not always appreciated that, with a properly-designed bolted connexion, fatigue failure was virtually impossible if the bolt was adequately pre-loaded.

The section in the paper dealing with heavy fuel operation was, of course, of the greatest possible interest, since a solution of the difficult problems involved was essential if the medium-speed engine was to be able to compete with the direct-drive engine, which was so much less fastidious as to its diet. Here again, the authors had given evidence of a careful scientific approach which should go a long way to ensure success. They had commented on the possible use of fuel additives. One could imagine this approach being successful where fuel supplies of constant composition were available, but this was seldom possible in marine practice and the chances of obtaining a cheap additive, which was effective with a wide variety of fuels, seemed somewhat remote. The authors' approach, by tackling the design, was certainly the right one. Their remarks on the drawbacks of water washing were also worth noting.

No reference had been made in this section to sump-oil contamination when using heavy fuels. Presumably this must occur to some degree in this trunk-piston design, and it would be useful if the authors were to give some information on the procedure involved in maintaining the lubricating oil in a suitable condition.

Dr. Pope had, over the years, made many investigations into the properties of cast iron. Few were, therefore, more familiar with its strength and frailties. Sir Harry Ricardo had once referred to cast iron as "the material which served our forefathers so well for lamp posts and kitchen ranges", but he was sure that Sir Harry would be the first to acknowledge the advantages which the authors had enumerated. Its use as the main structural material in the K Major engine had much to commend it, since weight was rarely of paramount importance in merchant ships.

On the subject of cast iron, he said that it might be inferred from the data given in Table IV that centri-cast piston rings were inferior to sand-cast rings. He felt sure that this would not be the authors' intention. Centri-cast rings had been widely employed with success. He took it that the authors' purpose had been simply to show that, with this type of material, undercooling and consequent presence of free ferrite was most undesirable.

The paper had touched in an interesting manner on so many aspects of Diesel design that to point to ommissions might seem somewhat churlish. He wished, however, that the authors had found it possible to touch on the subject of turbo-

* Cook, R., and McClimont, W. 1961. "The Influence of Screw Forming Methods on the Fatigue Strength of Large Bolts". *Trans. I.Mar.E.*, Vol. 73, p. 417.

charging. Perhaps they might, at some later date, find it possible to give a paper on their experiences in turbocharging up to the 250 lb./sq.in. in b.m.e.p. which was involved in the third stage of the development of the engine.

COMMANDER E. TYRRELL, R.N. (Member), in a contribution read by Mr. T. P. Everett, referred to the successful introduction, by the authors' company, of an engine which he considered met both industrial and marine needs which so far had only been filled by Britain's foreign competitors. As the authors had so rightly said, reliability was the main requirement of a marine propulsion engine, and nobody who had read this paper could fail to be impressed by the systematic way in which Mirrless National had carried out the research and development work necessary to ensure that this engine would operate satisfactorily at the ratings envisaged.

If the medium-speed geared Diesel engine was to compete with the slow-runing direct-coupled engine, it was essential that it should operate satisfactorily while burning heavy fuel, and it was evident that the authors had taken very considerable trouble and had expended a comparatively large sum of money in trying to ensure that this would be the case. There was, however, one point which he felt should have been mentioned in this respect, and that was the effect of various grades and compositions of lubricating oil on the problems associated with the burning of heavy fuel. There was little doubt that trunk-piston engines called for careful selection of the lubricating oil if satisfactory operation with heavy fuel was to be achieved. New types of lubricating oil and testing for quality and make-up by the addition of detergents, anti-oxidants, and alkalis could have a profound effect on the satisfactory operation of trunk-piston engines while burning this type of fuel. The wrong type of lubricant, or one which had been allowed to deteriorate unduly, could give rise to ring-sticking and crankshaft corrosion. The operating temperature of the oil was also important if these defects were to be avoided. Perhaps the authors would like to remark on the type of lubricating oil and its optimum operating temperature for this type of engine burning heavy fuel.

He thought that the title of this paper was slightly misleading. He could not agree that the Mirrlees K Major engine should be regarded as highly rated when operating at the conditions given in the paper. In his opinion there was considerable scope for further advances in b.m.e.p. These were important as they should give worthwhile reductions in the cost per horsepower. Introduction of a reliable engine operating at these higher brake mean effective pressures would do much to increase the competitive power of this type of engine against its competitors abroad and other types of prime mover. In this technically-competitive world, the main object of any Diesel engine manufacturer must be consistently to uprate his engines in order to give better value for money. He must at the same time retain reliability.

Many of those present would be aware that the Ministry of Technology had recently placed a contract with the Yarrow-Admiralty Research Department to investigate the use of medium-speed geared Diesel engines as propulsion units for ocean-going merchant ships. This survey was now almost complete. He thought that it was true to say that the results of this survey would give encouragement to those manufacturers of medium-speed Diesel engines who thought that the medium-speed Diesel had a future and could compete in many ships with the slow-running direct-coupled engine. The report showed that every type of ship and trade must be treated on its merits, but that a shipowner who failed to carry out a detailed economic survey into the possible use of medium-speed Diesel engines, as an alternative to the slow-running direct-coupled engine, did so at his peril.

MR. S. H. HENSHALL, B.Sc. (Member) said that, as an engine builder of medium-speed engines, he found that he was on the side of Dr. Pope in a lot of the things he had said. The paper, however, had been very stimulating and he would like to ask several questions about it.

With regard to Table I, the specific air flow for the 250 lb sq.in. b.m.e.p. showed a drop compared with lower b.m.e.p., and although this drop was only a small one, he would have though it desirable to go on increasing the specific air flow.

Turning to the piston design, he said that the features o it were, in many ways, those with which he agreed, but it wa mentioned that it was a steel crown and a cast iron junk. Th steel crown probably had a higher coefficient of linear expan sion as compared with the cast iron. This meant that there wer some problems in its connexion, for instance, it must mak life a little difficult for the sealing rings between the tw portions. Cold clearance between piston crown and liner mus be increased.

Figs. 2, 3 and 6 showed a double line round the junk. H wondered what this signified and whether it was some devic to overcome the cold clearance problem.

With regard to exhaust valves and operation on heavy fuel the importance of losing heat via the stems of the valves wa certainly to be considered. In the paper the clearance wa mentioned as being of importance. The wear rate was obvious kept down by the ingenious device of continuous lubricatio and he said that he would be interested to know what was t maximum allowable clearance of the stem to guide and wh sort of life the valve had in this respect. Also, it appeared th the guide could be renewed, although he was not sure wheth it was intended to be renewed.

He suggested that there was an argument for not wat washing fuel, in that the deposits also occurred on turbocharg blades, and turbochargers could be water washed more easily the sodium was allowed to remain in the fuel.

On the question of cast iron or steel as the main structu material, he said that steel had its own advantages, and stru tures could be designed with low stresses where welds occurre Modern techniques of manufacture and inspection could ensu good quality welds.

He said that surely greater reliability resulted from a desi in which the major loads did not have to pass through a jo between the crankcase and bedplate, and easy maintenance w not confined to the bedplate type of construction. The princi of a machined strip used for checking alignment of the eng quickly was also used on engines of fabricated design hav underslung crankshafts and light sumps instead of baseplate

IR. E. R. GRÖSCHEL said that he proposed to limit his o servations and comments to the fuel injection side of the pap departing only for a moment in order to fully endorse authors' statement under "a) Reliability: *General Conside tions*", where they said that the surest method of produc intrinsic reliability was to proceed by a process of log evolution from one successful design to the next, taking c that the critical parameters proved in the original design w maintained in the new design. This statement deser thunderous applause from both engine manufacturers and eng users, particularly marine engine users.

The use of test rigs for fuel-injection equipment devel ment was, of course, fully appreciated and valuable data rega ing performance and life of the equipment might be gai However, care should be taken, when applying results obtai from injection-equipment test rigs to engine conditions, un injection into a pressurized medium was strictly simulated. wondered if this was done in the case where nozzle ga pressure was found to be lower than the prevailing gas pres with a needle still open which permitted gas entrance into nozzle gallery. The cure adopted, he ventured to guess, wa lighten the reciprocating mass of the injector.

His company, having always designed and manufact their own fuel-injection equipment for their medium-s Diesel engines, had always been protagonists of the low in injector, i.e. having needle springs acting directly upon needle without the intermediary of a push rod. Of course spring was thus placed into a somewhat uncomfortable pos (heat and space), but unorthodox spring wire sections h with the space and chromium/silicon wire with the tempera

The "hydrodynamic condition" referred to in the paper, he thought, was a spill wave receding too fast, which the normal oscillating system of the injector was incapable of following. A stiffer spring might help in border-line cases. A trick, imparted to him some time ago by Mr. J. F. Alcock (and gratefully acknowledged) was to watch for gas bubbles in the injector leak-off connexion; should gas pass into the nozzle, it would generally pass through the cylindrical lapped part of the needle-nozzle bore causing lacquering of the lap fit and would finally appear in the leak-off pipe, where a plastic tube would facilitate observation. He said that, acting upon this recommendation, his company sometimes used development engineers as "bubble watchers".

He thought the authors should be congratulated on having such confidence in the precision of cathode ray oscillogram interpretation. He had tried something similar and the result had varied between ½ degree and 2 degrees cam angle, depending upon the thickness of the pencil point used and the condition of the interpreter.

He asked if the line pressures in Figs. 13(a) and 13(b) referred to full load conditions. If they did, they were remarkably low. Also, the difference between maximum line pressure and maximum gallery pressure was rather large, being 1,650 lb./sq.in. in Fig. 13(a) and 1,800 lb./sq.in. in Fig. 13(b). He wondered if the edge filter (which, after all, was the major throttling component between line and gallery) could be responsible for this pressure discrepancy. Wall friction could hardly account for it with the customary maximum flow velocities of about 90 m./s.

Dealing with the chapter on heavy fuel operation, he said that he was intrigued by the statement that the lack of cooling ability of preheated fuel should be responsible for high nozzle tip temperatures. The fuel temperature given was 160 deg. F. (71 deg. C.) for a blended fuel of 300 sec. Redwood 1 viscosity; gas oil, not preheated, would reach the nozzle, after passing through the compressive cycle of the fuel pump, not very much cooler. He ventured to think that the hotter nozzle tip of the heavy fuel operated engine was more the result of the slower burning of the heavy fuel, with a resulting larger heat rejection to, and heat absorption from, the nozzle.

He said that he understood that the mechanism of trumpet formation was a function of the lighter fractions of the blended heavy fuels boiling off in the nozzle sac and squeezing the heavier fractions out of the nozzle holes where they carbonized. Carbonization temperatures were much higher than the measured nozzle tip temperature of 392 deg. F. (200 deg. C.). He said that he would be very grateful if the authors could provide information about the precise location of the thermocouple on the nozzle tip. His company had measured nozzle seat temperatures (the thermocouple being located within a millimetre from the nozzle seat) and temperatures obtained on one engine type, depending on engine rating, cylinder head design and fuel, had reached 464 deg. F. (240 deg. C.). This surely indicated nozzle tip temperatures of a far higher order. These nozzles had been uncooled and made from heat-resisting nitriding steel which maintained the seat hardness at elevated temperatures.

He said that he envied the authors their low maximum cylinder pressure of only 1,350 lb./sq.in. at 220 lb./sq.in. b.m.e.p. This should enable them to get away with an injector release pressure of only 2,500 lb./sq.in., resulting in a closing pressure of 1,720 lb./sq.in., thus still having a comfortable margin available above the maximum gas pressure.

In conclusion, he said that the paper would always have a place of honour in his hydraulics department, having already sent several slide rules into a semi-heated condition.

MR. J. F. ALCOCK, O.B.E., B.A., said that the piston crown was described as high-tensile steel, which term covered a lot of compositions. It would be valuable to have either the thermal conductivity or the composition.

Fig. 6 showed a wet-side temperature of over 464 deg. F. (240 deg. C.). He said that it was a rough general rule that one was apt to get coking on the surface if one went over 392 deg. F. (200 deg. C.). He wondered if this had been observed by the authors. He also asked for the velocity of the oil.

Turning to valves in cages, he said that one considerable advantage of the cage was that it very much reduced the flux from the valve seats to the cylinder head. Since the cylinder head was a complex casting, a large concentrated heat flow from the valve via the seat was undesirable. In smaller engines, which did not have caged valves, this was a very common cause of cracking between the seats. He cited the paper of Mr. Fujita* as an example of this.

Turning to valves, he said that he had noticed in the paper that while the temperature of the seat had been reduced by the cooled cage, the temperature difference between the centre of the valve head and the seat was increased. This would increase the thermal stresses, and the risk of seat cracking due to thermal stress. He gathered that the idea of sodium cooling had been discarded, but he thought it might be a useful idea, not from the point of view of cooling the valve seat, but of reducing the thermal stress. Of course, what the sodium-cooled valve did was to pass the flux back to the valve guide, but there it could be coped with quite well.

He then referred to crankcase explosions. These were practically non-existent in small engines, but they did occur from time to time in large engines and were extremely nasty things. It would be valuable to have information on this subject and to know, from the point of view of safety, what the difference was between the trunk-piston and the crosshead engines.

DR. W. P. MANSFIELD was particularly interested in the authors' lubricating oil consumption tests which were briefly described on page 336 of the paper and to which Fig. 21 referred. In some investigations on this subject on smaller engines the British Internal Combustion Engine Research Institute Ltd. had tried increasing the oil pressure by reducing the area of the bearing surface of the ring, but this had had little effect. However, changes of wall pressure, made by varying the radial thickness of the ring, had a marked effect. He said that it would be interesting to know by what method the authors had varied wall pressure, apart from the change to a spring-loaded ring which was mentioned in the paper.

MR. A. J. S. BAKER (Associate) said that the authors had produced a remarkably full description of what could only be described as an exceptionally well developed engine. Of particular interest to people connected with lubrication research was the systematic work which had gone into piston development. This work had obviously paid off handsomely in the modest temperatures so clearly denoted in Fig. 6. It would be interesting to see how far the really excellent fuel utilization rates, indicated by the exceptional fuel consumption rates, had contributed to this. For instance, metal temperature comparisons taken at the same time as the variable valve timing tests described in Fig. 10, right-hand side, would perhaps illustrate this point. Fig. 10 itself suggested that an even broader area of minimum specific fuel consumption might be possible with automatically-varied valve timing. He wondered if the authors had considered such a possibility.

Looking at the Fig. 10 data points for a constant b.m.e.p. of 200 lb./sq.in., it appeared that a fairer mean curve would have a pronounced and steepening hog rather than the sag shown by the authors. He asked the authors to justify the mean line they had postulated.

With regard to the important work which had been done to optimize fuel injection characteristics, it was interesting to consider the authors' needle lift/sac pressure relationship conclusions, in the light of the modification which appeared to have been carried out. Apparently the unloading value had been increased between (a) and (b) of Fig. 13. Did the authors attribute this fact to the reduction in incipient secondary lifting indicated in the needle-lift diagram (b)? Presumably the injection rate

* Fujita, H. 1961. "Service Records of Mitsubishi Nagasaki Diesel UE Type Engines and Improvements Made on the Engines". *Trans. I.Mar.E.*, Vol. 73, p. 37.

had also been increased and this had been accommodated by permitting an increase of needle lift. If this were the case, might not needle-lift increases have to be closely controlled in service operation, he asked. Likewise, the fuel-line pressure diagram (b) was presumably taken at the nozzle end of the fuel pipe. He wondered if it had been necessary to tune the fuel pipe length to control the magnitude of the secondary pressure wave so as to eliminate secondary injection.

He thought that the general conclusion to be drawn from the fuel injection work was that engines of comparatively low speed needed the same careful attention as high-speed engines. It would be very interesting to see a comparative investigation in certain large-bore, slow-speed engines, having several nozzles connected to a single fuel-pump element. The results of the careful work done by the authors in this direction were demonstrated in Fig. 14. By extracting data points from the fuel consumption loops, it was interesting to note that the fuelling lines for both the K engine and the K Major at different speeds were virtually identical. The fuelling lines, from around 50 lb./sq.in., b.m.e.p., on the lowest curve, were unusual in their linearity. Perhaps the authors could supply fuel consumptions at very low loads which would give a clearer indication of the likely f.m.e.p. From the data published it was evident that this must be of a very low order and comparable to that obtained with the largest low-speed engines. This point might be worth bringing out since it was fashionable to quote mechanical efficiency for large two-stroke marine engines, and for a given f.m.e.p. this would generally favour the four-stroke engine with its higher b.m.e.p.

He asked the authors to indicate whether the performance curves were obtained with the fuel described at the top of page 333, and if not, he said that he would like them to give details.

He said that the water-cooled, exhaust-valve cage and valve rotators had made a major contribution to operation with low-cost fuels. Presumably some provision had been made to prevent boiling in the small seat-cooling passages in the event of sudden shut-down, as might be expected in main marine engine application.

He thought that the notes on lubricating oil consumption were very relevant, as was the investigation on oil control by the upper scraper ring. He asked the authors to elaborate on this by indicating the degree of control exerted by the other rings in the pack. For instance, could a reduction in radial pressure of the upper scraper be tolerated by increasing the load on the lower one?

The rig described to evaluate piston-ring quality resembled a variety of test rings used for different purposes. Experience with these had indicated considerable scatter of wear results, particularly at high rates of wear. Perhaps the authors could indicate the significance of the weight loss figures they had quoted in Table IV. He wondered whether they had observed a pattern of related ring to bore wear rates for the different material combinations tested. Had any significant differences in piston-ring groove and ring side wear rates been observed when different irons were run in the steel piston crown? The authors had not shown the metallurgy of the piston crown, but other applications of high-tensile steel had suggested that steels containing appreciable nickel contents might produce increased wear rates in the presence of boundary lubrication.

Mr. J. A. Cowderoy, B.Sc. (Member) said that, as the K Major engine had been developed specifically for marine propulsion, he had been surprised that Fig. 24 did not show the performance plotted against speed on a propeller law basis. He would particularly like to see the compression pressure included in such a plot, because he had the impression that many builders of marine Diesel engines overlooked the implications of the propeller law, which related power to speed in a ship, particularly when applied to turbocharged engines. If this law were assumed to be a cube law it meant that, if the engine was developing full power at full speed, it was only required to develop as little as 12½ per cent of that power, even at half speed, and as ships not infrequently proceeded at speeds lower than full, this condition did occur now and then.

The engine referred to in the paper had been developed to run on heavy fuel oil. He thought the authors would agree that the turbocharged four-stroke, trunk-piston engine could be troublesome from the point of view of combustion when operating at low loads on heavy fuel, and from some figures he had seen for other engines, which showed a drop in compression pressure, from 665 lb./sq.in., at full load and speed, to 355 lb./sq.in., at 60 per cent speed and 22 per cent of full load, he strongly suspected that the relatively low compression pressure under those conditions was one of the principal reasons for this. Whilst combustion might be quite satisfactory, under those conditions, when the engine was new and in first-class condition, with the accumulation of wear of not only liners, but injection equipment, he thought that the low compression pressure was certainly a contributory factor. He would be glad to have the authors' comments on this.

On the question of the operational control of turbocharged medium-speed engines in ships, particularly in view of the increase in the number of ships with bridge control of the engines, he was convinced that the fuel injection pump rack position should be governed to some degree by the booster pressure. A few years ago, in a certain cross-Channel ferry, which was propelled by two turbocharged Diesel engines under bridge control, it was found, a very short time after the ship had gone into service, that the engine crankcase oil had become very dirty indeed. The reason for this was soon discovered: the bridge control of the engines had been operated on leaving harbour as if it had been an engine room telegraph, with the result that the engines smoked like chimneys until the turbochargers had time to catch up and provide enough air for clean combustion. Under these conditions the oil soon became filled with fuel soot. Instruction to the master as to the correct rate at which to increase engine power soon cured the trouble. He felt that where turbocharged engines were installed in ships, some form of control over the rate of increase of the delivery of fuel to the engine was essential.

Mr. C. C. J. French asked a question concerning thermal stress. The thermal stress rig shown in Fig. 4 was interesting and provided an ingenious method of investigating a problem which was becoming more and more important as engine ratings were increased. This rig was useful in that it was applicable to asymmetric bodies, as well as to those that were bodies of revolution. In this respect the two-piece piston shown in Fig. 6 appeared to be a body of revolution. Computer programmes were now available for calculating the thermal stress of such components. He wondered whether the authors had tried a check calculation to see whether there was any sort of agreement between the rig and a computer. His own rather limited experience so far, with a computer approach, had been more valuable in showing up limitations in the computer programme than in giving realistic piston thermal stresses, the problems being largely the rather complex shape of pistons.

Turning to the inlet-valve wear, he said that he was glad that Dr. Pope, in his presentation, had elaborated on his wear factor, which Mr. French had found somewhat incomprehensible as it stood in the paper. He agreed that lack of lubricant was the main cause of heavy inlet-valve and seat wear in turbocharged engines. It was most interesting that the authors had found thickening the head of the valve so effective in reducing this wear.

Touching on service experience, he said that two years previously a paper* had been presented, giving details of service experience on an engine of very similar size and rating. He thought that everyone looked forward to the time when the authors would be able to give comparable details of exhaust-valve life, cylinder-liner and cylinder-ring wear on the K Major when operating on residual fuels. In this connexion, if the authors were proved correct in their aim of up to 3,000 hours between servicing of injectors and exhaust valves, this would be a most valuable step forward.

* Henshall, S. H., and Gallois, J. 1964. "Service Performance of S.E.M.T. Pielstick Engines." *Trans. I.Mar.E.*, Vol. 76, p. 445.

Correspondence

COMMANDER E. R. MAY, D.S.C., R.N. (Member) wrote that it was some ten years since the Pielstick PC1 had begun to make its significant contribution to the propulsion of ocean-going ships, and during the whole of this time it had been without any effective medium-speed competitor. The K Major must now be judged by comparison with the Pielstick PC2, with which it would be in direct competition in every field.

Power for power, the British engine was rather larger and heavier than its French competitor. In some applications this would not matter very much. Commander May imagined that the relative first cost of the two engines would be very significant, assuming that they had equal ability to burn heavy fuel. The Pielstick had never been a cheap engine and, in its PC1 form, its exhaust-valve life on heavy fuel did not always prove impressive. Its popularity had stemmed from its introducing high-speed engine standards of accuracy into the marine engine field, with a refreshing freedom from the very heavy maintenance work that marine engineers often experienced on propulsion engines, less well made and indifferently developed.

Over the last few years, the major British medium-speed Diesel firms had caught up the leeway in standards of manufacture, and also had undertaken most impressive programmes of detailed development. It therefore seemed that the K Major would meet international competition successfully, would extend the market gained by the K engine, and join the Pielstick in propelling large merchant ships.

Commander May noticed that the authors had made a rather misleading reference to short-life engines being permissible in naval work. This had never been so (except in motor torpedo boats). Submarine engines were designed and produced by the Admiralty between the two wars in an attempt to produce better—not lighter—engines than those available from industry at the time. After the last war, the Admiralty worked hard to persuade industry to adopt modern standards in development and manufacture of long-life engines up to 9½-in. bore, but success was only achieved gradually and at substantial public expense.

In Germany, before and after the war, and in France at the present day, engines designed partly for naval purposes had met with widespread commercial success. This had come about through recognition that naval and commercial requirements could be designed into the same engine with advantage to all concerned.

Possibly the most remarkable feature of the K Major was that it had achieved so much while retaining cast iron for frame and bedplate. Rigidity was essential to maintain bearing oil film geometry within acceptable limits and cast iron was about twice as flexible as steel. A cast iron frame must have heavier scantlings than a steel frame, the cylinder centres must therefore be further apart, and bending moments increased in consequence. On the other hand, cast iron was cheaper than steel, and development of modern cast irons had done much to make this material more attractive. Fairbanks Morse had used cast iron extensively in their new large opposed-piston, medium-speed engine. Other manufacturers were, the writer believed, using steel for comparable engines and had also chosen the two-stroke, valve-in-head arrangement.

Soon, at least four of these valve-in-head two-stroke engines (one of them British) would be competing with the K Major and the Pielstick in the rapidly expanding world market for large medium-speed engines. It was obvious from this paper that Mirrlees had planned to secure their share of this market.

It would be interesting to know the authors' view on transmission suitable for employing, say, two K Majors to drive a single propeller shaft, and whether their company proposed to offer complete propulsion units—engines and reduction gear.

MR. G. H. HUGHES (Member) commented, in a written contribution, that the increase in power output should in no way alarm prospective users, because even the ultimate aim of 528 b.h.p./cylinder, with 250 lb./sq.in., b.m.e.p., and 1,400 lb./sq. in., peak pressure, represented only 2·98 b.h.p./sq.in. of piston crown—almost identical to the power per square inch on the crown of the Maybach engine with pistons of similar construction.

It would be interesting to know the cooling oil flow rate, (he suggested approximately 1½–[illegible] gal./b.h.p.-hr.), since crown and ring life depended on adequate cooling and, in this respect, the oil feed through the connecting rod might prove to be the limiting factor. Given adequate cooling, it was known that this form of piston would stand greater power per square inch of crown area, as shown in the two-stroke cycle Ruston and Hornsby A.O. engine, when published figures showed over 5 b.h.p./sq. in.

His company's experience of materials for such piston crowns indicated that thermal fatigue tended to become the limiting factor and this depended principally on coefficient of expansion and thermal conductivity. Had the authors considered one of the high-nickel alloys to minimize the effect of high operating temperatures, or the high-conductivity copper chromium alloys?

A further aid to cooling was increased valve overlap. Had the effect of this been explored with respect to piston crown and piston ring temperatures?

The scraper ring arrangement permitted adequate lubrication of the skirt or crosshead length of the piston, but when oil control became a problem after extended service, there might be a temptation to fit a highly-loaded ring in the skirt groove, with possible risk of seizure. To avoid such possibilities, had the authors considered omitting the skirt-ring altogether and adjusting the upper scraper ring accordingly?

It was noted that three taper-faced rings were fitted below a parallel-faced, chrome-plated ring in the top groove. There might be a tendency to blow-by during the initial running of the engine with this arrangement. Had blow-by readings been taken during test work and had any indications been noted?

An important factor in piston-ring material was compatibility with cylinder liners. Not all materials were suitable in this respect, but might be metallurgically sound and, therefore, of good quality.

It was not surprising, therefore, that a random flake graphite iron had given satisfaction in this size of engine.

With regard to cylinder liner material, was this also random flake graphite? How was the bore machined, and what type of surface was produced?

With regard to the outside diameter, was the liner free from water side attack and what precautions might be taken to deal with this possibility at the higher ratings?

On the question of heat dissipation, was it known what proportion of heat was transferred through the piston crown to the cooling oil and through the piston rings to the cooling water?

MR. J. H. MILTON (Member) wrote that it was stated, on page 327, that to produce a reliable machine one had to pro-

ceed from one successful design to the next, taking care that critical parameters proved in the original design were maintained in the next.

With regard to the critical parameters shown in Table I, it was rather surprising to see that gudgeon pin, or small-end bearings, were not mentioned, as these bearings could be troublesome and also, on occasions, connecting rods had split lengthwise through concentrated eye loading.

Perhaps the authors would care to comment on this subject, and give details of the design of their small-end bearing with particular reference to the bush—whether it was floating or not—and its material.

With regard to the piston design, as shown in Fig. 3, it would be interesting to have the authors' views on the importance of the distance from the crown to the top piston ring, and also further enlightenment on their statements that: a) heat resisting "helicoil" inserts were used to carry the studs and that these acted as a "heat barrier" for these studs; b) that disc springs were fitted under the castle nuts on these studs to increase the resilience of the assembly. Did this mean that they had accepted the fact that movement must take place between the piston crown and the body, and if so, did fretting take place with ensuing leakage of oil across the jointing face?

With further reference to Table I, it was noted that the maximum permissible bearing loads for the main bearings and bottom ends were given as 2,500 and 5,000 lb./sq. in. respectively. Some enlightenment as to how these limitations were arrived at would be of interest.

On page 336, under "Space and Weight", the authors made a good case for the cast iron engine, stating that few fabricated structures were able to avoid fillet welds in load-carrying regions, and that the fatigue strength of such welds might be as low as plus or minus 1·2 tons/sq. in., and that even butt welds had only a fatigue strength of plus or minus 3·8 tons/sq. in., compared with 5 tons/sq. in. for a good quality cast iron. If these figures were correct, it was difficult to understand why, apart from the saving in weight, so many other engine builders had adopted fabricated designs, especially as also, in the event of damage resulting from the failure of a bottom-end bolt, a cast iron engine did not suffer distortion and could usually be "patch" repaired, whereas the fabricated structure was usually distorted and had to be renewed.

It was noted that oil was used for piston cooling and lubrication and in this connexion it would be interesting to know if the authors had any relative figures on lubricating oil capacity (e.g. gal./h.p., in circuit) for the engines forming the subject of this paper, as compared with slow-speed, direct-drive Diesels.

Furthermore, in the case of direct-drive, slow-speed engines burning heavy oil, it was found essential, on account of crankcase corrosion, to isolate the cylinder bottoms from the crankcase—what precautions, beyond using an inhibited lubricant, were being taken to prevent such corrosion taking place in the engines produced by the authors' company.

In conclusion, he would be grateful if the authors could briefly state why, in comparison with the builders of large, slow-speed, direct-coupled engines, they had chosen to develop the four-stroke cycle engine instead of the two-stroke cycle engine.

COMMANDER E. B. GOOD, O.B.E., R.N. (Member) wrote that, when a new engine design was introduced, it was natural to compare its rating with those of competitors. A true comparison of ratings should take into account many design features, but an indication of the mechanical and thermal loading problems which the manufacturers had to overcome could be obtained from the output per cubic inch of swept volume and the output per square inch of piston area. These factors had been plotted against cylinder bore, for a number of modern turbocharged engine designs, in Figs. 25 and 26.

The factors for the K Major engine had been plotted at each of the development stages referred to in the paper and it could be seen that these ratings lay neither too adventurously above, nor too much below, the lines marking the upper limit of current design.

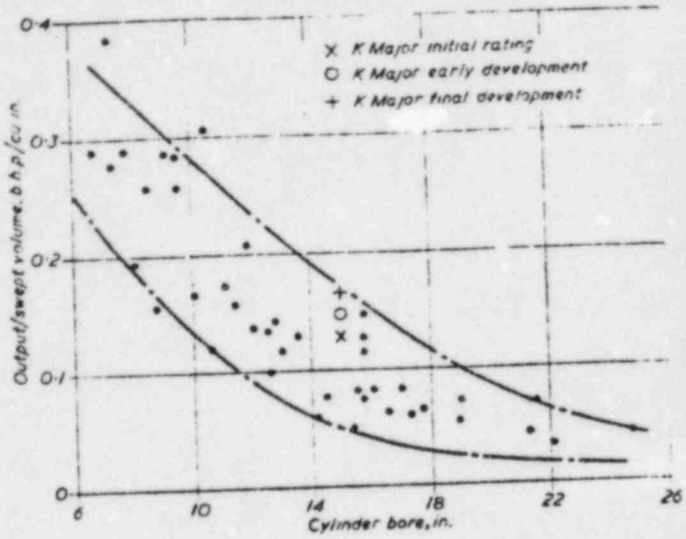

FIG. 25—*Relation between output/swept volume and cylinder bore*

Reference had been made to the use of small centrifuges mounted at the engine for the bypass purification of the lubricating oil. A marine installation of more than 3,000 b.h.p. would normally justify the use of a motor-driven centrifuge for the continuous purification of lubricating oil. Such a system also enabled the whole of the lubricating oil charge to be purified in harbour at the end of each trip, a practice adopted by many owners. Commander Good asked the authors whether they considered that the engine-mounted centrifuges would avoid the necessity for a separate motor-driven unit, particularly in a heavy fuel burning installation, and, if so, could they give an indication of the time after which cleaning of the units would be required. It was assumed that provision was made to cut off the flow to individual centrifuges, to permit them to be cleaned while the engine was running.

The attention which had been paid to the design of the fuel injectors and exhaust valves was very welcome. The maintenance of these items probably represented the largest work load for the ship's engineers. It was considered that a period of 5,000 hours between overhauls would not be an unreasonable aim for the exhaust valves.

Shipowners were becoming increasingly concerned about the noise levels in engine rooms and this was reflected in the

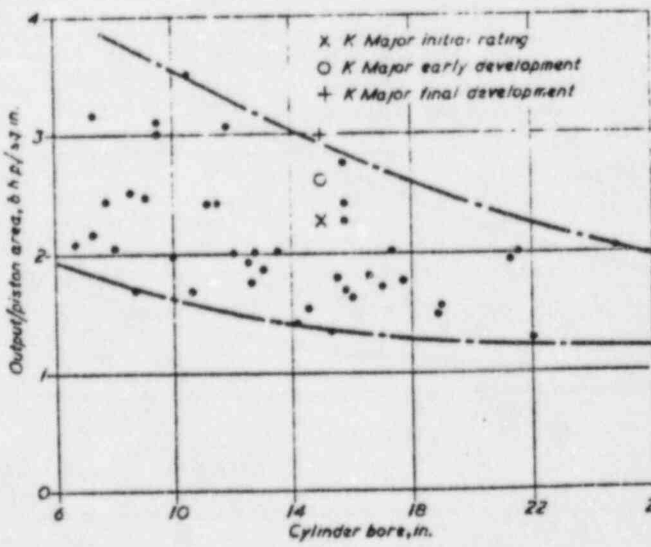

FIG. 26—*Relation between output/piston area and cylinder bore*

number of new ships which had insulated control rooms. With a maximum cylinder pressure 25 per cent greater and a maximum speed 18 per cent greater than its predecessor, the K Major might be expected to be considerably noisier. However, it was possible that the many design changes which had been made had at least partly counteracted the tendency to higher noise levels. It would be useful if the authors could provide any comparative noise measurements for the K and K Major engines.

With the advent of a new medium-speed Diesel engine, it was inevitable that designers of naval machinery installations must ask themselves whether this new engine was suitable for warships. In this respect, section f, on "Space and Weight" was relevant and one could observe that cast iron, whilst it had many admirable properties, was not the best of materials for shock. Perhaps the authors would like to comment on whether they intended to offer a naval version of this engine in due course.

Authors' Replies

Mr. Lowe (replying to the verbal discussion) referred to Mr. Cook's point about 36 cylinders being required for a ship of 18,000 horsepower and said that the paper had tried to show that the power available from that number of cylinders was now twice as great as it was a few years ago, with the same reliability and maintenance requirements.

A comparison of piston-crown temperatures, measured in the engine using templugs, with the temperature distribution produced in the rig and by an electrolytic analogue showed that in the engine, crown temperatures were rather lower than those produced in the rig and predicted by the analogue, while ring-belt temperatures were slightly higher. This difference was attributed to the rather more efficient cooling of the underside of the crown produced by the motion of the piston in the engine. The temperature distribution was proportionately the same in both rig and engine, so that thermal stress measurements should be of the right order.

The authors' company did not use a ratio as high as the 2½ to 3 times increase in fatigue strength of rolled threads over cut threads, mentioned by Mr. Cook. They used a value of 0·25 times the U.T.S. for rolled threads and 0·125 times U.T.S. for machine-cut threads, giving a ratio of 2:1 in favour of rolled threads for fatigue strength.

Both Mr. Cook and Commander Tyrrell had commented on lubricating oils for use with heavy fuel. It was difficult to be precise about the deterioration of oil lubricants because no oil specification defined accurately the requirements of a lubricant for Diesel engines. The type of oil found suitable for this engine was a "good" Supplement 1 level with a quite reasonable alkalinity. The authors used a maximum bulk oil temperature of 170 deg. F. (77 deg. C.) but it must be remembered that this implied that there would be local higher temperatures in the engine of the order of 210 deg. F. (99 deg. C.).

He confirmed that it was not the intention to imply that centri-cast rings were necessarily inferior to sand-cast rings, but the point should be made that the quality of the iron was more difficult to maintain in a centri-cast ring.

Referring to Mr. Henshall's point about specific air flow, he agreed that it was desirable to continue to increase the specific air flow as ratings increased, although the extrapolated figure in Table I showed a small reduction.

There had not been any problem with differential expansion between the steel crown and the cast-iron piston body, probably because the intensive cooling produced a low temperature at the interfaces, as could be seen in Fig. 6. The double line in Figs. 2, 3 and 6 represented a threaded portion which was used to establish the best crown diameter and was a well-known development technique.

The development engine had been run with valve guide diametral clearances as small as 0·001in. to 0·002in. with the force-lubricated guide, and a clearance of 0·003in. to 0·004in. had been arrived at for the final design. Wear rates with the lubricated guide were extremely low and a life of 10,000 hours was expected before replacement of the guide was necessary.

Mr. Gröschel had sounded a word of warning about fuel-injection test rigs and the authors agreed generally with him. Rigs were extremely useful if one was careful in interpreting the results. The suggestion that the problem of back-flow of gas from the cylinder had been prevented by reducing the reciprocating mass in the injector was quite correct and the K Majo[r] injector was of the low inertia type, as described by M[r.] Gröschel.

The fuel line pressures in Fig. 13 referred to full-loa[d] conditions and the authors considered the pressures levels an[d] the difference between line-pressure and gallery pressure to [be] quite normal in their experience.

He agreed that the higher nozzle temperature for the un[-] cooled injector with heavy fuel, in Fig. 19, was only partly du[e] to the loss of cooling from the fuel and was also a result of th[e] slower burning of the heavy fuel. The conclusion, however, w[as] unaltered that cooling of the nozzle was necessary with hea[vy] fuel and tests had been carried out, which were too extensive [to] be fully described in the paper, which showed that the tip tem[-] perature was dependent not only on engine load, but also [on] fuel temperature, water flow quantity and water temperatu[re.] The thermocouple for these temperature measurements w[as] located actually at the surface of the nozzle tip.

Mr. Alcock had asked for details of the composition [of] the high-tensile steel piston crown. This was a 55-ton tens[ile] 1 per cent Cr-Mo steel and he would gladly send exact pa[r-] ticulars of the material and of oil velocities to Mr. Alco[ck.] He pointed out that the 464 deg. F. (240 deg. C.) maximu[m] temperature in Fig. 6 was a bulk temperature of the mater[ial] and that the surface temperature in contact with the oil wo[uld] be lower. There had been no signs of oil coking on the surf[ace] of the cooling chamber.

He agreed that it was very critical to choose the ri[ght] material, not only for the seat of the valve, but for the va[lve] itself. The relative coefficients of expansion of these t[wo] materials and their conductivity were most important.

The subject of crankcase explosions was a general one a[nd] not confined particularly to the engine under discussion. [Fig.] 1 showed that the engine was fitted with explosion doors. [The] company had experienced one or two cases, in the last se[ven] or eight years, of crankcase explosions in other types [of] engine where the explosion doors had worked satisfactorily.

Referring to Dr. Mansfield's question about the met[hod] of varying the wall pressure of the piston rings, he said t[hat] this had been done both by altering the bearing area of a gi[ven] ring and also by re-designing the ring.

Mr. Baker had suggested that there should be autom[atic] timing on an engine. This was an attractive idea, particul[arly] if combined with automatic timing of fuel injection, but [was] very difficult to achieve. If some simple and foolproof wa[y of] doing this could be found it would be a real achievemen[t.]

He said that he could not justify the mean line for [the] 200 b.m.e.p. data on the right-hand diagram of Fig. 10. [The] measured points indicated the "hog", suggested by Mr. B[aker,] but he could not explain why this should be so and hence [he] merely indicated the downward trend which he thought t[o be] of greater significance.

Referring to the injection diagrams, he said that the [pre-] loading volume had been increased from (a) to (b) in Fig[.] and had a very strong effect on the tendency for secon[dary] injection. The injection rate had also been increased an[d, as] already stated in the reply to Mr. Gröschel, a low-inertia [type] of injector had been adopted. The fuel line pressure [was] measured halfway along the injection pipe in each case an[d]

injection pipe length was the minimum possible for the engine. Mr. Baker's remarks about the Willan's lines were quite valid and, in fact, the twelve-cylinder engine had shown a mechanical efficiency of 93 per cent. The performance curves were obtained using distillate fuel.

The piston-ring pack, described in the paper, was deliberately designed so that the scraper ring above the pin was more severe than the lower scraper ring. The authors considered it important to maintain an adequate oil film over the body of the piston so that the lower scraper ring was only intended to remove excess oil to prevent the upper ring becoming flooded. The weight loss figures, in Table IV, were only significant in comparison to one another as they were obtained from a rig running completely unlubricated and could not be related to conditions in an engine cylinder. There had not been any controlled tests to measure wear rates in the grooves of the steel crown with piston rings of different irons.

In answer to Mr. Cowderoy's question about compression pressures, he said that, at 525 r.p.m. and full load, the compression pressure was 900 lb./sq.in., coming down linearly with horsepower to 380 lb./sq.in., at 100 r.p.m., i.e., effectively no load. The compression pressure, corresponding to the 60 per cent speed, 22 per cent load condition, quoted by Mr. Cowderoy, was 430 lb./sq.in. He agreed with the contributor that low compression pressure, or rather compression temperature, could contribute to inferior combustion in a worn engine at low load.

Mr. Cowderoy's suggestion that marine propulsion engines should be treated like locomotive engines, from the point of view of preventing the driver from accelerating too fast, was a good point. The necessity for the turbocharger to accelerate was often overlooked when rapid increases in load were called for, and it might well be necessary to apply the locomotive type of fuel rate control to marine engines.

In reply to Mr. French, he said that the computer approach to piston crown thermal stresses had been to construct an electric analogue which, as mentioned earlier, gave quite good agreement with the thermal rig. Work was currently in progress on a digital programme which would calculate stresses directly, whereas the analogue only gave temperature distribution from which stresses could be calculated.

The numerical value of the wear factor for inlet valves obviously applied to one's own engines, but a manufacturer could apply the formula to obtain values from his own engines. In this connexion, he said that the experimental work, from which the wear factor was developed, was described fully in reference (2).

Dr. Pope said that before the meeting closed he would like to make one or two comments about some generalities which had come up during the discussion.

He said that they were getting to the stage in the medium-speed engine industry where the research and development effort of the engine builders was outstripping the component builders, and he could foresee, in the not too distant future, that engine development might well be held up because of lack of blowers and injection equipment. He hoped the supply industry would persevere with enthusiasm for the highly-rated medium-speed Diesel engine as much as the engine builders were.

He thought that the problem of the maintenance of medium-speed Diesel engines in the marine world should be judged objectively. Obviously, in the medium-speed engine one was going to have more parts, but there was a world of difference between handling a 15-in. piston and handling a 30-in. piston. The factors involved were not just the number of parts, but the way in which they could be manipulated and a statistical analysis of what were the major and minor faults. His view was that if this analysis were carried out scientifically one would find that the medium-speed engine could stand on its own, even with regard to maintenance.

With regard to cast iron, he said that each problem must be judged on its merits. The fatigue strength of a good cast iron was near to that of a good steel. There were other advantages in using cast iron. One knew that one could obtain good castings with cast iron for the size of engine he was discussing and it was an easy material to handle. Size for size, his experience had been that cast iron came out cheaper, therefore if one had a material which was as good as another and was cheaper, one had to have a very good reason for not using it. He could only see one reason for precluding its use and that was if weight were a predominating factor. However, when one considered the rest of the engine room equipment, the tankerage and the fuel capacity, one would find that there was a difference of one or two per cent between a welded design and a cast iron design, so that, for commercial shipping, this was a marginal consideration.

He pointed out that with an in-line engine with an underslung crankshaft, one could have a very nice stress line pattern which, on the drawing board, looked very attractive and almost impossible to improve upon, but when one came to a "V" engine with side by side connecting rods, so that the opposite liners could not be in line with each other, the stress pattern did not look quite so elegant. He accepted that the cast iron bedplate was more difficult to design because the stress pattern was more complex, but once one had designed it and got a good design one was simply comparing one good design with another. It was also a question of continuing from a well-tried engine to the next generation, without departing from well-proven design principles. His company had now completed its one thousandth K engine and had over 300 of them in marine application. Over a third of the engines were running day in and day out on heavy fuel.

AUTHORS' REPLY TO WRITTEN CONTRIBUTIONS

The authors wrote that they entirely agreed with Commander May's appreciation of the rapidly increasing demand for large medium-speed engines. They had deliberately restricted the paper to the development of the K Major engine itself but their company was certainly proposing to offer complete propulsion units for geared installations with either single or multi-engine inputs.

In reply to Mr. Hughes, the authors wrote that the piston cooling-oil flow was 1·7 gallons/b.h.p.-hr. at the current full-load rating. The cooling design was such that the operating temperatures of the piston crown were well within the thermal fatigue limit of the steel used, as could be seen from the isotherms of Fig. 6. The work that had been carried out on the optimization of valve timing and combustion characteristics, which was described, had been aimed at obtaining the best thermal efficiency from the engine and not at reduction of component temperature by scavenge air cooling, which they considered was a relatively inefficient method of controlling component temperatures. The temperature of the critical parts of the engine, such as exhaust-valve seat, injection nozzle, and top piston-ring groove, was controlled by direct cooling.

As mentioned in the reply to Mr. Baker, the lower scraper ring in the skirt was relatively mild and the upper scraper more severe to ensure adequate lubrication of the piston skirt. The taper-faced compression rings were an advantage in initial running as they bedded in very quickly on a narrow circumferential band. Initial running-in had been the subject of a good deal of investigation on the test bed and the best results had been obtained with a relatively rough liner surface, which was honed to a C.L.A. of about 100μ, the liner being a random flake graphite iron, slightly softer than the piston-ring material.

Calculations of liner frequency and vibration amplitude were included at the design stage to avoid the possibility of water-side attack. The heat dissipation through the piston rings to the cooling water could not be directly measured in the engine because of the heat received directly by the liner from the combustion gases. By reproducing temperatures and total heat flow in the rig, the proportion of heat flowing to the cooling oil was about 75 per cent of the total and the heat to the liner was 25 per cent.